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Foreword

VOLUME 52 of the TRANSACTIONS OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS contains 35 individual sections published during 1930 under the sponsorship of the Society's 15 professional divisions, as well as its annual Record and Index. The technical papers and reports that make up this volume represent the Society's annual contribution to the permanent record of mechanical-engineering achievement. Most of these papers and reports were presented at meetings of the Society and of its professional divisions and local sections. Other papers published by the Society but not included in the Transactions may be located by means of the Record and Index which forms a part of this volume.

The papers published in this volume were issued originally in pamphlet form. These pamphlets appear in sections, each section being sponsored by one of the Society's professional divisions. A member of the Society may register in any three of the fifteen professional divisions and receives in pamphlet form as they are issued the sections of Transactions sponsored by those divisions.

All sections of the Transactions are bound together at the end of the year for the convenience of libraries and of engineers who wish all of the papers in permanent form. Copies of the bound Transactions will be found in depositories located in selected engineering, university, and public libraries throughout the world. A complete list of these depositories will be found on page 97 of the Record and Index. Copies of the bound Transactions have also been set aside for sale.

At the end of Volume 52 will be found the Record and Index of the A.S.M.E. for 1930. This is the permanent record of the Society's activities for the year, and includes lists of committees, reports of meetings and committee activities, and memorial notices of deceased members. The indexes include those of special publications, and of *Mechanical Engineering*. Printing of the Record and Index in this form was ordered by the Council of the Society, and returns to the Transactions material of permanent historical value which has been issued for the past three years (1927–1929) as a separate publication.

NUMBERING AND ARRANGEMENT OF PAPERS IN COMPLETE TRANSACTIONS

In this volume the papers of each section are grouped together and are numbered serially. The different sections follow each other alphabetically. Each paper is designated by a symbol composed of key letters followed by the volume number and paper number. The letters refer to the section of the Transactions to which the paper is assigned. Thus AER refers to the Aeronautic section, IS to the Iron and Steel section, etc. Each of the two books forming parts I and II of Volume 52 contains a complete index of subjects, authors, and discussers to both parts of the volume so that either may be consulted. The arrangement of the index and designations for locating material are explained at the beginning of the index.

CONCERNING BLANK PAGES

The papers in each section of the Transactions are printed separately so that they may be available as reprints. This makes it necessary to print each paper so that the number of pages is a multiple of four. Because of this, blank pages will be found at the ends of papers which do not completely fill the printer's form.

THE PUBLICATIONS COMMITTEE.

Progress in Aeronautics

Contributed by the Aeronautic Division

Executive Committee: Edwin E. Aldrin, *Chairman*, Alexander Klemin, *Secretary*, Elmer A. Sperry, Wm. F. Durand, Charles H. Colvin, and Thurman H. Bane

Power Plants[1]

AN EXAMINATION of Table 1 indicates that the development of airplane engines during the past year has proceeded apace. The radial air-cooled engine is still the most widely manufactured and used, but it is being pushed in the low-horsepower class by air-cooled in-line engines of various makes.

The development of new engine types is proceeding very slowly. The Packard Diesel made a spectacular non-stop flight from Detroit to Langley Field in May, but since then has been kept well under cover. The Curtiss H-1640 (Chieftain) engine has not yet progressed beyond the development stage. The Wright V-1460, a twelve-cylinder inverted-vee air-cooled engine, is being installed experimentally in some Army Air Corps airplanes.

The Pratt and Whitney R-1340 (Wasp) has progressed to the Series C engine, which is slightly heavier than the series B, but has increased durability. The Pratt and Whitney Hornet has progressed from the R-1690 to an R-1860 (Hornet Series B) engine of 575 hp., which is already in service in both military and commercial airplanes.

The Wright Aeronautical Corporation has brought out the new J-6 (Whirlwind) engines, the R-540, R-760, and R-795. The J-5 has been withdrawn from production and the J-6 engines are being used extensively. The Wright R-1750 (Cyclone) has undergone various improvements. The Wright Company is bringing out an American edition of the De Havilland Gypsy.

The Curtiss Company has installed several of their water-cooled V-1570 (Conqueror) engines in commercial airplanes. The Curtiss R-600 (Challenger) in a Curtiss Robin airplane set up a world record for sustained flight of over 400 hours at Saint Louis. The Curtiss Company has recently produced a small 125-hp. inverted 6-in-line air-cooled Crusader engine.

One of the outstanding engine developments of the past year was the release, by both the Army Air Corps and the Navy Bureau of Aeronautics, of information regarding the work done on high-temperature cooling of liquid-cooled (hitherto called "water-cooled") engines. The most efficient cooling medium was found to be ethylene glycol (Prestone), which can be used with a jacket outlet temperature of 300 deg. fahr. It has the advantage over water cooling of permitting a 75 per cent reduction in radiator area, due to the greatly increased temperature difference between the cooling medium and the air.

Another outstanding power-plant development of the past year was the development of the venturi type of cowling for radial engines. The use of this cowling removes the greatest objection to the radial air-cooled type of engine—its excessive drag. In the free-for-all high-speed race at the National Air Races in Cleveland last August, radial air-cooled engines with venturi cowling placed one, two, three against the field.

An interesting trend in airplane power plants is the increasing use of geared engines, both direct-air-cooled and indirect-air-cooled (liquid-cooled), for the purpose of reducing propeller speeds to a point where the propeller can operate at its optimum efficiency. With the problems involved in the use of reduction gearing successfully solved, it seems logical to expect that engine speeds will increase to a point comparable with those of automobile engines, thus giving increased thrust horsepower for a given engine displacement and size.

[1] Prepared by Lieut.-Commdr. James M. Shoemaker, U. S. Navy Washington, D. C.

TABLE 1 AIRPLANE ENGINES—APPROVED-TYPE CERTIFICATES ISSUED IN 1929 BY THE U. S. DEPARTMENT OF COMMERCE

Key

4—number of cylinders
R—radial (arrangement)
V—vee
L—in line
I—inverted
A—air cooled
W—water cooled
G—gear drive

A.T.C. No.	Date	Description		Rated hp.	Rated r.p.m.
10	10-11-28	Curtiss D-12	12VW	435	2300
11	4-4-29	Dayton Bear	4LA	100	1500
12	11-30-28 1929	LeBlond 60	5RA	65	1950
13	1-26	Wright J5 Whirlwind	9RA	220	2000
14	1-22	Pratt & Whitney Wasp	9RA	450	2100
15	12-19-28	Pratt & Whitney Hornet	9RA	525	1900
16	2-5-29	Axelson Axelson	7RA	115	1800
17	1-26	Wright Cyclone R-1750-A	9RA	525	1900
18	1-26	Packard 3A-1500 Direct	12VW	525	2100
19	1-26	Packard 3A-2500 Direct	12VW	800	2000
20	2-6	LeBlond 90	7RA	90	1975
21	2-15	Wright J6 R-975	9RA	300	2000
22	3-11	Arnold Harris	8VW	90	1400
23	3-27	Wright J6 R-540	5RA	165	2000
24	6-3	Alliance Hess Warrior	7RA	115	1925
25	6-8	Michigan Rover	4LAI	55	1900
26	7-9	Wright J6 R-760	7RA	225	2000
27	8-14	Lycoming R-645	9RA	185	2000
28	8-30	Pratt & Whitney Hornet, R-1860 Series B	9RA	575	1950
29	8-30	Pratt & Whitney Hornet, R-1690-A Geared 2:1	9RA	500	1900
30	9-13	American Cirrus Mark III	4LA	90	2100
31	9-23	Aircraft engines LA-1 A.C.E.	7RA	140	1800
32	10-5	Continental A70	7RA	165	2000

Airplane Design and Construction[2]

In 1929 airplane design and construction made rapid advances, more so, perhaps, than in any of the immediately preceding years. Unlike the year before, a number of really new types were brought out and successfully proved. The decrease in plagiarism of designs and the increase of individual effort were especially noticeable. A number of new methods of construction were developed, and design with a view to mass production was vastly improved. As in the past few years, much attention was paid to improvements in details, both in the aerodynamical properties of airplanes and in their structure. Many new large airplanes were completed as well as many exceptionally small ones. The number of open-cockpit biplanes still predominated, as in the past, but did not increase as it had in the years past but rather decreased. On the other hand, much more attention was paid to the development of flying boats, seaplanes, and amphibians.

In design, the influence of the Aeronautics Branch of the Department of Commerce was shown by the considerable decrease in "fly-by-night radical and revolutionary designs," and as well in general improvement in all planes with respect to thoroughness in engineering and proper attention to details.

In the first half of 1929, 72 new approved-type certificates were awarded by the Department of Commerce as against 59 in the last half of 1928, an increase of 22 per cent. In the third quarter of the year 52 certificates were awarded as compared with 36 in each of the two previous quarters, an increase of 44.5 per cent. It should be remembered that these certificates signify

[2] Prepared by Richard M. Mock, Aeronautical Engineer, Bellanca Aircraft Corp (at the time of writing, on leave and with the Ernst Heinkel Flugzeugwerke, Warnemünde, Germany).

only the approval of new types for production and do not represent the quantity of planes of each design approved.

As to the detail design of airplanes produced in 1929, advance in aerodynamic "cleanness" is immediately apparent.

More than ever the tendency has been toward the monoplane rather than the biplane, and again toward the "clean" cantilever monoplane. Besides the efforts made to reduce resistance through the reduction of frontal area, much attention was paid in the past year to the elimination of induced drag caused by the mutual interference between the various exposed parts of the airplane. The efforts in this direction have resulted especially in cleaner landing gears, with more attention given to the relative location of the individual members. In some landing gears these members are completely eliminated by supporting the wheels by a cantilever beam. The landing gear of the Bellanca Pacemaker is an example of the truss type, much attention being paid to the mutual interference of the individual members. Many production designs now include as standard equipment fairings for wheels and tires, while on the Eaglerock Bullet the resistance of the wheels is almost practically eliminated by retracting the greater portion of them into the wing.

In the design of wings there has been a slight tendency toward increased aspect ratio to give better cruising performance. This has been more apparent in some European designs than in American ones. Little progress was made during the year with regard to the development of better profiles for wing sections. This was chiefly due to the dependence of commercial organizations upon government laboratories for information. However, at present many manufacturers are developing aerodynamical laboratories of their own, which should effectively advance the investigation of their individual problems.

Though the N.A.C.A. type of engine cowl was developed in 1928, it was not until the past year that it was really tried on a commercial scale. With the increased use of this cowl for radial air-cooled types and the development of clean air-cooled in-line engines, designers have paid a great deal more attention to cleanness in fuselage lines, resulting in somewhat rounded fuselages, many of which are oval in section. An excellent example of the result of good streamlining is the low-wing Travel Air monoplane of conventional American construction of welded steel-tube fuselage and wood wings, all fabric-covered. This plane attained a speed of 200 m.p.h. Also worthy of mention is the Eaglerock Bullet, a land plane with cantilever low wing and retractable landing gear.

Aerodynamically, most planes were quite conventional except for the Bellanca Tandem and the Furnic Ente. The Bellanca Tandem opens an entirely new field of development for multi-engined planes. The engine installation is new, while the wings are a further development of the Bellanca Roma or K-type brought out in 1928. The Furnic Ente is a twin-engine design differing from the usual Ente type in that it has two stabilizers, one in front and one behind.

From a structural point of view there has been an increasing use of duralumin, and especially of the corrosion-resisting alloy Alclad. In the United States there has been little or no work done with light-gage sheet steel, especially of the so-called stainless variety. This is used extensively in England and is being tried experimentally by Heinkel in Germany.

One of the most interesting developments in wing construction is the Barling monoplane of Nicholas-Beazley. Practically all American wings employ the two-spar construction, but one of the leading manufacturers uses three spars. In Europe there has been considerable development in single-spar metal construction, especially the English "mono-spar" and the German B.F.W. commercial planes.

More planes have been produced with dural fuselages than heretofore, and the number of dural tail surfaces has also increased quite rapidly.

From a production viewpoint there have been many advances. The development of the Consolidated Fleet Junior is the most outstanding example of a good production design. Also in this class is the new Great Lakes training plane.

Toward the close of the year there was evidence of an increased activity in the development of seaplanes and flying boats. This branch of aeronautical development has been somewhat neglected in the United States, except by Loening, Sikorsky, and Ireland. However, 1929 brought the development of the large Consolidated Commodore, the Hall, the Keystone-Loening Commuter, and the Great Lakes Amphibian, as well as a number of others that are not yet in production. Many European manufacturers have arranged for American production of their products. Savoia-Marchetti flying boats are already being built. Three German manufacturers are planning American production; Heinkel has developed an amphibian of medium size expressly for the American market, while Rohrbach has announced an American company presumably to produce the Roman and Rocco types. Dornier, too, has an American connection for fabrication of the Super Wal and perhaps the 12-engined giant DO-X, which last fall carried 169 people, establishing a world's record. Also planning American production under license are the English Blackburn and the French Schreck companies, both experienced organizations in flying-boat development.

In America the large passenger land transport plane has been receiving much attention. In 1929 the Curtiss plane Condor, the Fokker F-32, and the twin-engined Burnelli were developed, while production was increased on the Ford tri-motor, the Fokker tri-motor, and the Keystone Patrician. At the time of this writing the four-engined Fokker F-32 is the largest passenger land plane, carrying 32 people, and work is being rushed in Germany to complete the 50-passenger Junkers J-38, a design closely resembling the large Junkers G-31.

As regards cabin passenger planes, it is the writer's opinion that the German transports are much more comfortable, though great progress is being made in this direction in America.

All over the world there has been a tendency to increased power in airplanes, and in many cases, instead of being used to provide greater safety or better performance, it has been employed to overcome difficulties in design in what would otherwise be called exceptionally poor airplanes. The increased use of higher-powered engines, especially the Wasp, Hornet, and Cyclone, is noticeable, while in Europe builders have gone to even higher powers, employing 700–800-hp. and even 1000-hp. water-cooled engines on commercial airplanes.

Aerodynamics[1]

The past year has seen practical progress in the art of streamlining. Commercial designs have shown a tendency to imitate racer practice in reducing parasite resistance to a minimum. For example, commercial designers are using retractable landing gears (as in the Alexander Bullet) which have a resistance approximately 50 per cent of that of the fueslage. Monocoupe fuselages of nearly perfect streamline form are being built in both wood and metal. The filleting of wings into fuselages, and of struts into wings, is being considered as worthy of attention from a resistance point of view. Without evidence of systematic wind-tunnel experimentation, attempts are being made to decrease wheel resistance by providing wheels with fairings. The N.A.C.A. or venturi cowling has been successfully employed on a number of designs to reduce the head resistance and increase speed, although designers of single-engined planes are not adopt-

[1] Prepared by Alexander Klemin, Daniel Guggenheim School of Aeronautics, New York University. Assoc-Mem. A.S.M.E.

ing this type of cowling any too rapidly as it undoubtedly impedes vision. In England the Townsend ring, an annular surface placed round and at some distance from the cylinder heads, is apparently producing real aerodynamic improvement, and the device is of the simplest possible character.

In the design of airfoils, particularly in England and France, it is now more frequently the practice to design airfoils theoretically, and to employ airfoils with a constant center of pressure, partly with a view to increasing the stability of the airplane and partly with a view to decreasing the stresses on the spars. The successful employment of constant-center-of-pressure wings has also led designers to think more seriously of the single-spar cantilever monoplane. With a constant center of pressure achieved, single-spar construction offers decided possibilities. Designers are also giving much more attention to the careful aerodynamic design of cantilever monoplane wings. Besides taper in plan and thickness so as to provide both greater aerodynamic efficiency and greater strength at the root of the wing, it is now customary to give consideration to the center-of-pressure travel. By suitable design, and appropriate choice of sections along the span, it is apparently possible to produce tapered wings of real efficiency, high lift, and constant center of pressure. At the same time, by using constant-center-of-pressure sections at the tip, it is possible to minimize the dangers of flutter.

As regards control surfaces, a number of tendencies may be noted. Floating wing-tip ailerons, always in the line of wind, no matter what the angle of incidence of the main wing may be, have been tried out in a number of wind tunnels. The advantage of such ailerons lies in the fact that, even if the main wing is at or near the stall, the ailerons are always at zero near the wind, and therefore maintain effective rolling moment, while the adverse yawing moment either disappears or becomes negligible. In the Guggenheim Safe Aircraft Competition, the Curtiss Tanager is employing this type of control, whereas the Handley Page entry employs the so-called slotted aileron, in which the Handley Page automatic slot is placed at the leading edge of the wing and helps the lateral control at the stall. The so-called slot-and-flap aileron has been universally adopted on all machines used by the British Government.

The Handley Page Company has successfully developed a so-called "interceptor," in which the action of the ailerons working with a front slot may be still further strengthened. In this device the front slots may remain automatic, but the upgoing aileron raises a "spoiler," a flat plate normally lying flush with the wing. This spoiler still further decreases the lift on the side of the upgoing aileron, and also increases the drag on the side of the upgoing aileron.

The Safe Aircraft Competition has brought to light a series of interesting attempts to improve the airplane at the lower range of the scale. The special devices employed in wing design include: The automatic front slot working in conjunction with rear flaps, with which it is mechanically connected (Handley Page); the automatic front slot working in conjunction with a manually operated rear flap (Curtiss); variable camber and area (Burnelli); variable camber and incidence (Wentworth); and conversion of a monoplane into a biplane of very small gap (Cunningham-Hall).

The realm of aerodynamics, broadly speaking, may be considered to comprise the activity which has been shown in the design of variable-pitch propellers. Mechanical control, electrical control, and hydraulic control are now being supplemented by efforts at purely automatic control, with action dependent on the fine correlation of aerodynamic and dynamic forces.

The establishment of the Autogiro on American soil has led to increasing interest in this form of aircraft. One of the main difficulties previously experienced with this machine has been the slowness of starting of rotation of the main blades. By raising the tail surfaces in appropriate fashion so as to break the symmetry of flow, it is now possible to secure blade rotation with much greater rapidity.

The Isacco Helicogyre may perhaps be said to be a development of the Autogiro. It involves freely hinged blades as does the Autogiro, but the lifting blades are driven round not by windmill action alone, but by the placing of auxiliary engines near the tips, which actuate auxiliary propellers. It is hoped in this fashion to augment the admirable qualities of the Autogiro by increasing steepness of climb and rapidity of take-off.

Air Transport and Other Civil Flying[4]

Continuing the development described in the Progress Report for 1928, further extensions of air routes have been made in the United States. Perhaps the most outstanding feature in this connection has been the extension of the U. S. air lines to South America. This began with the extension of the Pan-American Airways route from Havana to Central America and thence through affiliated companies on the West Coast. At the same time a competitive group has obtained mail contracts from certain South American nations, has organized its personnel, and is just about to place into operation a competitive route running down the East Coast of South America to Rio de Janeiro and Buenos Aires. The traffic carried by practically all lines in the United States has undergone considerable increase, and this increase shows no signs of slacking up. While the air mail has had a very consistent growth, the outstanding feature on the traffic side has been the changed attitude of the public toward passenger services and the increased willingness of passengers to fly. As a result of this, passenger traffic, which was formerly lagging behind that of Europe, is now experiencing a very healthy growth and promises to become, within a short time, a most important source of revenue for the operators of air routes. Similar increase has been noted in the miscellaneous flying services and in the operation of flying schools. Because of its bearing upon transport operations, it is noted here that the increase in pilots is keeping up quite closely with the increase in airplanes. Thus it is evident that there will be no shortage of pilots as a whole, although for a time there may be some shortage of the very experienced types of pilots.

Among the elements of economic importance to the operators of air routes might be noted the recent tendency toward consolidation which has resulted in the merging of a considerable number of smaller operators into a lesser number of substantial groups controlling connecting routes. This development is somewhat comparable with the consolidations of the early days of railroading, and the air routes are becoming combined into well-organized systems somewhat similar to those which now exist in railroading. Incidental to this development, a considerable volume of additional capital has been drawn into the operating side. As this is being written, there is under consideration by the U. S. Post Office Department the possibility of some adjustment of air-mail contracts which may affect those operators holding contracts at the higher rates. The indications are, however, that the attitude of the Post Office Department will be quite reasonable and that no operator will be expected to accept a readjustment of rate which forces him to run at a loss, if this can possibly be avoided.

The movement toward the creation of private flying clubs, which began here within the past two years, is now producing some tangible results and is thus providing facilities which should give great encouragement to the private owner of airplanes and result in further increase in private flying.

[4] Prepared by Archibald Black, President and General Manager, Black & Bigelow, Inc., Air Transport Engineers, New York, N. Y. Mem. A.S.M.E.

Among the technical developments affecting air-transport operation might be noted the increase in size and horsepower of airplanes, of which the 32-passenger Fokker and the 50 passenger Dornier DO-X are recent examples. This is a very natural trend due to increased traffic and to the demand for still higher speed. As the traffic grows still further, similar further increases in size of airplanes may be expected until the equipment in use attains capacities very much greater than at present. The recently developed N.A.C.A. cowling is of most important economic advantage to the airway operator as it increases the speed of his airplanes without affecting the gasoline consumption. This is important from three viewpoints: First, it enables him to speed up his operations; second, it enables him to get more mileage with the same flying cost, or, conversely, a lower flying cost for the same mileage; third, it has a still further advantage in making it possible to get more miles per year with the same equipment. Recently, some experimental flights were made with the Opel rocket-propelled airplane. At present, however, this is purely a scientific toy.

TABLE 2 MILEAGE AND TRAFFIC STATISTICS FOR PRIVATELY OPERATED AIR-MAIL ROUTES WITHIN THE U.S.A., FOR YEAR ENDING JUNE 30, 1929[1]

Month (all routes)	Length of routes, miles	Miles of service actually flown	Total weight of mails dispatched, pounds	Amount paid to contractors
July, 1928	12,021	653,883	214,572	$445,238.41
August, 1928	11,977	731,714	419,357	821,005.01
September, 1928	11,977	749,156	423,838	843,691.44
October, 1928	12,169	784,663	465,688	915,998.56
November, 1928	12,504	731,027	425,405	851,390.06
December, 1928	13,791	891,811	541,632	1,094,782.54
January, 1929	14,060	795,293	488,709	955,114.33
February, 1929	14,060	778,329	434,260	865,042.52
March, 1929	14,264	912,760	524,537	1,047,657.78
April, 1929	14,400	885,737	508,672	1,003,718.32
May, 1929	14,334	1,135,953	588,428	1,160,089.23
June, 1929	14,345	1,148,403	598,494	1,161,461.00
Totals	159,902	10,198,729	5,633,592	$11,165,189.20

[1] Compiled from Post Office Department figures by Black & Bigelow, Inc., New York, N. Y.

Within the past twelve months the Department of Commerce has made certain modifications in its regulations governing the licensing of pilots. These changes constitute a general tightening up of the previous regulations, so that a pilot who is licensed to fly a certain type of aircraft will hereafter be required to take an additional examination to fly a type very much larger than the one on which he obtained his license.

As this report is being written, the manufacturing side of the industry is undergoing a certain amount of readjustment, due to an increase of production at a greater rate than the increase in sales, combined with the seasonal drop in sales which usually occurs in the fall. This addition is most noted in the case of the smaller types of airplanes such as are sold to private owners. While this readjustment period will be a painful one for certain of the manufacturers, particularly those who are not very well financed, it is a passing phase, and the manufacturing side of the industry and should again be in sound condition when the seasonal increase in sales develops in the spring of 1930.

One of the interesting developments of the past year was the entrance of General Motors Corporation into the aviation field through the purchase of a substantial interest in the Fokker Company, and by close association with the Bendix Aviation Corporation through ownership of about twenty-five per cent of its stock.

Table 2 shows the continued growth of air-mail traffic and mileage development in the United States, while Table 3 shows the air-transport mileage of the world. It will be noted from this latter table that the lead which was attained by the United States between 1925 and 1926 has not only been held but has been very substantially increased in the past year.

TABLE 3 AIR-TRANSPORT MILEAGE OF THE WORLD—OPERATIONS OF SCHEDULED SERVICES IN AIRPLANE-MILES

	1919	1920	1921	1922	1923	1924	1925	1926	1927	1928
Australia	....	....	.....	.....	235,582	262,895	352,847	417,970	453,580	641,831
Austria	....	....	.....	.....	.....	.....	.....	.....	245,043	383,002
Belgium	....	....	.....	.....	.....	160,000	144,500	.....	.....	.....
Canada	....	....	.....	.....	.....	294,778	446,648	631,715	.....	.....
Colombia	....	2,687	53,650	125,678	134,375	169,500	183,206	.....	.....	.....
Czechoslovakia	....	....	.....	.....	.....	126,400	48,300	170,895	257,888	605,314
Denmark	....	.....	.....	.....	56,000	.....	114,500	126,730	116,798	122,215
France	....	505,500	1,460,000	2,015,000	2,115,000	2,249,000	2,946,000	3,243,900	3,755,369	3,753,133
Germany	.....	.....	.....	842,986	885,881	1,860,000	3,070,000	3,816,144	5,921,593	7,030,565
Great Britain	168,000	644,000	225,000	717,000	943,000	936,000	862,000	840,000	769,000	.....
Italy	....	....	.....	.....	.....	.....	.....	.....	824,474	1,236,913
Mexico	....	....	.....	.....	.....	.....	.....	96,800	146,400	.....
Netherlands	....	50,850	217,000	246,200	336,000	482,800	679,753	597,500	813,510	1,007,920
Poland	....	....	.....	66,293	142,057	233,923	465,847	.....	654,873	630,175
Soviet Russia	....	....	.....	.....	.....	288,600	292,595	311,000	.....	.....
Sweden	....	....	.....	.....	.....	69,280	248,610	222,000	206,766	208,729
Switzerland	....	....	.....	.....	.....	268,400	87,427	210,340	459,720	330,496
United States	393,066	880,028	1,828,354	2,329,296	1,743,030	2,220,761	2,910,611	4,407,263	6,009,226	9,888,307

The recent experiments of the Guggenheim Fund in connection with fog flying are being carefully watched by all operators and have been very promising up to date. Because of these experiments, the coming year may find the operator in a much better position to master the question of fog than he has been at any time heretofore. The use of radio in airways in the United States is becoming much more general, and several operators are now depending upon radio communication with airplanes in flight for the purposes of furnishing weather information and keeping in touch with pilots en route.

The railroads are now showing considerable interest in air-transport development, and in a few cases joint air-rail services have already been organized in which the railroad and airplane operations are coordinated and through tickets sold so that the passenger can make a through flight, using the train for part of the way and the air-service company for the remainder of the trip.

Progress of Naval Aviation[5]

Naval-aviation progress during the past year has continued to demonstrate the wisdom and soundness of the Navy's Five-Year Building Program as authorized by act of Congress approved June 24, 1926. Progress toward the accomplishment of this program is continuing satisfactorily, and it is important that this program be completed as authorized, on time and without material changes.

During the year, naval aviators flew a total of 205,421 hours. The average pilot flew about 216 hours, or about 19 per cent more than in the previous year. In the Fleet approximately 285 planes were being operated. Scouting, torpedo and bombing, observation, and fighting planes were carried on the *Lexington*, *Saratoga*, and *Langley;* and fighting and observation planes were operating from catapults on the battleships, while each light cruiser operated two observation planes.

The year opened with the Fleet's annual cruise in southern waters, participated in, this year, for the first time, by the two carriers, *Saratoga* and *Lexington*. Together with the battleship

[5] Prepared by Rear-Admiral Wm. A. Moffett, U. S. N., Chief of the Bureau of Aeronautics, Washington, D. C.

and cruiser planes, the carrier planes demonstrated every possible form of offense against surface craft and shore bases. Torpedo planes delivered attacks through smoke screens, fighting planes delivered attacks on surface vessels and shore stations, while radio-equipped observation planes kept the ships constantly informed of all phases of the "War." A night air raid sent out by one of the carriers while she was yet 150 miles off the coast of Panama, theoretically destroyed the locks. Two hundred forty-seven planes took part in the maneuvers and flew 4397 hours over nearly half a million miles, in every kind of weather and under varying conditions, without a single serious accident. The authorization of an additional aircraft carrier evidences Congress' realization not only of the need for, but of the expediency of, this type of tonnage in the Fleet. However, there is still an urgent need of additional carrier tonnage to provide for the expansion of the air arm in accordance with its proportionate



FIG. 1 NEW TYPE OF PURSUIT PLANE MANUFACTURED BY THE BOEING COMPANY AND POWERED WITH A PRATT & WHITNEY 420-HP. "WASP" ENGINE

(This type is known as the P-12 and is in use at the present time in certain service squadrons.)

relationship to the Fleet as a whole, and in order to provide the parity of aircraft-carrier tonnage allowed in the Washington Limitation of Armaments Treaty.

The Navy now has on hand or in production aircraft suitable for carrying out any mission which the forces afloat may be called on to accomplish. There are two new types of fighters being produced and a third type being experimented with; deliveries are being made on a large order for a modification of the present type of observation plane, while two new experimental types are under test; a modification of the standard torpedo and bombing plane is in production, and several experimental types are being tested; development of the patrol-type program, along both monoplane and biplane lines, is proceeding satisfactorily; of these, 55 airplanes have been contracted for on the basis of the development of the biplane PN-12, and a small production order placed for monoplane patrol boats similar to the XPY-1.

To provide all types of planes with suitable power plants, aircraft-engine development has been stressed particularly on the score of increased reliability along with further reduction in weight—factors which are mandatory in the design and construction of naval aircraft because of the limitation imposed by shipboard operating conditions. The air-cooled engine ranging in power from 200 to 600 hp. is standard in all types of Navy planes.

The two detachments of the naval air service which provided perhaps the most spectacular demonstration of successful aircraft operation during the year were the Marine Detachment in Nicaragua and the Naval Aerial Survey Detachment in Alaska. In Nicaragua, six tri-motored transport planes have been in daily use transporting personnel, food, ammunition, and other supplies through every sort of weather over rough, wild country. Ground units have been communicated with at least once daily from the air, and aerial escorts have been provided marching units. In Alaska, four amphibian planes continued the work of mapping which was so successfully initiated in 1926. The survey has proved to be of great value in the investigation of the mineral resources of Alaska, the location of forests, the discovery of new sources of water power, and in road and trail building through hitherto unpenetrated territory. The most outstanding single accomplishment of the expedition was the discovery of a 900-acre lake within three miles of the coast and at an elevation of over 2500 ft. The fall from the lake level to sea level is so abrupt that dam construction is unnecessary.

Progress on the Navy's lighter-than-air program is marked by the beginning of actual construction work on the first of the two dirigibles contracted for in 1928, and by the completion of the *ZMC-2* (metalclad) airship. Operations with the *Los Angeles* have included training flights and those on which tests of various apparatus have been made, principally the device for attaching airplanes to the airship, and mooring devices. A number of plane hook-ons have been successfully carried out.

Developments of the past year have conclusively shown the immense value of aircraft carriers, and it is considered that the most important requirement of naval aviation at the present time is to secure as soon as possible the additional carriers necessary to give the Navy the authorized carrier parity of tonnage allowed under the Washington Limitation of Armaments Treaty.

MILITARY AIRCRAFT PROGRESS IN 1929[6]

A more complete and general acceptance of aircraft as a most vital factor to any scheme of national defense coupled with steady increase in their performance and effectiveness, has marked the progress of military aeronautics during the past year. Keeping pace with the development of new and better equipment, those in charge of its employment have enlarged the field of aerial activities until hardly a tactical problem is now considered without giving much serious attention to the action of friendly and enemy air forces. Military aircraft should not be expected to displace



FIG. 2 EXPERIMENTAL THOMAS-MORSE PURSUIT AIRPLANE KNOWN AS THE "VIPER"

(This plane is interesting from two angles: construction—corrugated aluminum alloy; powered with a 650-hp. Curtiss "Chieftain" engine.)

any essential element in our present organization, such as the artillery, infantry, or the corps of engineers. A most important function is to cooperate with these various arms, increasing the efficiency of each as well as bringing a tremendous offensive machine into play against the enemy.

There are six general types of heavier-than-air craft in general

[6] Prepared by S. P. Mills, Matériel Division, Air Corps, U. S. A., Washington, D. C.

use by the United States Army today. They consist of training airplanes, observation, attack, pursuit, light bombardment, and cargo or transport airplanes. While there are many sound arguments against any considerable number of different types of airplanes in the service, yet it appears absolutely necessary to build many types in order to obtain airplanes of greatest effectiveness for the particular work which is required.

During the year the training planes in use were of the same general type as used the previous year. In place of the 180-hp. water-cooled Wright Hispano-type engine, we find a 9-cylinder air-cooled radial engine of the type which Colonel Lindbergh used in his famous transatlantic flight. The change represents an increase of approximately 45 hp., without an increase of overall weight, and was made due to the availability of the new engine and the desire to use the air-cooled type. A complete overhaul is given these engines after approximately 200 hours' training service. The air-cooled engines are now fitted with elaborate air filters attached to the carburetor air intake or air-intake pipes which conduct the air from the top and inside of the fuselage, schemes which eliminate a considerable portion of the usual cylinder wear due to operation upon dusty fields. It is predicted by many that much lighter and lower-powered airplanes will be used in the future, but the average Army pilot apparently favors a liberal amount of power, giving the airplane a maximum speed which is at least 40 m.p.h. above minimum. All Government training airplanes are now of the steel-tube-fuselage type, with a biplane structure of fabric-covered wooden wings. The steel-tubing structure has proved more satisfactory in every way, and appears to offer a surprising amount of protection to the crew in event of a crash. Training airplanes of this type have crashed into the ground from tail spins repeatedly without fatally injuring the occupants. The ability to stand abuse has been given a great deal of attention by engineers, a quality which has inspired the development of more rugged landing gears and wheels. The "Musselman" or "doughnut" type of wheel whose tire is very large in section and is mounted directly upon the hub, uses an air pressure of only 4 to 8 lb. per sq. in. This equipment give promise of cushioning the severe landings experienced while training students.

In the observation type of airplane the tendency to adopt the air-cooled engine is also present, although not so universally recognized yet as in the case of the training airplane. For several years the majority of the observation airplanes have been equipped with 410-hp. water-cooled Liberty engines. The wide use of the Liberty engine in recent years has been due to a surplus of such engines since the war period. Each year has added its improvements to the Liberty engine, until today many pilots still consider it a most dependable aircraft engine. The Government has definitely dropped the Liberty engine as equipment for any new-type airplanes. The use of radial air-cooled engines effects a saving of from 300 to 400 lb. installed weight in an observation airplane, a fact which greatly increases the rate of climb, and improves the take-off and all-around performance at higher altitudes. The maximum speed at the ground is apparently no greater, but increased performance along other lines appears to justify such installations.

There are three observation airplanes in use today: the Douglas series known as the O-2 (the O-2H being used in considerable numbers), the Curtiss series, and that manufactured by the Thomas-Morse Corporation. The Douglas O-2H airplane which is standard in the Air Corps this year is equipped with a water-cooled Liberty engine mounting a nose radiator and providing an exceptionally roomy cockpit for the gunner immediately to the rear of the pilot. It is of interest to note here that the Mexican Government has purchased nine new airplanes of the general O-2H type but equipped with Pratt & Whitney 525-hp. Hornet engines. The Mexicans report excellent performance from these airplanes, particularly under conditions where they are operated from the many very high-altitude landing fields throughout Mexico. The Curtiss series of observation airplanes mount either the water-cooled Liberty or the water-cooled Curtiss D-12 engine. Future plans for this series will use both air-cooled radial engines and liquid-cooled engines of higher horsepower. The Thomas-Morse observation airplanes will use radial air-cooled engines.

Three different types of fuselage construction are represented by three different makes of airplanes: the Douglas using a structure made of chrome-molybdenum steel tubing welded at the joints, the Curtiss, aluminum-alloy tubing riveted or bolted, and the Thomas-Morse, using a corrugated aluminum-alloy shell. Time alone will tell which type of construction will find the greatest favor in the service. Each type apparently offers the



FIG. 3 CURTISS PURSUIT P-1B EQUIPPED WITH CURTISS D-12 ENGINE AND ETHYLENE GLYCOL SYSTEM OF COOLING
(Note the very small radiator necessary to give the required cooling.)

crew of a plane a much greater protection in the event of a crash than the wood-wire construction. As judged from the experiences of the Air Corps during 1929, the tendency in observation practice appears to increase the performance by the addition of more horsepower and the material reduction of weight of the power plants. Another marked tendency is the increasing use of metal. It is not unlikely that in the future we shall see all-metal observation airplanes whose performance is very little behind that of the combat and pursuit class, some of which may use two or more engines and possibly carry a crew of more than two men.

In attack aviation the types generally used have been a sort of converted observation airplane powered by either an air- or water-cooled engine. The year has witnessed an increase in speed of attack airplanes and a great deal of additional experience on the part of attack personnel. Indications point toward the development of a specialized attack airplane whose forward speed at ground level will be considered the most important single feature, with maneuverability, vision, and machine-gun operation important additional considerations. It is possible that certain classes of attack airplanes may be provided with limited quantities of bullet-glancing armor plate placed around certain vital points of the cockpits and engines. Future attack airplanes may be equipped for doing much of their work under cover of darkness. The attack upon troops, and particularly upon supply trains, bridges, roads, and other small objects immediately in the rear of the scene of action will undoubtedly play a most important part in military campaigns.

The Curtiss and Boeing companies have provided the majority of service pursuit airplanes in present use. The past year has witnessed the equipping of certain squadrons with new pursuit airplanes powered with the radial air-cooled engines. As in the observation, the radial engines increase the all-around performance of pursuit airplanes. The speed due to the change of engines alone has not materially increased, but the airplanes themselves represent an advancement in this line. The new radial-engine-powered pursuit airplanes do not appear to have quite the diving properties of the water-cooled, but their rate of climb and performance at altitude are unquestionably better. A most interesting development during the year has been the substitution of ethylene glycol for water as a cooling medium in the radiators of pursuit airplanes. As a result of this scheme, the weight of a pursuit airplane is reduced about 100 lb., the size of the radiator is decreased 70 per cent, and the fuel economy for a given horsepower is considerably improved. Due to the high boiling point of this liquid (387 deg. fahr.), it is possible to operate the engines at an outlet temperature of 300 deg. with a loss of only two per cent in horsepower. Service tests of over 120 hours' satisfactory operation with this cooling system, indicate that the length of life of engines will not be materially decreased. Researches are now being conducted which will determine this more accurately. Should the new cooling system meet favor throughout the service, it is likely that it will be much used in the future by racing airplanes and certain classes of military aircraft.

Paralleling the development of ethylene glycol cooling has been the design and development of a series of cowls capable of entirely enclosing radial air-cooled engines. Speed increases of up to 17 m.p.h. have been recorded on pursuit planes using such equipment. In certain cases the cooling of an engine at full throttle has been somewhat impaired and the visibility of the pilot considerably reduced. While using a Matériel Division cowl on a P-12 pursuit airplane equipped with a supercharged Pratt & Whitney radial air-cooled engine, Captain R. G. Breene of Wright Field averaged 186 m.p.h. over a triangular course in the free-for-all race at Cleveland on September 2, 1929. The work of this development has been carried on by the Matériel Division of the Air Corps and also by the National Advisory Committee for Aeronautics. Although such cowls possess certain disadvantages, nevertheless any equipment which increases the speed of an airplane by 10 per cent must be reckoned with in future developments.



FIG. 4 NEW DOUGLAS ADVANCED TRAINING AIRPLANE
(Similar to the Douglas observation plane except that it is powered with a Pratt & Whitney "Wasp" engine instead of a Liberty engine.)

An effort to increase the speed of pursuit airplanes which is being followed closely is the use of a Curtiss Chieftain, a 12-cylinder double-bank radial air-cooled engine. Such an engine, while delivering in excess of 600 hp., has the same overall diameter as the Wright J-5 Whirlwind engine which delivers 230 hp. The Chieftain engine is still being experimented with in the laboratories of the Curtiss Company and the Matériel Division, and every effort is being made to develop this engine for pursuit and observation work. Possessing as it does its high horsepower and light installed weight characteristics of air-cooled engines, it has the further advantage of streamlining satisfactorily into a fuselage

and obstructing the vision of the pilot to a less extent than the single-bank radial of equal horsepower.

American pursuit airplanes are generally of the biplane type and constructed usually of welded steel tubing or of square duralumin tubing riveted and bolted together. As a result of observing the winners of the races of recent years, it would seem that the low-wing, all-metal monoplane will some time find its place as a combat and pursuit machine. It is likely that the retractable landing gear and increased horsepower with greatly reduced radiators will be the tendency in future pursuit development.

The development in bombardment aviation has been of a power plants. The year 1929 has witnessed the real beginning of the use of gearing in Air Corps bombing and observation airplanes. The twin-engine type of bomber appears to be still held in highest favor by the service as it gives a better distribution of defensive armament. The development of the Fokker F-32, a large transport monoplane powered with four 525-hp. engines, is being watched with much interest by various countries. This plane, with a reported maximum speed of 155 m.p.h., is likely to furnish some valuable ideas to the designers of bombardment airplanes. The construction of the modern bombing airplane follows much the same type as other classes: namely, the welded-steel-tube fuselage as used by Keystone Company and the bolted



FIG. 5 THOMAS-MORSE O-19 OBSERVATION AIRPLANE POWERED WITH A PRATT & WHITNEY "WASP" ENGINE
(Note the corrugated aluminum-alloy fuselage.)

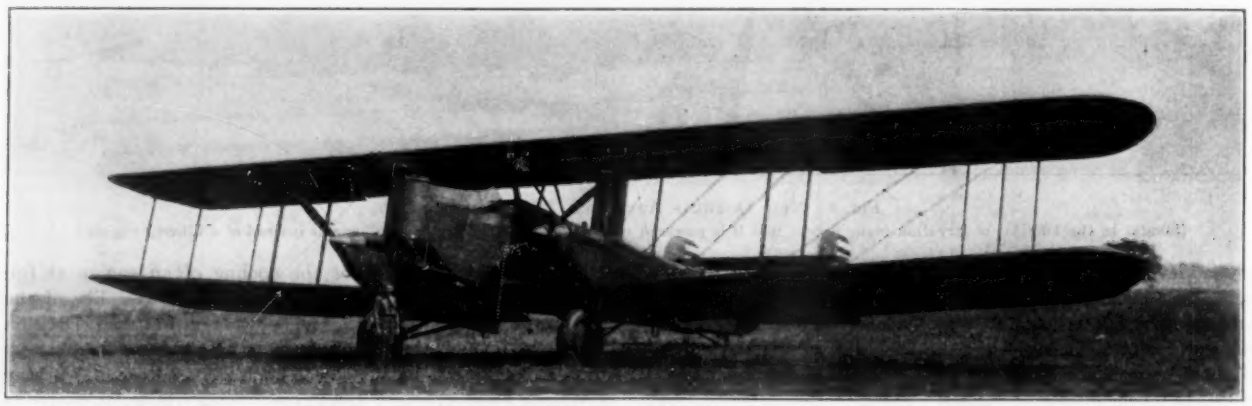


FIG. 6 CURTISS "CONDOR" BOMBER, POWERED WITH TWO 620-HP. CURTISS GEARED "CONQUEROR" ENGINES
(Note that a gunner is placed in the nacelle behind each engine.)

sound, consistent nature, resulting in a steady though not revolutionary increase in performance. The Liberty engines have gradually been displaced by Pratt & Whitney Hornet and Wright Cyclone engines of 525 hp. each, with a resulting conservative increare in speed and a considerable improvement in rate-of-climb and performance characteristics. The Keystone bombers thus equipped have shown speeds of over 120 m.p.h. coupled with satisfactory ability to handle big loads. The year has witnessed the supplying to the service of several Curtiss Condor bombing airplanes each equipped with two 625-hp. geared Curtiss Conqueror engines. The use of geared engines appears to give a material increase in propeller efficiency, and it is probable that the geared engine will be used to a greater and greater extent in future machines of this kind. The United States is probably a little behind certain other nations in the employment of geared and riveted duralumin fuselage as used by Curtiss Company. Again the tendency is toward a greater and greater percentage of metal construction and the addition of more horsepower.

Cargo and transport airplanes for the transportation of ammunition, supplies of all kinds, and even of troops, will undoubtedly be in great demand during the next emergency. In general it is expected their designs will follow that of a modified bomber with ability to carry tremendous loads from restricted fields, which means the probable employment of geared engines and controllable-pitch propellers. In all probability the speeds of bombers will be very much increased, using speed as a defensive measure rather than carrying so much weight in the defensive armament.

The year 1929 will be remembered in history of military aeronautics as the year during which the wartime Liberty engine was

finally discarded, being replaced largely by the new radial air-cooled engines. It will also be remembered for the attempted "come back" on the part of water-cooled engines through the employment of new cooling systems and the installation of several models of geared engines. Other developments too numerous to mention, such as the successful use of propellers which can be adjusted during flight, new high-altitude equipment, photographic equipment, parachutes with a more satisfactory operation, have all been pushed with much interest and painstaking care. Increasing use of metal construction and increased power have each contributed to increasing the performance of our military aircraft. It is imperative that this work of development be continued and that we learn all lessons possible from races, experiences of other countries, and commercial activities, in order that this important branch of the national defense shall be maintained at its highest point of efficiency.

Airships[7]

The circumnavigation of the world by the airship *Graf Zeppelin* in twenty-one days total time, or twelve days of actual flying time, was the outstanding aeronautical event of the year. In addition to the around-the-world flight, the *Graf Zeppelin* twice crossed the Atlantic between Lakehurst and Friedrichshafen. Notable flights were also made by this airship over Europe and Asia Minor. In May, the *Graf Zeppelin* made an unsuccessful attempt to cross the Atlantic, but because of engine failure was obliged to put into the French airport Cuers, near Toulon. This event caused some shaking of heads among the airship doubters, but an actual study of the course of events on this occasion showed the extraordinary ability of the airship to make port safely in spite of a must unusual accident. Four out of the five engines suffered broken crankshafts due to torsional vibrations induced by stiffening the couplings between the engines and propellers. With only one engine remaining in operation the *Graf Zeppelin* was able to make port in spite of strong adverse winds. The Zeppelin and Maybach companies, builders of the hull and engines, respectively, of the *Graf Zeppelin*, profited by this accident to study the subject of torsional vibration. The faulty couplings in the *Graf Zeppelin* were replaced by others which eliminated the disastrous periodicity in the power plant.

The two large British airships, *R-100* and *R-101*, of over 5,000,000 cu. ft. gas volume, have been completed and inflated with hydrogen. At the present writing they are practically ready for their trial flights. These airships are of a novel type, differing considerably from the Zeppelin pattern to which practically all previous rigid airships have closely conformed. The usual development troubles in a new type may be expected, but there is no reason for the prognostications of failure of these airships which have been broadcast by their opponents.

The small experimental metalclad airship *ZMC-2*, designed and constructed for the Navy Department by the Aircraft Development Corporation, of Detroit, Michigan, has successfully passed through its trials and been accepted by the Navy. Considering the extreme novelty of this airship, its trials were extraordinarily successful and free from troubles. On the other hand, it should be remembered that this ship has not yet been subject to the crucial tests of flying in rough weather, to continued exposure to the atmosphere, or to severe alternating stresses. A true estimate of the value of the metalclad construction for airships cannot be made until after the test of continued service.

The U. S. S. *Los Angeles* has been throughout the year in almost constant service, although no particularly long or spectacular flights have been made. Important work in the development of handling airships has been carried on with the *Los Angeles*. These developments include extended use of the stub mooring mast, and successful trials with a mobile mooring mast which can be towed in and out of the hangar with the bow of the airship attached to it.

A trapeze for carrying, releasing, and hooking on airplanes has been installed and successfully tested on the *Los Angeles*. Similar experiments have been carried out before with very light airplanes, but the installation on the *Los Angeles* is the first of its kind suitable for continued use, and adaptable to planes of real military value.

The Goodyear-Zeppelin Corporation at Akron has made good progress in the detailed design of the new 6,500,000-cu. ft. airships *ZRS-4* and *ZRS-5* for the U. S. Navy. Many of the girders for the first airship have been constructed, and assembly is now under way in the corporation's new huge airship factory and dock. A great deal of experiment and development work in connection with the details of these airships has been carried on during the year.

Airports[8]

A careful inspection of many of the nation's largest airports will probably disclose the fact that many of them are "white elephants." The financing of airports has frequently been of a sensational nature, and all too frequently the planning has been

TABLE 4 AIRPORTS AND LANDING FIELDS IN THE UNITED STATES ON JUNE 30, 1929

	Municipal	Commercial	Private	Auxiliary	D. C. lighted inter.	Lighted	Army and Air Corps	Navy	Total	Sq. miles per airport
Alabama	2	4	..	...	4	...	2	..	12	4330
Arizona	23	4	..	9	...	2	1	..	37	3090
Arkansas	3	5	..	6	...	2	1	..	15	3550
California	53	42	..	23	16	19	6	3	143	1110
Colorado	9	6	..	3	6	5	1	..	25	4160
Connecticut	4	1	..	4	1	2	..	..	10	497
Delaware	...	1	1	4	...	1	..	..	6	325
Dist. of Columbia	...	3	..	...	...	3	1	1	5	14
Florida	15	12	..	8	...	2	1	3	39	1505
Georgia	11	2	..	1	5	2	2	..	21	2810
Idaho	3	...	..	2	2	1	..	..	7	11984
Illinois	8	35	..	8	14	11	2	1	68	835
Indiana	8	15	..	3	10	2	1	..	37	981
Iowa	9	15	..	1	15	6	1	..	41	1370
Kansas	15	9	..	1	6	6	2	..	33	2490
Kentucky	3	4	..	...	2	1	..	..	9	4511
Louisiana	6	5	..	2	...	3	1	..	14	3240
Maine	2	2	..	1	...	1	..	..	5	6000
Maryland	1	4	..	7	...	2	3	..	15	663
Massachusetts	3	12	1	3	2	2	1	..	22	375
Michigan	19	9	..	7	1	5	2	..	38	1525
Minnesota	5	7	..	4	3	2	..	..	19	4460
Mississippi	1	4	..	2	...	...	..	..	7	6695
Missouri	9	6	..	2	2	6	..	..	19	3620
Montana	7	5	..	6	1	1	..	..	19	7700
Nebraska	5	6	..	3	13	3	1	..	28	2750
Nevada	9	1	..	5	3	2	..	..	18	6310
New Hampshire	5	...	..	1	...	...	..	..	6	1557
New Jersey	7	15	..	4	2	5	2	1	31	265
New Mexico	10	2	..	8	1	...	1	..	22	5570
New York	13	21	..	9	1	8	4	..	48	1025
N. Carolina	9	7	..	2	6	2	1	..	25	2100
N. Dakota	5	2	..	1	...	...	..	..	8	8855
Ohio	13	27	..	10	20	12	3	..	73	562
Oklahoma	10	14	..	16	8	2	1	..	49	1430
Oregon	11	4	..	6	3	2	..	..	24	4025
Pennsylvania	15	33	..	15	24	6	1	1	89	507
Rhode Island	...	4	..	4	...	...	..	..	8	156
S. Carolina	3	4	..	3	4	1	1	..	15	2060
S. Dakota	6	7	..	3	...	...	..	..	16	4850
Tennessee	5	3	..	3	...	1	..	..	11	3820
Texas	30	14	1	34	2	8	15	..	96	2660
Utah	3	...	..	4	2	1	..	..	9	9440
Vermont	2	2	..	5	...	...	..	..	9	1075
Virginia	3	10	..	3	8	3	4	2	30	6337
Washington	10	7	..	8	...	3	2	1	28	2460
W. Virginia	2	3	..	5	...	...	..	..	10	2413
Wisconsin	13	19	..	1	8	5	..	..	41	1366
Wyoming	6	2	..	2	16	2	..	..	26	3760
Totals	414	419	3	262	211	153	64	13	1386	2180

NOTE: The number of airports in column marked "Lighted" is included in the other columns and is not additional.

[7] Prepared by C. P. Burgess, Bureau of Aeronautics, Navy Department, Washington, D. C.

[8] Prepared by John Bonforte, Transcontinental Air Transport, Inc. Washington, D. C. Jun. A.S.M.E.

more of the kind to catch the eye of the investor rather than that of the flier.

However, much real constructive work has been done. Competent architects and better engineers are being employed for airport work. Plans are generally made with an eye to the future as well as to the immediate present. Airports have been laid out in many parts of the country. A list of airports and landing fields in the United States on June 30, 1929,[9] is given in Table 4.

It should be clearly understood that few of these airports and flying fields are beyond the preliminary stages of construction. Airports are like cities: they cannot be built in a day, a year, or even two or three years. It is probably true to say that airports, like many other progressive works of construction, are forever being built but are never completed.

Various trends noted in airport construction during the past year are indicated below.

Proximity and Accessibility of Airports to City. A strong present-day tendency is the location of airports closer and closer to the heart of the city. Close-in airport locations have probably been overemphasized, and the economic pressure of overhead costs will probably soon force airports out in the open country again.

The means of communication, the convenience of communication, and the time of travel from the business sections of cities to their airports are now given full consideration in the selection of an airport, and a general improvement in these features is readily noticed.

Building Construction. One extremely interesting construction this year has been the erection of the "hex-hangar" as a repair shop for the Western Air Express Company at Los Angeles. This innovation will undoubtedly be closely watched, and if the results prove successful, the widespread use of such a hangar as a repair depot is almost sure to follow.

Hangar doors have received special study. Practically all those used today are of the sliding, all-steel-construction type. Some hangar doors are of tubular construction, but the majority are of angle construction. The construction of hangar doors is also divided between the straight-rolling type and the round-the-corner type. The former is apparently in the majority. A recent hangar construction included the installation of motor-operated cantilever doors. The installation of this type of door is still too new for an opinion to be given regarding its general adoption by the aircraft industry.

Many airports now have comfortable waiting rooms and restaurants for passengers comparable to the best found in railroad stations. In addition, clubhouses, swimming pools, and other outdoor sports features are being installed at a few airports for club members.

Weather, Radio, and Communication Service. Reliable weather data are a feature of all the major airports. This has been made possible through the cooperation of the Department of Commerce. The use of radio and of the automatic telephone-typewriter system have also aided greatly in expediting the dissemination of weather data, as well as other pertinent information.

The telephone-typewriter has been found especially useful in the relaying of telegraph messages between the airport and the central telegraph office in the business section of the city.

Service Facilities. Improved service and repair facilities are noticeable at all airports. One of the major improvements is the underground electric fueling pits. The pumps used in conjunction with these electric fueling devices generally have a fueling capacity of 20 to 30 gal. per min.

Field service has also been greatly aided by the improved hangars and repair shops that have been built, where oil and water may be heated on cold days and general repairs on planes and engines conveniently made.

Drainage and Runways. These two parts of airport construction go together, for neither should be planned without fully considering the other.

The importance of adequate drainage at airports can hardly be overestimated, for it is truly the foundation of a good airport. Natural drainage is depended upon at some airports, but it is seldom adequate for all weather conditions. The increased use of hard-surfaced runways is making at least partial artificial drainage imperative. The tendency is, however, to tile the entire airport with pipe of such size that it will continue to function even after many years of silt filtration have partially clogged the pipes.

The trend in runway building is difficult to set down precisely. It varies with the climate, the materials available, the money available, and the uses of the airport. Concrete will probably never come into general use as a runway surface so long as its cost remains where it is at present, namely, between $2.50 and $3.00 per sq. yd. of 8-in. thickness.

The trend seems to be to surface the runway as cheaply as possible with oil, asphalt, or other bituminous material, and then to improve it from year to year as finances and requirements dictate.

Cinders, broken stone, and other loose materials have also been used as runway surfaces. While all these have many advantages, such as low cost, excellent drainage surface, etc., they also have such fundamental disadvantages as being very dusty in dry weather, and flying up and knicking the propellers of airplanes taxying across the field. These objections will undoubtedly limit the use of these loosely bound materials to the inferior airports.

Airport Lighting. Airport lighting has probably advanced least in point of efficiency of all the airport construction problems. The reasons for this slow advancement are:

1 Airport managers have only a meager knowledge of airport lighting

2 Few of the lighting companies understand the airport lighting problem. Consequently, they simply modify existing lighting equipment intended for railroad, marine, or airway use and market it for airport use.

In the future, adequate means of lighting runways distinctly will have to be developed. One or two firms are already marketing lighting products for such a purpose. Their efficacy and value are still undetermined.

Marking of Airports. The Department of Commerce is leading the way in the better marking of airports by installing and painting metal cones around each boundary light at intermediate fields, and placing a 6-ft.-diameter ground marker of white crushed stone at the base of each boundary light.

Another necessary marking at airports is the increased use of wind cones. In these days of mile-square airports, a single dirty wind cone flapping indistinguishably a mile away from the point of landing is of little value.

Landscaping of Airports. Airport landscaping is at present only the sketchy dreaming of architects. One finds it in drawings but not in actuality. However, as planning must precede achievement, it is certain that better- and better-looking airports will be built.

[9] Reprinted from *Aviation*, Oct. 5, 1929.

The Theory of the Autogiro

By JUAN DE LA CIERVA,[1] SPAIN

The author, realizing the fundamental limitations of the ordinary airplane, has worked for years to produce a new system of flying. Passing the helicopter and ornithopter stage, a machine was devised with articulating blades that could flap freely, the dissymmetry of the reactions on opposite blades being thereby overcome.

MOST persons know what the autogiro is, how it looks, and how it works, so that I am not going to enter into a detailed description of what the autogiro is. Instead I will try to convey briefly the philosophy of the autogiro, as an introduction to its theory.

I have been working for many years on the ordinary airplane. I am a great enthusiast of the airplane, and I think it is the most marvelous invention of modern times; but a long time ago I came to the conclusion that the plane had fundamental limitations due to its own nature, and I tried to produce a new system of flying free from those limitations. I thought that the airplane could be improved quantitatively in every direction, but my thought was and still is that no improvement can change the very nature of the airplane and its qualitative limitations.

After a certain amount of theoretical work I reached the conclusion that the only method for removing those fundamental limitations was to make the wings relatively independent of the frame or body of the machine, so as to make it possible to keep them going fast through the air while the machine could be going slowly or even be stationary. I saw the possibility of achieving this result without the introduction of mechanical complications such as those which inevitably will happen in the helicopter or ornithopter, or, generally speaking, in flying machines with driven wings, and I started then the study of a very primitive autogiro which only was related to the existing machine by the fundamental idea of freely rotating wings.

I am not going to explain in detail the long and painful experimental period through which the autogiro has passed to the present moment. I only point out to you that I did not achieve success at all until I had the idea of articulating the blades of the autogiro so as to let them flap freely, since there were too many mechanical and aerodynamical problems that could be solved only in that way.

The main problem I solved in this way was the dissymmetry of the reactions on opposite blades, since there is a single group of them rotating in the same direction. On one side of the machine the speed for a given point of the blade will be the addition of the general speed of advance of the machine and the peripheral speed, while on the opposite side there will be the difference.

To balance the lift in those two opposite planes by means of controls that would periodically change the incidence of the blade is a most difficult problem, and this solution of the articulation does it automatically and in a very simple way.

The aerodynamics of the autogiro is a very complicated problem, if one considers how difficult are the problems involved in the calculation of ordinary propellers and that the autogiro can be likened to a propeller with three or four more parameters. It is a propeller which is advancing through the air with a great obliquity and with blades which are free to flap. The speed and the incidence at every point of every blade at every moment are different. I cannot try to make a detailed analysis of the aerodynamical qualities of the autogiro and compare them with those of the ordinary airplane. Nevertheless, and in answer to an extended criticism to the autogiro, I want to point out that there is no fundamental reason why the autogiro should be aerodynamically inferior to the airplane. It has been said that, since the speed of the blades of the autogiro through the air is higher than the corresponding wings of the equivalent airplane, the energy expended will be greater. The autogiro because of its special qualities in landing is free from the necessity of having big wing surfaces, and in consequence it can utilize very high wing loadings, which means higher aerodynamical efficiency.



FIG. 1

The autogiro by its aerodynamical nature is not subject to any of the risk of stalling. The wings always move fast through the air, whatever the speed of displacement of the fuselage. It can be stopped in the air for a moment without losing completely its lift. If by a sudden maneuver, such as the pulling back of the stick, the machine is brought to a standstill in the air, the wings still continue for a moment to rotate under the action of their momentum, and they still give a certain amount of lift. This feature makes this machine fundamentally different from the airplane, which, when it is stalled, loses all grip on the air. As the movement of the blades continues even when the machine is not moving with any forward speed, the autogiro can become a parachute at the will of its pilot; and a most interesting aerodynamic problem can be pointed out in this connection. I had predicted as far back as 1921, by theoretical considerations, and since then I have confirmed them by experiments which have been checked by the officials of several governments, that the efficiency of the autogiro as a parachute is considerably higher than that of the ordinary parachute. As a fact it does not need discussion, and I should like to point out that the efficiency of this machine as a parachute is three times as high as that of an ordinary parachute formed by a disk of the same diameter as its rotor.

[1] United States representative: Pitcairn-Cierva Autogiro Company of America, Philadelphia, Pa.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The physical explanation of that phenomena is that, contrary to what most aerodynamists think, the mass of air influenced every second is much larger than that corresponding to the volume equal to the product of the surface of the rotor by the speed of descent. The ordinary parachute practically has an efficiency corresponding to the optimum in this assumption, and if there is considered the considerable amount of energy wasted in the complex vortices in its wake, it is obvious that a much greater efficiency can be achieved by a parachute like the autogiro in which the flow is much more smooth and regular. My theoretical results are that the optimum possible parachute would have an efficiency equal to six times that of the disk, and the present autogiro achieves about 50 per cent of that optimum.

Structurally the autogiro is based on a very different principle from that of the airplane. The wings of the airplane are constructed in such a way as to resist the stresses with rigidity, while the autogiro is a flexible structure. Sudden changes in incidence or speed do not increase the structural stresses in the same measure as they increase those of an airplane. The centrifugal force which keeps the blades approximately horizontal acts as a spring of long travel and gives the autogiro the quality of a much greater comfort. It has a smoothness in flight which is greatly different from the feeling that is produced in the flight of an airplane in bumpy weather. Many people ask whether the autogiro can fly in bumpy weather, and I can say that the autogiro is much less sensitive to bumps than is an ordinary airplane and that this is due to the flexible nature of its structure.

AER-52-3

The Problem of Catapulting Aircraft From Commercial Vessels

By LIEUT. WM. M. FELLERS, U. S. N.,[1] WASHINGTON, D. C.

After describing the catapult mechanism and the launching car, the author discusses the development of catapult launching by the U. S. Navy. Following this he outlines the several steps in launching a plane, stating the precautions which should be taken. In conclusion he discusses the possibilities of a ship-to-shore plane service, using catapults. It is his belief that the same mechanisms employed successfully in the Navy can be transferred almost intact to any commercial vessel, and that planes can be launched from shipboard with a degree of regularity and safety compatible with commercial operation.

THE unsolved part of the problem of catapulting aircraft from commercial vessels is largely an economic one. The technical part has been solved, as is demonstrated by more than seven years of service on the battleships and cruisers of the United States Navy. During this time experimentation and development have been continuous. Thousands of launchings have been made, and to date there has not been a single fatality or even serious injury to personnel as a result of catapulting.

It seems certain that the same mechanisms employed successfully in the Navy can be transferred intact to almost any commercial vessel, and that planes can be launched with a degree of regularity and safety compatible with commercial operation.

The Catapult

The catapult is a device to bring the airplane from a state of rest to one of minimum flight speed within the restricted space available aboard ships of conventional design. Catapults have been built through large ranges of length, and with various types of prime movers. In all cases it has been necessary to devise a prime mover capable of delivering a relatively large force for a comparatively short time, and the consideration of weight has been of major importance. Compressed-air and powder engines are both used in the service. A flywheel is used on one type to store up energy and deliver it during the launching interval. The author has been informed that a hydraulic catapult has been tried but abandoned. Probably it, as well as steam types, could be developed, although it is doubtful whether such catapults would be as serviceable as the ones now used.

To engineers accustomed to thinking in terms of accelerations and decelerations of trains, automobiles, and other vehicles used for passenger carrying, it will be interesting to know that accelerations of 90 ft. per sec. per sec. are regularly used during the launching period of airplanes. This is nearly three times the acceleration due to gravity. The car on which the airplane is held before and during the launching interval is brought to rest at the end of its launching stroke and returned to the starting point to launch the second plane. To conserve length of structure and to devote as much of it as possible to bringing the airplane up to speed, the stopping of the launching car is accomplished in as short a distance as practicable, and the deceleration is of the order of 100 times gravity.

[1] Bureau of Aeronautics, Navy Department.

Presented at the Great Lakes Aeronautic Meeting, Cleveland, Ohio, August 31, 1929, of the Aeronautic Division of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The Launching Car

The launching car is, to many, the most interesting part of the whole mechanism. It must hold the airplane in the proper attitude before and during launching, and must be strong enough to resist the forces acting on it during that time. In addition, it must resist a force of 100 times its weight during the deceleration or stopping process. If the car weighs 1000 lb., it must resist the action of 100,000 lb. during stopping. One engineer aptly said, "It lives in a little universe of 100 g, and lives!"

If a launching car were taken from its catapult and placed in a vertical position, and if a hundred other launching cars were placed on top of it in such a way that the upended one supported all that weight, the force acting on the bottom car would be the same as that acting during the deceleration period immediately following a full-speed launching.

Of course, the pilot and plane are not subjected to this deceleration. It is much beyond anything that could be endured by either. Instruments in planes have registered $12\,g$ during certain maneuvers. The pilot lived, but experienced a dislocation of his organs that put him in the hospital. A deceleration of $3\,g$ seems to be within the capacity of nearly all normal people to stand without discomfort. However, commercial passenger-despatching catapults will probably be made with slightly less acceleration, with either longer runs or slower end speeds.

The sensation of being launched is similar in degree to that experienced when an inapt automobile driver suddenly engages the clutch of a car with the gear in first and the engine racing. The only difference is that the catapult keeps on pushing, and the automobile engine either stalls completely or slows down in short order. The duration of a launching is from one to one and one-half seconds, depending on the run and end speed. This time is so short that it is not until after several launchings have been experienced that an individual learns to separate events into start, run, and release—it's just one grand push. Let the word "push" be emphasized; it is a push, not a blow.

Catapult Development in the Navy

One reason for the lack of casualties in connection with catapult development in the Navy has been careful design in the first place, followed by thorough proofing of each unit. Each report of malfunctioning has been studied, and whenever possible, even when the cause has obviously been due to lack of care in operating, simpler or better mechanism has been developed to replace the parts which gave trouble. It has been the ideal to furnish fool-proof mechanism. This ideal has not been reached, but progress has been made.

On the first catapults there were so many valves to be operated, each in its proper sequence, that operators needed long and careful training to make them proficient, and even with the best training many individuals were found who could not be depended upon to operate them properly. The control mechanism has now been simplified to the point where it is only necessary to pull a string or move a single lever to place the catapult in motion. As a result, reports of failures to complete an attempted launching satisfactorily have practically disappeared.

It is hoped that commercial development, when it comes, will follow along conservative lines and maintain the good record made so far.

Precautions Taken in Launching a Plane

Before the pilot climbs into his plane to be catapulted, the plane is inspected carefully and serviced. Particular care is taken to see that nothing is left adrift to fly back in his face or hit him in the solar plexus. Any freight to be carried must also be secured against motion aft, or it would all wind up in the extreme aft position, or possibly carry away completely and left behind.

The plane itself is then secured on its launching car so that it will not capsize due to a side wind, rotate forward due to the force acting through the propeller during the engine warming-up process, slide forward on the car due to the same force, or slide off backward during the accelerated run, or rear up in front. Whatever the means employed, it is imperative that the plane be released so that at the end of the accelerated run it is free to leave the car and will be in an attitude to start normal flight.

The loaded launching car is brought back to "battery" position and locked there. The pilot takes his place and the engine is started. As the engine becomes warm it is speeded up until finally it is running all out. This is done to test the engine and make sure that it is developing full power. The lock serves to hold the plane and car in the starting position during this test and also allows all slack to be taken out of the entire catapult system. Any slack would result in a jerky start, and sufficient slack would result in waves of force reaching the plane that might wreck it.

The plane is now slowed down to idling speed, and the signal to launch is awaited. In the fleet this signal comes from the flagship, and is made by first hoisting a series of flags, then pulling them down. The hoisting is the preparatory signal, the hauling down is the execution signal. On commercial vessels this signal would come from the bridge, and would be made when the ship had arrived at the desired point and had been maneuvered into a position that would enable the plane to be launched into the wind.

When the pilot receives the "execute" signal he waves his arm in a circular motion. The catapult crew make the last few preparations, such as putting in the powder or adjusting the air to the exact pressure required, and then the officer in charge of the launching replies to the pilot's signal with a similar circular motion of the arm.

The pilot then opens up his engine full out, and after satisfying himself that it is acting normally, settles himself in his seat, puts his feet on the rudder and one hand on the stick. He then holds out his arm. When he withdraws it, the officer conducting the launching counts, "One, two, three," waits until the middle of the next up-roll, and the plane is launched.

The actual signal to fire is not made by the aviator, but by an officer on the deck. This is done because the officer on deck can choose his position and can more easily see that nothing is in the line of flight. He can judge the phase of the ship's roll better, and thereby avoid launching the plane while the catapult is pointing down at the water. He can also hear the engine of the plane and can suspend the launching if it begins to slow down or if for other reasons it becomes desirable to hold up the launching.

If the engine should stop or slow down just as the plane started, no great harm would be done. The end speed attained would be only one or two miles per hour less than if the engine were running normally. The plane would have sufficient speed to make a safe landing without stalling or spinning, but would be in the sea, close to the ship, with her mission unfulfilled. Consequently every precaution is taken to see that the engine of the plane is running properly before the start, to insure its proper functioning when it is needed, which is immediately after launching is over.

The plane stays on the car throughout the accelerated run. It does not fly off at some indeterminate point when sufficient speed has been attained, but leaves the car as the car engages the brake at the forward end of the catapult track.

After the car has stopped it is brought back to loading position, and the next plane is put on it and prepared for launching.

During the actual launching, the pilot is only so much baggage. He may move his controls, if he so desires, but they have no effect on the plane, since it is held securely to the launching car. From the time he gives the signal to fire until his plane is flying off the end of the catapult, he is by no manner of means a free agent. At the end of the run he takes charge of the plane and can immediately control it, but not before.

Proposed Ship-to-Shore Service Using Catapults

In a paper delivered before the Metropolitan Section of the Society of Automotive Engineers, it was pointed out that the first step in the development of the combined use of aircraft and commercial ships should be a ship-to-shore service using a catapult. This conclusion was reached after giving consideration to the ships, aircraft, and machines now in existence, without discounting the future in any degree. It would be well to operate one or more planes from a single ship until all the special conditions surrounding such a service were experienced. Apparently desirable extensions of the service could then be given careful consideration in the light of experience gained. There is little doubt that new material could be evolved to meet any real need, provided profitable operation could be foreseen.

With material now available it is possible to shorten the time in transit of valuable mail and express both east and westbound across the North Atlantic. Planes could take off the morning before sighting land and in a few hours land in London or Paris on the eastbound trips, or in Boston, New York, or Norfolk on the westbound trips.

If planes took off 200 miles at sea, mail could be delivered or forwarded within three hours, while the ship would still be more than 100 miles out. This would result in a saving of from five to seven hours in New York. More time could be saved to Paris and London where stops and transfers are now necessary.

It is possible to build catapults of any capacity whatever. With larger catapults, planes with greater safe radii of action could be used and the time saved could be increased accordingly. With sufficiently long-range planes, passengers could be landed directly in inland cities or ones remote from the port used by the ship, and they could, in many cases, be landed at their remote destination while their fellow-passengers were descending the gangplank at the port, and sometimes sooner.

Places along the east and west coasts of the Americas that are now inaccessible because of being off main trade routes or because of having inadequate harbors, could be reached quickly by a plane flying from a ship whose regular route took her within a few hundred miles. Such a service might, in some instances, save weeks of delay.

Before any extensive plans are made, prolonged tests should be made in the North Atlantic. These tests should cover a period long enough to experience all the phases of weather, which will be worse there than on most trade routes. The equipment, including the catapult and planes, should be the best available. The pilots and operators should be experienced in operating from shipboard, and each problem encountered should be completely solved before attempting a more ambitious program. No attempt to operate under extreme conditions should be tolerated. Everything should be done to insure safety. Increase of range and load should be made by small increments. As in all transportation problems, the motto should be "Safety First."

To sum up, the problem of launching planes from commercial ships is one of procuring enough high-tariff traffic to offset the cost of operation, which will be high, even after every unnecessary expense has been eliminated. The physical problem of despatching an airplane from the deck of a ship has been solved.

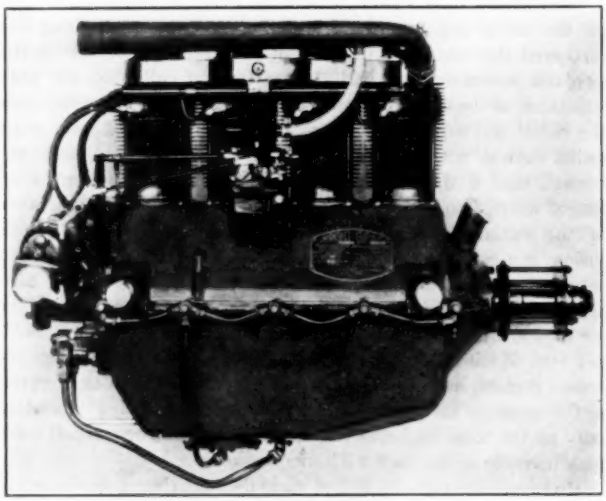


FIG. 1 WRIGHT GYPSY

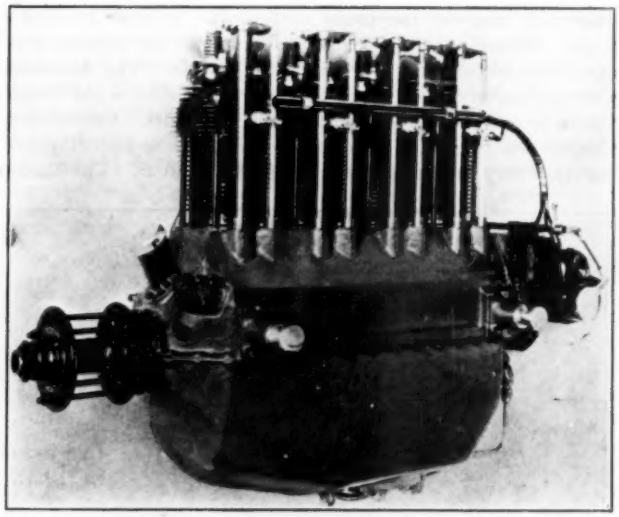


FIG. 2 WRIGHT GYPSY

Recent Developments in Aircraft Engines

BY JOHN H. GEISSE,[1] MADISON, WIS.

Improvements in engine design for airplanes during the year have been gradual. Production facilities now exceed the demand and developments should come more rapidly. There has been a trend during the year toward the "in-line" or air-cooled types of engines. However, the radial type still possesses a number of advantages. Manifolding in radial engines is quite a problem due to the impossibility of getting symmetry except in nine-cylinder types where three three-way manifolds may be used.

Cylinder construction with a steel sleeve having machined cooling fins and a cast-aluminum head is accepted quite generally, and there is a trend toward the use of heat-treated alloys for the crankcase. Many interesting developments in the Diesel types of airplane engines can be expected.

ABOUT one year ago the author prepared an article for *Aviation* on the development of aviation engines during the preceding 25 years. In that article he mentioned that the development was noteworthy in that there had been no radical or startling improvements made. The same statement holds true today. There has been no outstanding development in the last year or in the last 26 years. The most noteworthy event of the last year or two has been the number of new concerns that have entered or have become closely affiliated with this field and the tremendous expansion of production facilities on the part of the older companies. Outstanding among the new ones may be mentioned such well-known companies as the Continental Motors Corporation, Lycoming, E. W. Bliss Company, Le Blond, and the author's associates, the Gisholt Machine Company.

Before considering the products of these companies, it will be interesting to look into the possible market for their wares. A little over a year ago there was a definite shortage of engines. The stock of war-surplus engines was rapidly dwindling, and with the increased demand the production facilities of the country were not equal to meeting the situation. This curtailed the production of planes, and the market was quite definitely a sellers' market. Under this condition it was quite easy, and perhaps quite natural, for the plane manufacturers to make rather broad statements of the number of planes they could sell if only they could secure the engines. The existing engine manufacturers, and many others who were not then manufacturing, were influenced by these optimistic statements and immediately took steps to provide against another shortage this year. How admirably they succeeded is evidenced by the fact that present production facilities far exceed any possible demand. Whereas the year started with plane sales predicted to exceed 14,000, the latest estimates place the probable year's sales at less than one-half of this figure. The actual engine production for the first six months of this year as reported by the Aeronautical Chamber of Commerce was 3275, and it is doubted whether this figure will be greatly exceeded in the present period. If it is doubled in this period, it will mean a production rate of not over forty engines per day. In contrast to this it is very easy to total up claimed production facilities of over one hundred per day, and each day brings news of new enterprises, all planning on going into production, at the rate of ten per day.

It is not desired to paint a gloomy picture for the industry, but the author does want to impress the fact that there is no shortage in production facilities for the time being and that there is not likely to be another shortage in the immediate future. He confidently expects the industry to have a phenomenal growth and that the demand will catch up to production facilities, if the growth of the latter only will slow down to a saner rate.

This rush to get into production has, to a considerable extent, held back engine development during the last year. Now that production facilities exceed the demand, developments will come along more rapidly. Not only will the engineers have more time to concentrate on improvements in their products, but competition will demand them.

Reverting to the engine developments of the last year, one is

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[1] Vice-President of Engineering, Comet Engine Corporation.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

impressed most by the trend toward the "in-line" air-cooled types. Whereas there were practically none in use here one year ago, there are now quite a few on the market. The American Cirrus Company has secured the American rights to the 95-hp. Cirrus four-cylinder-in-line engine, and the Wright Aeronautical Corporation has taken a license to build the 85-hp. "Gypsy" engine of very similiar construction (Figs. 1 and 2). The Michi-

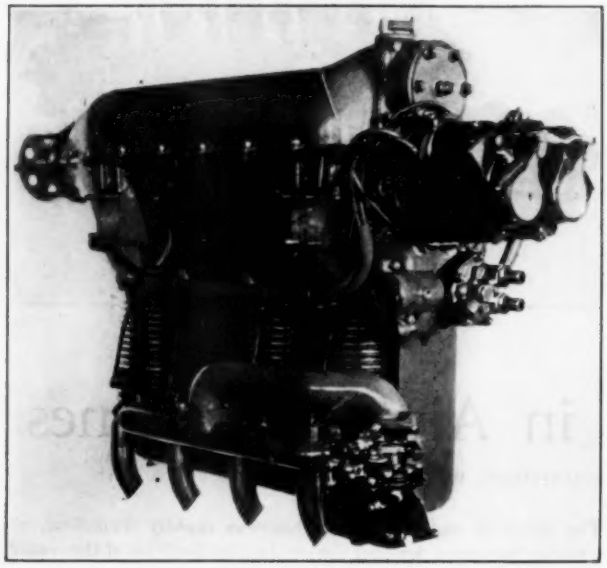


FIG. 3 MICHIGAN AERO ENGINE COMPANY ROVER

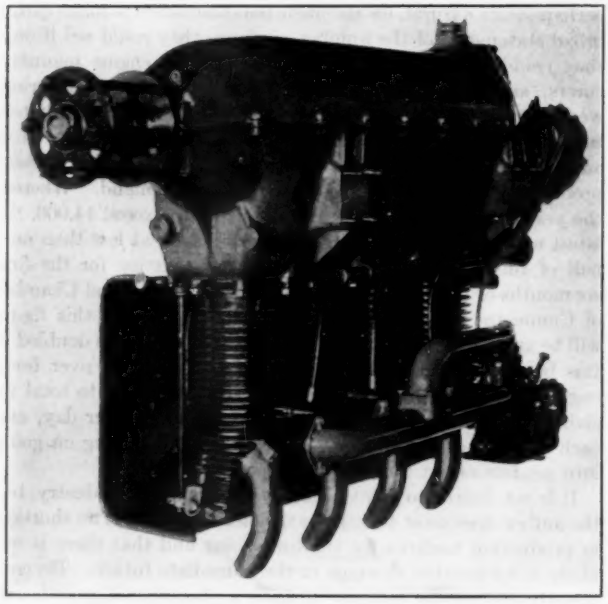


FIG. 4 MICHIGAN AERO ENGINE COMPANY ROVER

gan Aero Engine Company has brought out a 55-hp. inverted four-in-line known as the Rover (Figs. 3 and 4). The Dayton Airplane Engine Company manufactures a larger four-in-line developing 110 hp. (Fig. 5). The Curtiss Company has just added to its line of engines an inverted six-in-line of 125 hp. There are also several others in the development stage or about to be placed on the market.

Strange as it may seem, it is believed that the advent of cowling for the radial engines, which materially reduces their drag, has furthered the sentiment in favor of the in-line type. Until the airplane manufacturers had their attention called to the high resistance of the radial by cowls designed to reduce this resistance the radial was supreme. One manufacturer who had been using radial engines without cowling and without complaint made the remark that if the radial must have all of that cowling he no longer wanted to use it. However, jesting aside, the resistance of the radial, together with its interference with the pilot's vision, is a serious objection to the type and must be considered especially in the larger units and in planes designed for high speeds. The power required to drive a radial engine through the air at 100 miles per hour has been found to be as much as 20 per cent of the total power. At high speeds the percentage becomes greater, as the engine and other parasite resistance varies as the square of the speed, whereas the wing drag, being dependent only on the total load and the lift drag ratio of the aerofoil used, may increase or decrease with increase of speed.

However, the radial still has a number of advantages over the in-line type and undoubtedly will continue to be used by many

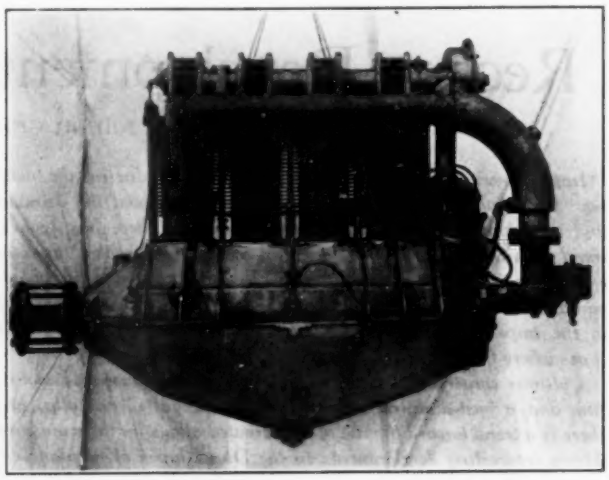


FIG. 5 DAYTON AIRPLANE ENGINE COMPANY BEAR

manufacturers. The radial offers no serious problem in securing uniform cylinder cooling, makes excellent use of crankcase and crankshaft material, and does not have crankshaft torsional vibration to contend with. The in-line air-cooled engines are inferior to both the radial air-cooled and the in-line water-cooled in these respects. Uniform cylinder cooling is not easy to attain and requires cowling to direct the air flow. Due to the finning of the cylinders and the necessity for providing sufficient spacing for air flow between cylinders, the cylinder spacing is comparatively large. This accentuates the poor utilization of crankcase and shaft material by making these parts longer than otherwise would be necessary, and by requiring a longer shaft introduces shaft-whipping problems. The increase in weight per horsepower over that of the radial, due to the poorer use of crankcase and shaft material, may be partly or entirely offset by utilization of higher speeds, made permissible by the lower crank pin loading with only one rod and piston per crankpin, and to the decrease in valve-gear reciprocating weight made possible by the use of an overhead camshaft.

There is one engine, the Curtiss Challenger (Fig. 6), that falls between the radial and the in-line engine so far as the crankshaft and crankpin loading is concerned. This engine, rated at 170 b-hp., has a two-throw crankshaft with three cylinders operating

from each crankpin. In addition to the reduction in crankpin-bearing loads, a reduction in weight of the counterweights is accomplished by this construction.

Another engine, the Comet, the product of the author's company (Fig. 7), has an intermediary position in so far as valve gear is concerned. It has a positive cam which relieves the valve springs of all inertia loads other than the valve itself. Since in all valve mechanisms the cam at some period of operation has to accelerate all of the parts in the gear as well as to overcome the valve-spring tension, it is apparent that this method of drive reduces the cam loading considerably.

The valve mechanism of the radial engine, when not compensated, has certain disadvantages which are not encountered in the in-line types. This refers to the very considerable change in tappet clearances with change in cylinder temperatures. It is not unusual for the tappet clearance to increase from 0.010 in. with a cold engine to 0.060 in. with the engine warmed up. Were

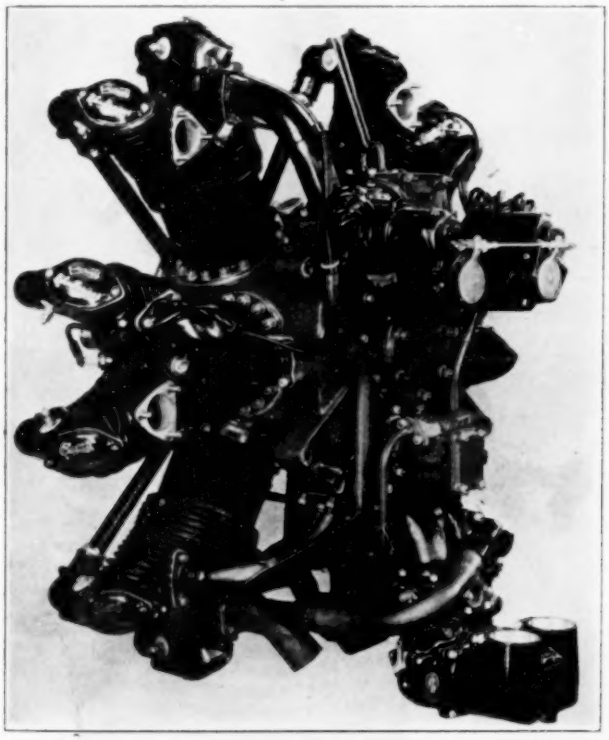


FIG. 6 CURTISS CHALLENGER

it not essential to limit the tappet velocity at the time of opening and closing of the valve and not have excessive accelerations, this would not be serious. However, with these limitations, this change in tappet clearance, even with careful cam design, causes a very appreciable difference in valve timing between a cold and a hot engine. Since the cam is designed to give the correct timing with the engine warm, the timing with the engine cold is very poor. In some engines the cold timing has as high as 50-deg. overlap and the intake valve does not close until 80 deg. after bottom center. This has a considerable detrimental effect on the starting and idling characteristics of the radial.

In the Bristol Jupiter engine a compensating arrangement is used. In this engine the rocker-arm fulcrum is not supported on the cylinder head, but is carried by a rod extending down to the crankcase. Since the supporting rod and the pushrods expand and contract together, very little change in tappet clearance occurs.

In the Wright J-5 engines (Fig. 8) a partial compensation was secured by supporting the rocker fulcrum on a lever, one end of which was fastened to the cylinder head and the other to the crankcase. The rocker box constituted the lever and the push-rod housing furnished the attachment to the crankcase.

In the Comet engine (Fig. 9) the intake valve is over-compensated and the exhaust valve has no compensation. The over-compensation of the intake valve is attained by the use of a

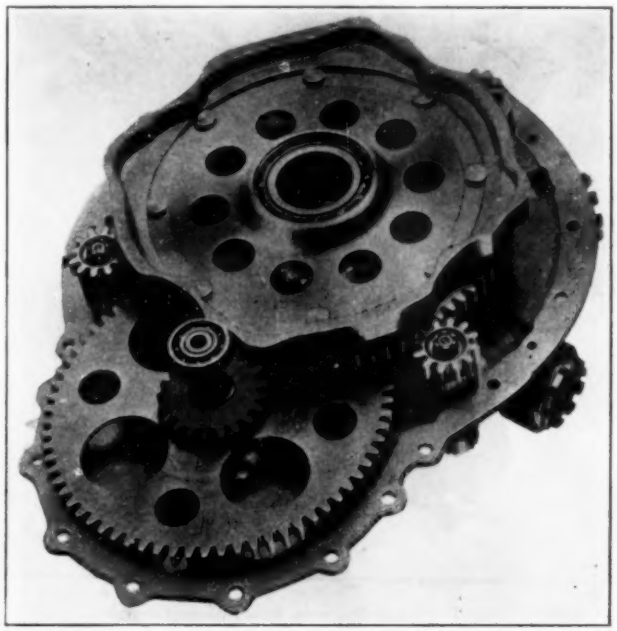


FIG. 7 COMET ENGINE CORPORATION ENGINE CAM

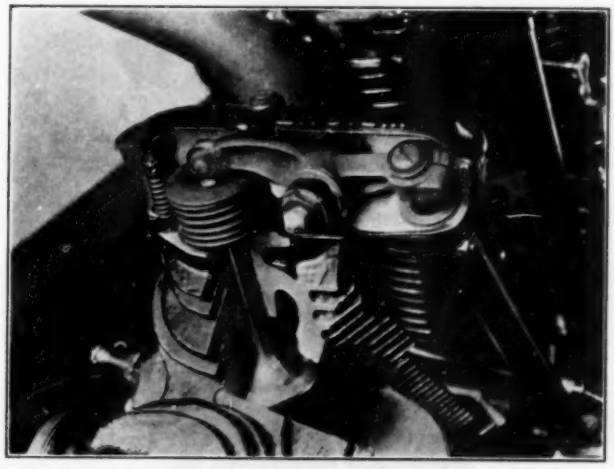


FIG. 8 WRIGHT J-5 VALVE GEAR

single rocker arm and push-and-pull rod, the intake valve being operated by pull on the rod. With this construction the tappet clearance is less with a hot engine than with a cold engine, and consequently the period of intake-valve opening is less with the engine cold. This greatly improves both idling and starting characteristics.

Manifolding is also somewhat of a problem in a radial engine due to the impossibility of getting symmetry, except in the nine-cylinder types, where three three-way manifolds may be

used. There has been a trend this year toward the use of diffusers or superchargers with the object of improving distribution and also, in some cases, of increasing the volumetric efficiency.

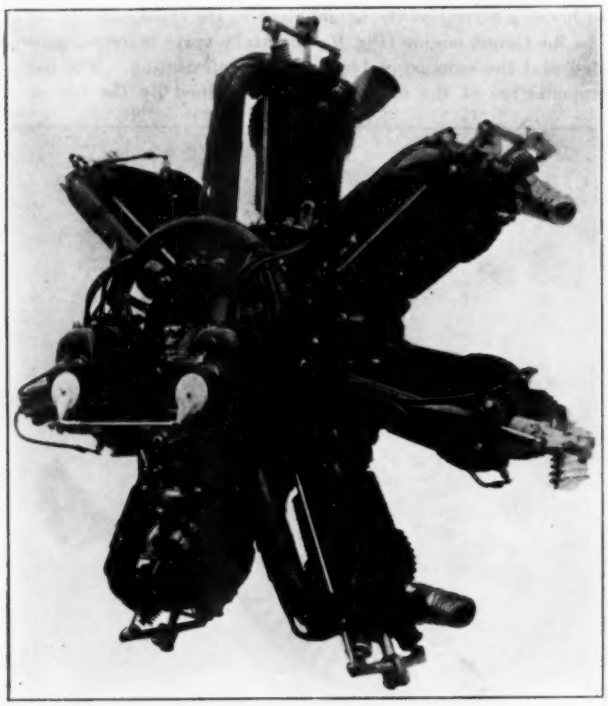


FIG. 9 COMET ROCKER-ARM CONSTRUCTION

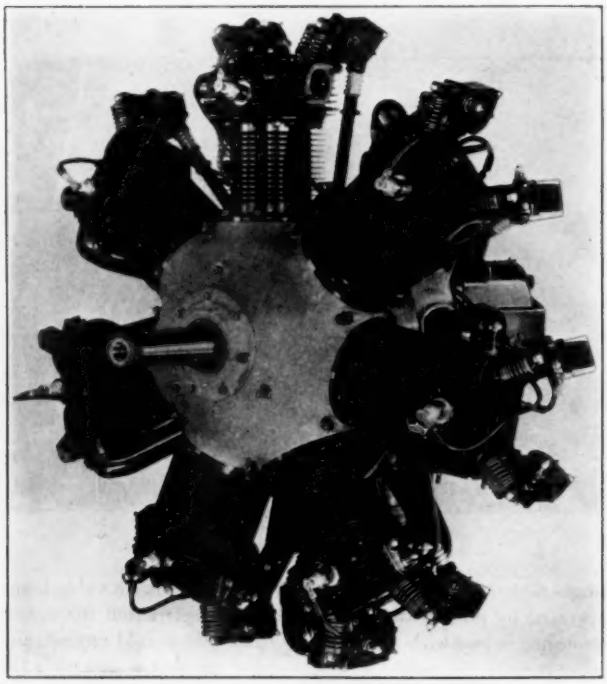


FIG. 10 WARNER SCARAB

Whether this device will be the ultimate solution of the problem is somewhat questionable. In one test of three engines of the same power rating, the engine without a diffuser gave better fuel economy than the two so equipped. However, in this case, all engines had nine cylinders, and the one without the diffuser had symmetrical three-way manifolds. It is possible that, were the comparison made of the best seven-cylinder manifold without diffuser and a seven-cylinder with diffuser, the results would be reversed.

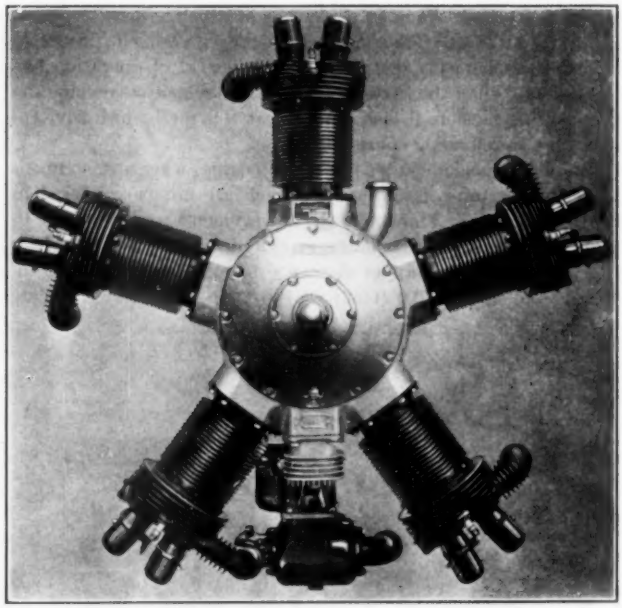


FIG. 11 KINNER, WITH BOLTED-ON HEAD

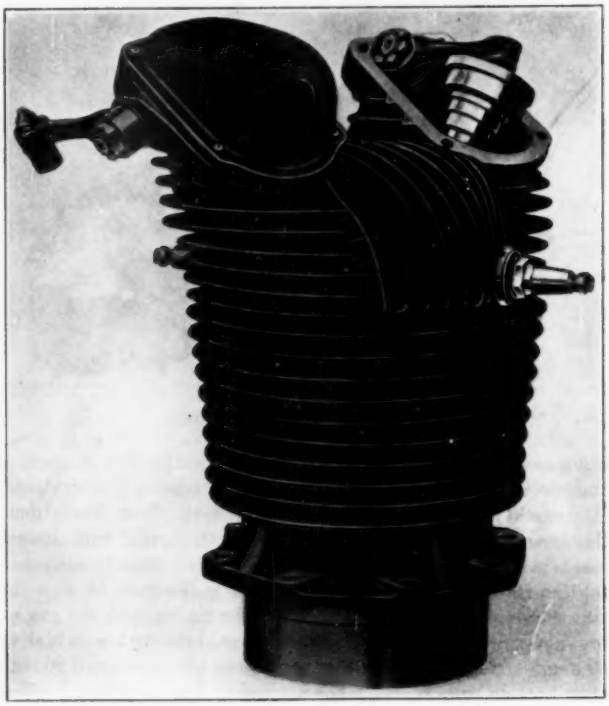


FIG. 12 LE BLOND CYLINDER

The advantage of increasing the volumetric efficiency with geared-up diffusers, or superchargers as they should be termed, is also subject to argument. Assuming a certain grade of fuel, it

is of course necessary to use a lower compression ratio with the supercharged engine than with one not so equipped. This means a reduction in thermal efficiency and an increase in fuel consumption. In addition, supercharging increases cylinder, valve, and piston-cooling problems.

There has been an almost general acceptance of the cylinder construction comprising a steel sleeve having machined cooling fins and a cast-aluminum head screwed and shrunk on the sleeve. Exceptions to this practice are noted in the Warner Scarab (Fig. 10), which has a bolted-on head which is not intended to be removable, in the Kinner (Fig. 11), which has a bolted-on head that is removable, and the Le Blond (Fig. 12), which uses a one-piece cast-iron cylinder. In most cases the heads are cast in "Y"-alloy which retains a relatively high strength at elevated tempera-

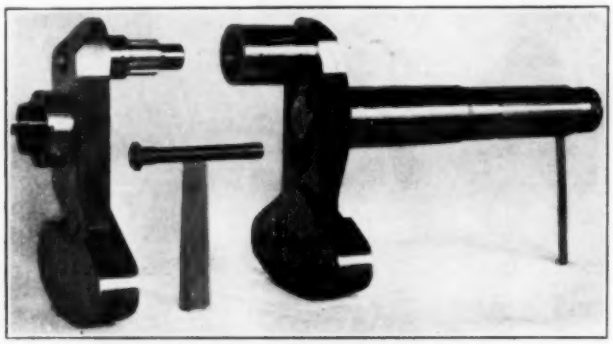


FIG. 13 PRATT & WHITNEY SHAFT

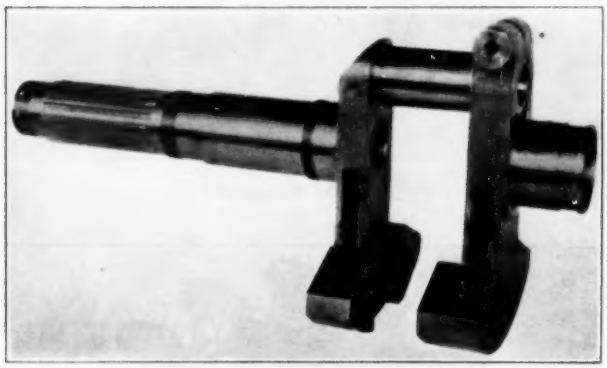


FIG. 14 WRIGHT J-6 SHAFT

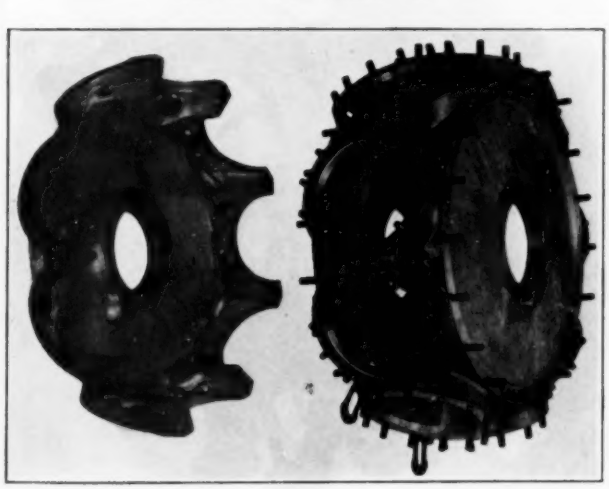


FIG. 15 PRATT & WHITNEY FORGED CASE

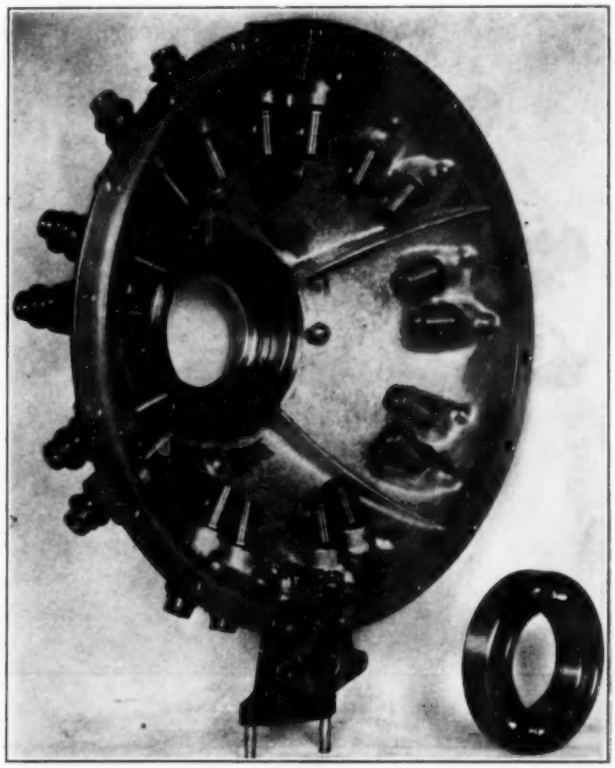


FIG. 16 PRATT & WHITNEY FORGED NOSE

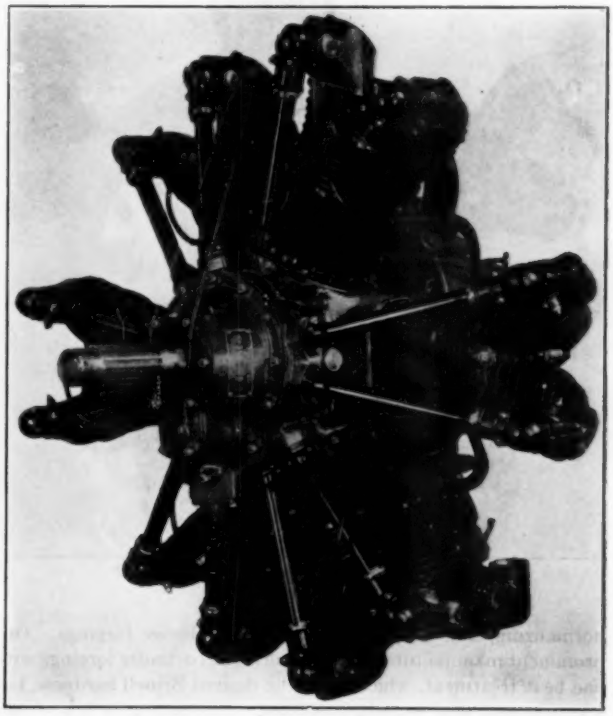


FIG. 17 CURTISS CHALLENGER

tures, giving about 26,000 lb. per sq. in. at 500 deg. fahr. The cylinders are generally machined from a 1050 S.A.E. heat-treated forging. A new development of this year is the elimination of

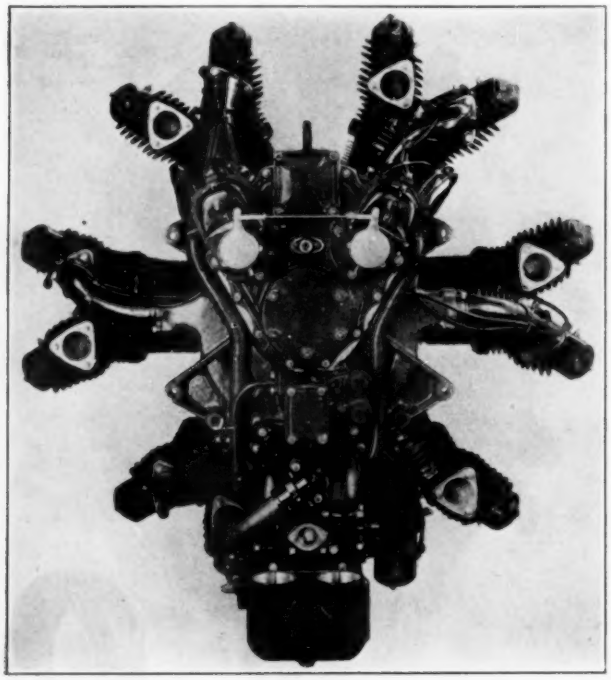


FIG. 18 CURTISS CHALLENGER

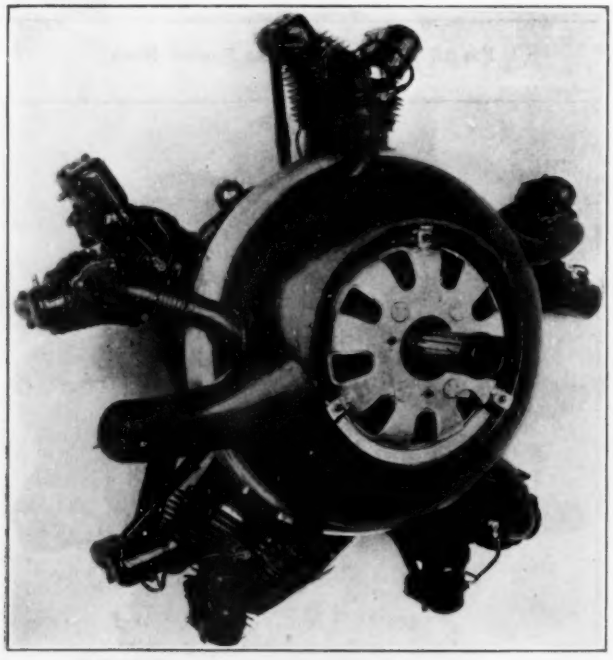


FIG. 19 WRIGHT J-6

normalizing, and the reheat-treating of cylinder forgings. One prominent manufacturer is now securing his cylinder forgings with one heat treatment, which gives the desired Brinell hardness, but leaves a very coarse grain structure. The saving in heat-treatment cost is not particularly great, but the material in this condition is very much easier to machine than when normalized and reheat-treated to give the same Brinell hardness.

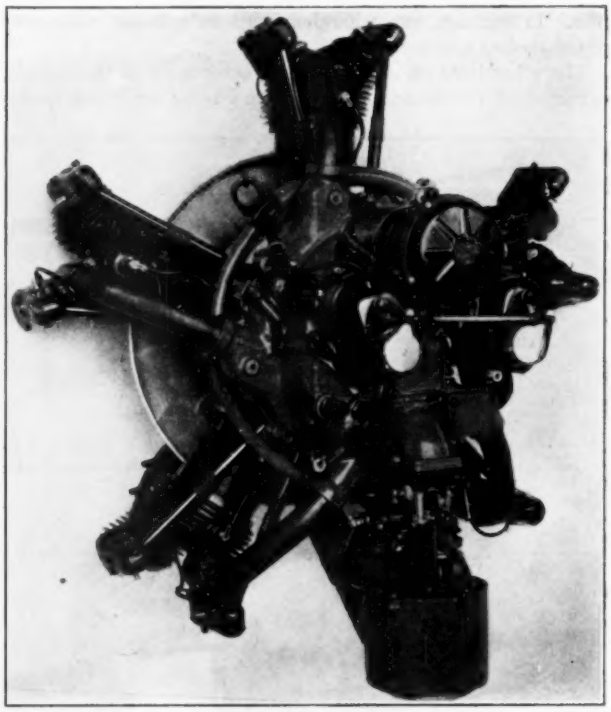


FIG. 20 WRIGHT J-6

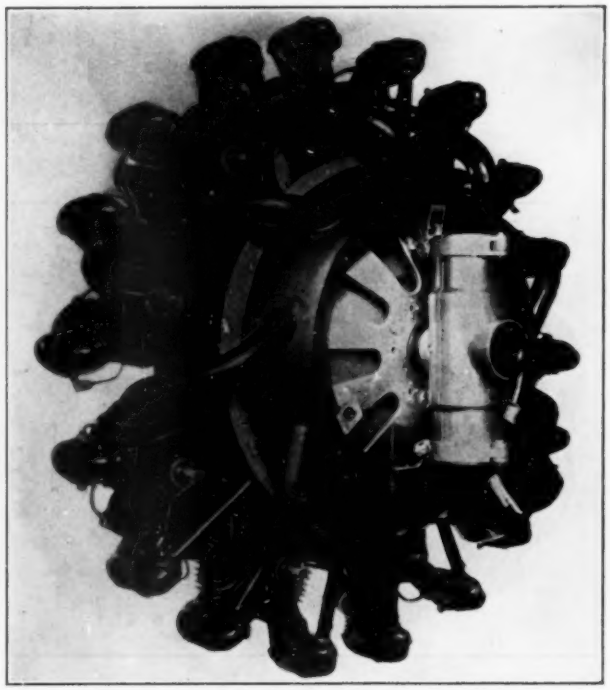


FIG. 21 LYCOMING

Split crankshafts permitting the use of solid master rods also have gained adherents during the last year. Two methods of joining the halves are in use. One is used by Pratt & Whitney

(Fig. 13) and consists of a maneton on the rear half fitting into the crankpin of the front half. The other, of which the Wright J-6 is a typical example (Fig. 14), consists of clamping the rear cheek to an extension of the crankpin. In some of the latter

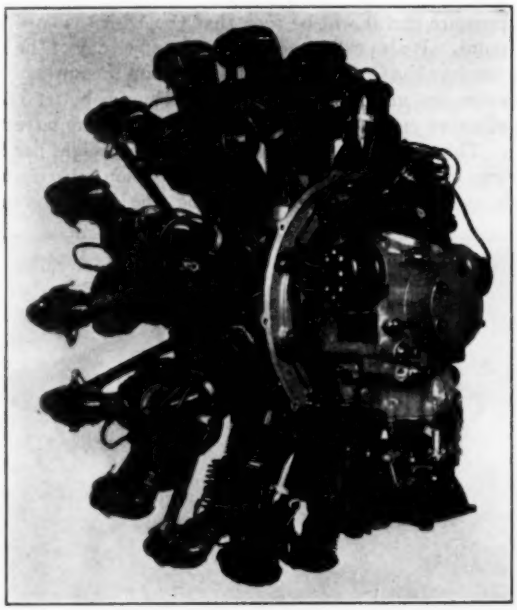


FIG. 22 LYCOMING

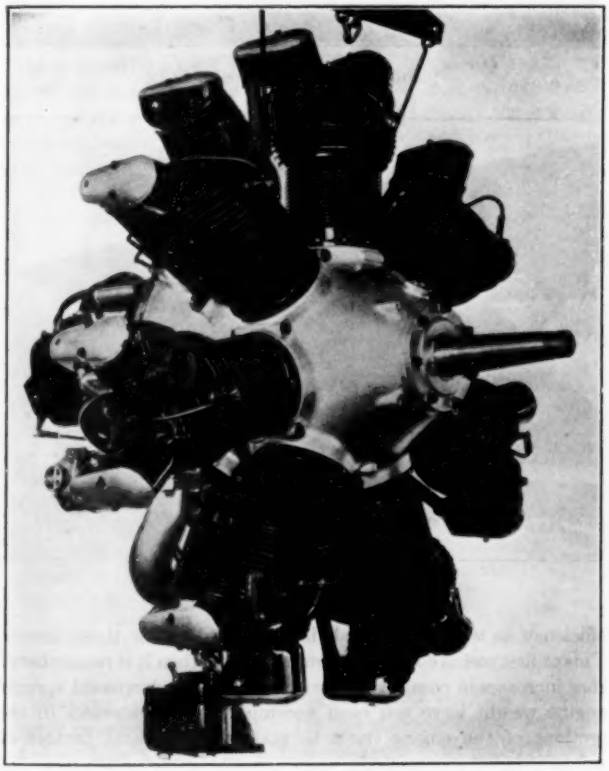


FIG. 23 CONTINENTAL

adaptations a key is used to resist torsional stress in the joint, and in others the friction of the surface is relied upon to prevent slipping.

There has been a general trend toward the use of heat-treated alloys for the crankcase. These alloys have a tensile strength of 30,000 to 40,000 lb. per sq. in. in the heat-treated condition in contrast to the approximately 20,000 lb. per sq. in. obtained from the more common No. 12 alloy. Due to the limitations imposed

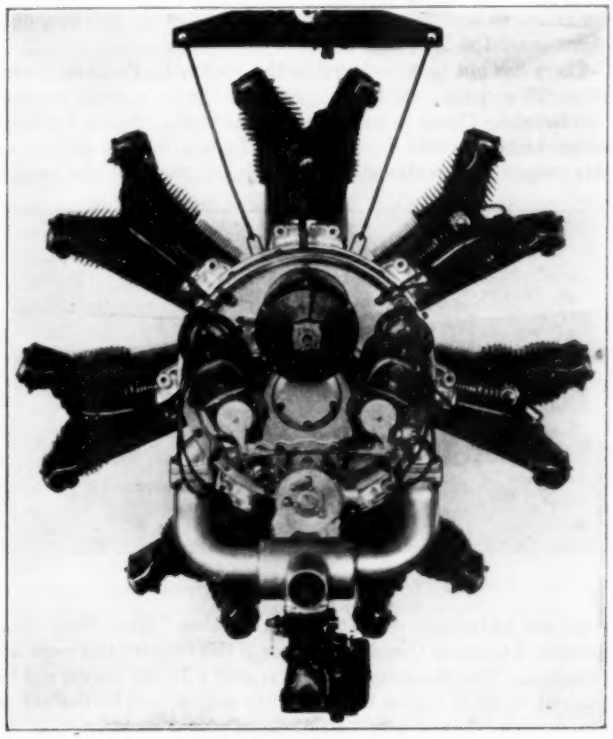


FIG. 24 CONTINENTAL



FIG. 25 PACKARD DIESEL

by casting difficulties on minimum wall thickness and to the variations in wall thickness in the usual castings which may adversely affect the heat treatment, all of this gain in strength cannot be utilized for weight reduction. These considerations, along with the higher price of the castings, make the use of these alloys questionable in many cases. The use of forged crankcases is

still limited, it is believed, to Pratt and Whitney (Figs. 15 and 16). Curtiss is using some magnesium castings, which permit a considerable saving in weight where strength is not required, but which are quite expensive.

Of the new engines, those drawing the greatest interest are the Curtiss 150-hp. Challenger (Figs. 17 and 18), Wright 165-hp. J-6 (Figs. 19 and 20), Lycoming 150-hp. (Figs. 21 and 22), and Continental 150-hp. (Figs. 23 and 24.)

There has not been included in this group the Packard Diesel (Figs. 25 and 26), which of course does create a great amount of interest in Diesel power plants for aircraft. Before Packard stepped into this field it was rather difficult to interest any one in this project. The almost general opinion was that the weight



FIG. 26 PACKARD DIESEL

could not be brought down to a usable value. Now there are a number of aviation Diesel enterprises in this country and more are expected. It is the author's opinion that a Diesel engine will be evolved which is lighter than the Otto engine, and he has yet to find any one who can give a good reason why it should not be.

To secure such an engine, three things will be necessary. First, it must be a two-stroke-cycle-engine—not at all as difficult a problem with Diesel engines as with Otto engines. Second, it must be operated at speeds common to Otto practice. Third, it

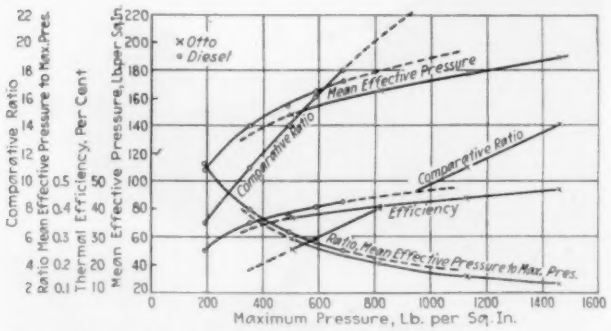


FIG. 27 COMPARISON OF DIESEL AND OTTO CYCLES AND THERMODYNAMIC CHARACTERISTICS

must burn the fuel at substantially constant pressure. This is where the difficulty comes in. Many will say that it cannot be done. It would be better to say that it has not been done, at least generally. The National Advisory Committee claims to have operated a Diesel on such a cycle at 1700 r.p.m. But regardless of this, is there any fundamental reason why it cannot be accomplished? It is not believed so. It is even possible now to operate an Otto engine with almost constant-pressure combustion by use of special fuels. How much has been done on developing fuels for the Diesel engine with this same purpose in view? Nothing that is known.

With reference to the curves of Fig. 27, the author will explain why he contends that the Diesel engine will be lighter than the Otto. The relative weights of the two types will be roughly proportional to the ratio of brake mean effective pressure to maximum pressure. These ratios are plotted against maximum cylinder pressure and it will be seen that the Diesel is superior at every point. It also will be noted that the efficiency of the Diesel is greater than the Otto for identical maximum pressures. These curves also are interesting in that they show a better ratio of mean effective pressure to maximum pressure as the latter is reduced. This means that in the Otto engine the weight per horsepower can be made lower with a low compression ratio than with a high compression ratio, maintaining the same volumetric



FIG. 28 WATER-COOLED D-12

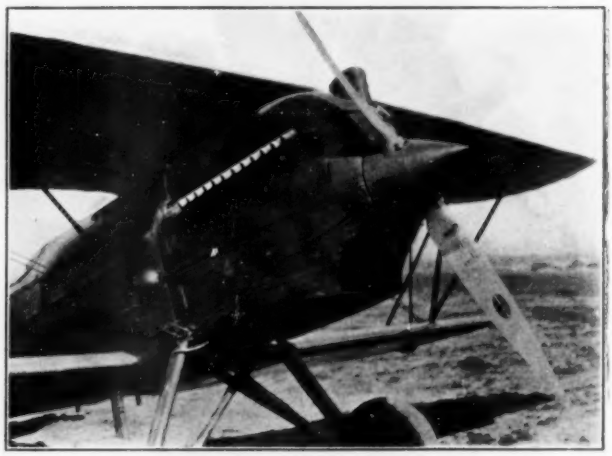


FIG. 29 PRESTONE-COOLED D-12

efficiency as was done in calculating the data for these curves. This at first seems contrary to practice, but when it is remembered that increases in compression ratio which have decreased specific engine weight have not been accompanied by increases in the sections of the engine parts to maintain the same factors of safety it is not inconsistent.

The author's belief is that for transport work in the not too distant future the Diesel engine will reign supreme. The advantages of the Diesel are well known to all and need only be itemized here. They are dependability, reduced fire hazard, better fuel economy, and lack of radio interference. The Diesel of the future, it is

expected, will be Prestone-cooled, and that brings up the development which is considered to be the most interesting of the year.

At the Cleveland S.A.E. meeting, Mr. Frank, of the Army Experimental Station, Wright Field, presented the first paper on the subject of high-temperature liquid cooling. Both the Army and the Navy have been working on this development for a number of years, but have not made public the results of this work. The Army has been concerned with the reduction in radiator size which can be attained by the operation of the radiator at temperatures greater than water permits. Ethylene glycol, sold under the trade name of Prestone, was found to be a suitable liquid for this purpose, having a boiling point of about 380 deg. fahr. and a low freezing point. Tests were made in the Curtiss D-12 and Curtiss V-1540 engines, and the results were entirely satisfactory. Only minor changes in these engines were required to adapt them to the use of high jacket temperatures. The reduction in radiator size accomplished is well shown by Figs. 28 and 29. Fig. 28 shows a plane with a radiator for water cooling. Fig. 29 shows the same plane with the same engine, but with the radiator required for Prestone cooling.

But interesting as this may be, the goal of the Navy Department is still a step farther. The Navy also started out with the goal of reduced radiator size due to higher radiator-surface temperature, but about three years ago the author became convinced that the radiator could be eliminated entirely. Up to that time all had been overlooking the reduction in radiator size which was being accomplished by increasing the heat dissipated from the cylinder jackets themselves. A few calculations showed that, with the temperatures permissible with Prestone, sufficient finning could be provided on the cylinder-jacket surface to dissipate all of the heat.

There was some delay in getting started on this phase of the development due to rush of other work and the lack of an engine which would lend itself to the modifications necessary. However, a Wright E-4 engine without any modifications was successfully run without a radiator on a relatively cold day in the course of tests to determine the effect of the higher jacket temperatures on detonation.

This development appears to offer the solution of most of the present problems. In-line engines are in demand due to their better visibility and lower head resistance. As pointed out earlier in this paper, they have some objectionable features in the direct air-cooled type. The difficulty of obtaining uniform cylinder cooling, the necessity of the use of cowling, and the relatively large cylinder spacing resulting in increased weight and more flexible shaft were mentioned. None of these applies to the Prestone-cooled engine; and furthermore, the most serious objection to water-cooled engines, i.e., trouble with the so-called plumbing system, also does not apply, since no plumbing system will be required. It is even quite possible that the circulation pump can be dispensed with and dependence placed on thermosyphon circulation in the jackets.

In closing, the author wishes to express his thanks to the engineers of the various engine companies who so kindly supplied him with data and photographs.

Discussion

ROBERT INSLEY.[2] As the author points out, the old controversy between the radial and the line engine seems to be in grave danger of settlement by actual comparison of the types in service. It is unfortunate that the majority of the line-engine experiments have been undertaken in a power field where the radial engine has such outstanding advantages. The relative advantages of the line engine become more attractive as the size of the power plant increases. The inherently superior features of this type (smoothness, low frontal area, small obstruction of vision) are rather insignificant in the smaller sizes (roughly 200 hp. or less), and such disadvantages as weight and cost are magnified in contrast with similar radial engines. From that point of view it seems regrettable that the first direct comparison should be made in this field. But this much is certain: if the results of this comparison should favor the line engine at all, its superiority in the larger sizes is fairly well assured. It would seem to be a fair guess, however, that the radial engine will hold its own in the smaller sizes and that the line engine will appear attractive in the larger horsepower classes. One sentence of the paper in this connection seems to deserve comment. With reference to the line engine the author remarks that "uniform cylinder cooling is not easy to attain and requires cowling to direct the air flow." That is perfectly true, but it is doubtful if the amount of cowling required by the line engine is much greater than seems to be necessary for satisfactory operation of large radial engines.

The comments concerning the usefulness of the so-called "diffusers" are particularly interesting in the light of recent comparative performances with and without this method of induction. The behavior of some later models of unsupercharged radial engines with respect to distribution, acceleration, and ease of starting suggests that perhaps our expectations in the way of supercharged performance have been too optimistic. It cannot be denied that the supercharger affords a means of developing more horsepower per cubic inch of displacement; but have not some of our recent unfortunate experiences with such superperformance engines indicated that the unsupercharged engine with reasonable compression ratio will develop as high volumetric efficiency as is desirable for general commercial service? The author's doubts on this subject seem well justified.

The paper cites a new development in "the elimination of normalizing and the reheat-treatment of cylinder-barrel forgings," but makes no comment on the effect of this process on physical properties. Unquestionably this shortened process saves a few cents per forging, but it also definitely reduces the strength and wear resistance of the material. One is tempted to inquire whether in the normal course of development a reversal of this process would not be considered a step in advance.

The author questions the advisability of using heat-treated aluminum castings for crankcases. It is true that in unstressed wall sections the minimum thickness required for good casting properties affords sufficient strength in non-heat-treated castings. In the highly stressed parts, however, such as mounting bolt lugs, cylinder flanges, and bearing webs, the saving in weight possible in the heat-treated castings will well repay the slight additional cost. In discussing the use of magnesium-alloy castings in parts where strength is not required, the author apparently has iforgotten that the strength of the cast magnesium alloy is distinctly better than common aluminum casting alloys, approximately the same as "Y"-alloy, and that the elongation is nearly twice that of the best aluminum alloy in use for aircraft-engine parts. The cost, piece for piece, is approximately 25 per cent higher than aluminum, which, together with its susceptibility to corrosion, has restricted the general use of this material.

In the midst of the widespread enthusiasm for the Prestone-cooled engines the author's comments on this subject are particularly interesting. His suggestion that total elimination of the radiator is not at all out of reason is very alluring; but, after all, it may be asked if some of the chief advantages of the liquid-cooled engine are not the possibility of placing the radiator where it will do the most good and the least harm, the "buttoning up" the engine inside the cowling where it is protected from the elements and from violent temperature changes, and the possibility

[2] Vice-President, Continental Aircraft Engine Company, Detroit, Mich.

of direct control of the engine temperature—all of which would be impossible if the engine were exposed to the direct air blast without the benefit of streamlined cowling. Is there not a possibility that by this semi-direct cooling there may be sacrificed some of the benefits of the compactness and low head resistance of the line engine and thus combine some of the disadvantages of both types? If not, the possibilities of weight and cost reduction are very attractive.

C. Fayette Taylor.[3] The author is to be congratulated for his truthful statements about the condition of the aircraft industry. The industry has suffered considerably from overoptimism and from failure to face the facts.

The author makes the statement that a radial engine does not have crankshaft torsional vibration to contend with. Recent experiments have indicated that this is not the case, but that torsional vibration in radial engines is a serious problem. It has been found that, when radial engines are run at or near the torsional critical speed of the crankshaft, failure always occurs within a comparatively short time.

In his discussion of Prestone cooling the author mentions the possibility of using thermosyphon circulation. The writer does not believe that this will be possible, because in order to take advantage of the high temperature of the Prestone, it will be necessary to hold the surface of the radiator or cooling fins at approximately the temperature of the liquid. This would be impossible without an extremely high rate of liquid circulation. In other words, when we extract heat from the cooling surface at an extremely high rate, we must also make sure that there is an equally high rate of heat transfer from the liquid to the cooling surface. Without this, the temperature of the cooling surface will fall and the rate of heat dissipation will be correspondingly reduced. This may prove to be an important factor to contend with in Prestone cooling systems.

P. B. Taylor.[4] The paper points to the excessive drag and obstruction to vision of the radial type, but does not refer to the sight developments which aim to eliminate these disadvantages in large-displacement engines.

The in-line engine accomplishes this result for small engines, but is limited to six or possibly eight cylinders by a practical consideration of length available for the power plant. The largest air-cooled cylinders at present displace in the neighborhood of 215 cu. in. This size is not likely to be increased greatly in the near future as the experience of engine builders has demonstrated a limiting law; engine trouble varies as the square of the displacement of one cylinder.

Using the largest air-cooled cylinders developed, the in-line type is limited to 1290 cu. in. for a six and to 1720 cu. in. for an eight. Both the six and eight will be extremely long because of the large cylinder bore. The maximum rotative speed also will be limited by the weight of reciprocating and rotating parts for the large cylinders.

The twelve-cylinder inverted air-cooled V-engine is suggested as one solution to this problem. This arrangement has excellent balance and presents the possibility of doubling the displacement of an in-line six without interfering with vision or adding more than 30 per cent to the frontal area, considering both engines as equipped with cowl, without which neither will operate.

The displacement per cylinder for the twelve-cylinder engine when compared with a similar single-row radial or in-line engine is necessarily considerably less. The twelve-cylinder engine can therefore be constructed with small-displacement cylinders which permit of high rotative speed. The necessary spacing of cylinders to allow air to flow between the barrels leaves room for adequate bearings for the high-speed operation.

The bottom and side cowlings are simple pieces of sheet metal which can be designed by the engine manufacturer for engine cooling without reference to structural considerations on the individual airplane. The cowl forms a streamline over the cylinder block and adds little or nothing to the drag. Cooling of this engine as observed from block and flight tests is somewhat better than of the radial engine due to the efficient utilization of the air entering the V in passing over all sides of the cylinders.

The engine can be constructed with overhead cams and full-lubricated valve gear, thus eliminating the maintenance wear and high inertia forces of the push-rod construction and again permitting high operating speeds. The weight, while somewhat greater than the radial, is less than the water-cooled V-type due to the elimination of water, radiator, and plumbing. Since no counterweights are necessary, this feature compensates to some extent for the increase of crankcase, crankshaft, and connecting-rod weight over that of the radial engine.

The possibilities of the air-cooled V-engine are shown by the characteristics of the Wright V-1560, which is under development by the Wright Company for the U. S. Army. These are as follows:

Model	V-1560
Type (inverted air-cooled)	60° V
Number of cylinders	12
Bore, in.	5
Stroke, in.	$6^{5}/_{8}$
Piston displacement, cu. in.	1560
Normal r.p.m.	2400
Maximum designed r.p.m.	2700
Normal b.m.e.p., lb. per sq. in.	$128^{1}/_{2}$
Normal b. hp.	600
Weight bare, lb.	825
Weight with reduction gear, lb.	900
Overall length, in.	$70^{7}/_{16}$
Additional length of reduction gear, in.	$6^{11}/_{16}$
Overall height, in.	$35^{1}/_{4}$
Overall width, in.	$32^{1}/_{4}$

(A supercharger is incorporated for sea level pressure at 12,000 ft. altitude.)

It will undoubtedly be possible to increase both the r.p.m. and the b.m.e.p. after sufficient experimental work. A rating of 140 b.m.e.p. for military work at 2600 r.p.m. is within the realm of immediate possibility, giving a horsepower rating of 715 and a weight rating of 1.15 lb. per hp. Since the frontal area of this engine is no greater than a small fuselage, this type should help to fill the present demand for greater performance.

Sanford A. Moss.[5] The author gives scant attention to superchargers, although the centrifugal supercharger is in use on the majority of aircraft engines now being sold. Doubling the A.C.C. figure of 3275 would give approximately 6550 engines sold for 1929. About 5000 superchargers will be sold, nearly all of which have been used on American engines. Superchargers are therefore used on a very large proportion of all engines sold in 1929.

It would be desirable if the author could give some truly comparative data regarding fuel consumption. It is to be noted that the true comparison would have to be between two engines delivering the same power at maximum efficiency. In this event the cylinder displacement of the supercharged engine would be 10 to 20 per cent less than that of the unsupercharged engine, so that the friction and radiation loss would be less.

The power for compression in the supercharger of course relieves the engine cylinder of this duty.

[3] Consulting Mechanical Engineer, Massachusetts Institute of Technology, Cambridge, Mass.

[4] Wright Aeronautical Corporation, Paterson, N. J.

[5] Engineer of Supercharger Department, General Electric Company, West Lynn, Mass. Mem. A.S.M.E.

The meager data which exist indicate that there is at least no loss in efficiency with a supercharger. There is certainly a decrease in weight and cost of engine for a given power and a marked improvement in smoothness of operation, due to better vaporization.

It is sometimes assumed that, when an improvement is made in utilization of materials so that a greater amount of service is secured with a given weight of parts, there is necessarily a loss in durability. This of course is not true, and supercharged engines are as durable in service as the non-supercharged engines. The matter of durability is entirely one of detailed design of the individual parts, and this may be done well or poorly, entirely independent of the use of superchargers. A manufacturer who does not use a supercharger, and thereby necessarily has a greater weight of engine for a given power, cannot make merit of this fact alone and claim that because it is heavier his engine is more durable. In other words, it does not necessarily follow that a heavy engine is a good engine. The criterion for durability is not a matter of weight for a given power at all or a matter of the use or omission of a supercharger, but is a matter of detailed design of the engine.

Author's Closure

Mr. Taylor is perfectly correct in stating that radial aircraft engine crankshafts are subject to torsional vibration. However, the recent experiments to which he probably referred indicated that the radial engine could be run for long periods at its critical speed without resulting in crankshaft failure. Mr. Taylor is also perfectly correct in his statements relative to the use of a thermosyphon circulation system. It undoubtedly does require greater cooling area than would be required with a forced circulation. However, the simplicity of the thermosyphon system makes it very attractive, and it is quite possible that the increase in weight which would be required in the heat-dissipating elements would be more than offset by the saving in weight due to the elimination of the circulating pump.

Dr. Moss requests some comparative data regarding fuel consumption, which I am sorry to say I do not have. He states that the comparison should be between two engines delivering the same power, but to be at all conclusive the data really should be on one engine, both with and without supercharging. I would like to refer Dr. Moss to a test which was made a long time ago on a Liberty engine which I believe will give him the information he desires.

This test was made by the Navy Department so as to determine the best combination of compression ratios and carbureter choke sizes for the engines used in the NC-4 boats which were the first to cross the Atlantic. These tests indicated that the best performance was obtained with the higher compression ratio and smaller carbureter chokes.

I take exception to Dr. Moss's statement in which he indirectly accuses me of claiming durability goes with a high weight per horsepower. I make no such claim, but do claim that the use of low mean effective pressures has a very great tendency to increase durability.

Due to the fact that Dr. Moss's statement is almost a direct challenge to the company I represent, I would like to call his attention to the fact that though our engine has a low mean effective pressure, its weight per horsepower compares very favorably with some engines using a General Electric supercharger.

The Aircraft Diesel Engine

By LEIGH M. GRIFFITH[1] and EDWARD T. VINCENT,[2] BELL, CAL.

In this paper the authors present information to show that the Diesel or fuel-injection engine in some of its modified forms is well adapted to the requirements of airplane and airship propulsion. They believe that its first and most important commercial application will be in relatively large units on large aircraft and that increased fuel economy, both as to weight and cost, will be an important factor in the progress of commercial air transportation. They consider that the greatly reduced fire hazard will be an important factor in the development of quantity air transport of passengers and that the absence of radio interference will greatly simplify the problems of navigation and communication. The potential advantages of two-stroke cycle operation will be realized in the Diesel engine. In large airplanes from one to three engines of large size will be used, and these will be mounted and enclosed so as to allow of attendance during flight. Such engines will be fluid-cooled, either by liquid circulation or evaporation. The service reliability of the Diesel, they believe, will exceed that of the carbureted, air-cooled engines inaccessibly mounted, and the problems of weight and cost are less serious than are generally supposed, and will soon be solved commercially.

DURING the past few years there have been many published discussions regarding the possibilities of the Diesel engine as a power plant for aircraft propulsion. While the detailed opinions of the various writers have differed greatly, there has been an underlying agreement that the Diesel engine will have a large field of application in aeronautics, when and if the weight per horsepower is reduced to a value sufficiently low to compete with the standard carbureted type of aircraft engine. Most Diesel engineers have taken a pessimistic attitude. The flight of the Packard-Dornier engine a few months ago probably provided the first real jolt of evidence to American engineers that the aircraft Diesel was no longer a mere dream, although the Beardmore airship Diesel had yielded comparable results some time previously, and tests at the N.A.C.A. research laboratory had long shown the entire feasibility of Diesel operation with engine structures of acceptable weight. The recent flight of the Junkers 600–700 hp. Diesel has provided a further demonstration.

The problems connected with the development of Diesel engines of low weight and possessing operating characteristics satisfactory for this service are most interesting and have attracted the careful consideration of engineers connected with both the aeronautic-engine industry and the Diesel-engine industry. It is proposed in the present paper to discuss the general considerations involved, to make an attempt to evaluate quantitatively the possibilities of the Diesel engine as compared to the present carbureted engines, to analyze briefly the technical problems involved, to note the magnitude of recent advances and accomplishments pertinent to this application, and to discuss the probable type of airplane most suitable for Diesel application.

The merits of the Diesel engine as an aeronautic power plant have been frequently pointed out, but may well be again enumerated for the purpose of the present discussion. They include:

(1) Decreased cost of fuel
(2) Decreased fire hazard
(3) Non-interference with radio communication
(4) Increased radius of operation
(5) Special adaptability to two-stroke cycle
(6) Special adaptability to large units.

The disadvantages usually cited against the Diesel as an aircraft engine are:

(*A*) Increased specific weight
(*B*) Increased first cost
(*C*) Increased difficulty of maintenance
(*D*) Fuel less cleanly
(*E*) Fuel not generally available.

In order to establish a basis for analysis of the subject let us consider the comparative merits of the two forms of engine from the standpoint of installations at the opposite ends of the

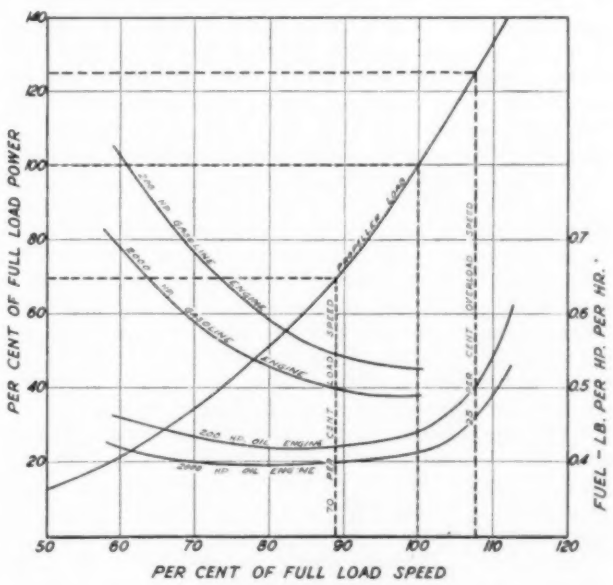


FIG. 1 FUEL CONSUMPTION OF GASOLINE AND OIL ENGINES AT VARYING LOADS

TABLE 1 COMPARISON OF TWO FORMS OF ENGINE INSTALLED IN TWO TYPES OF AIRCRAFT

	Sporting	Transport
Type of service		
Rated power, hp.	200	2000
Cruising power, 0.75 load, hp.	150	1500
Cruising speed, m.p.h.	120	120
Carburetted Engines:		
Gasoline consumption, lb. per hp. per hr.	0.55	0.50
Gasoline consumption, lb. per hr.	82.5	750
Gasoline consumption at 6 lb. gal., gal. per hr.	13.75	125
Gasoline cost at 20c. per gal., dollars per hr.	2.75	25.00
Gasoline cost per mile, dollars per mile.	0.023	0.208
Diesel Engines:		
Fuel consumption, lb. per hp. per hr.	0.42	0.40
Fuel oil consumption, lb. per hr.	63	600
Fuel oil consumption at 7.2 lb. gal., gal. per hr.	8.75	83.3
Fuel oil cost at 5c. per gal., dollars per hr.	0.44	4.17
Fuel oil cost per mile, dollars per mile.	0.0036	0.035
Saving per hour, dollars per hr.	2.31	20.83
Saving per mile, dollars per mile.	0.019	0.173
Saving in fuel weight, lb. per hr.	19.5	150

size scale, such as a 200-hp. airplane used largely for sporting purposes and a 2000-hp. airplane employed in regular commercial-transport service. Consider also that these four aircraft

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

are powered with a single engine each and that all cruise at 120 m.p.h. on three-quarters power. Table 1 gives data for these units as based on factors believed to be reasonable.

The cost of gasoline is relatively unimportant for the small airplane, being less than 2½ cents per mile, while for the large unit it approximates 21 cents per mile. The use of a fuel costing one-fourth as much as aviation gasoline would mean a saving of only 2 cents per mile or $2.30 per hour for the 200-hp. unit, but would approximate 17 cents per mile or $21 per hour for the large unit. The $2.30 saving would have little effect on the total operating costs of the small airplane, even though flown frequently, but the $21 saving would be an important item in the operating costs of the large airplane used continuously in regular transport service. Decreased operating costs of the large unit due to lowered fuel costs alone would approximate the fare of two passengers, and this saving certainly would be a material aid in putting air transport on a paying basis.

The probable decrease of fire hazard realizable by the use of the less-inflammable fuel is difficult to evaluate. Certainly it must be admitted that Diesel fuel with its flash point at not less than 150 deg. fahr. is far less likely than aviation gasoline, or even automobile gasoline, to ignite as the result of a fuel leak or a crash. There can be no question of this if the source of potential ignition be a spark or flame. There is a difference of opinion, however, when the potential ignition cause is a hot surface. Laboratory experiments with hot plates indicate that, under some conditions of limited temperature and air flow, some fuel oils will ignite from a hot plate more readily than will aviation gasoline, thus leading to a limited conclusion that the fuel oil was no safer. However, the experience to date with aeronautic and automotive Diesels, as well as the extensive experience with land and marine Diesels, tends to show that the hot-surface ignition risk is in reality practically non-existent. This opinion has been confirmed by several personal experiences in which fuel oil was sprayed from burst fuel lines or spilled over exhaust pipes or other hot surfaces during tests, but in no case did inflammation result. Even when the fuel oil is ignited, the spread of the flame is relatively slow, and there is therefore a much greater opportunity for extinguishing or rescuing.

While the passenger in the small airplane would probably appreciate the lessened fire hazard as much as those in the large plane, it appears that the pilot of the smaller aircraft usually keeps closer to landing fields, and has a relatively much better opportunity to make a prompt landing in case of actual or threatened fire. The large airplane on a transport route is usually not as free to make immediate landings, and the larger passenger capacity makes the risk responsibility much greater, so that it appears that the large airplane, especially when used for passenger transport, is most in need of the fire-hazard reduction offered by the Diesel engine.

Elimination of the high-tension ignition system made possible by the Diesel greatly simplifies the problem of radio communication with the airplane. The standard ignition system can be shielded so as to nearly eliminate interference with radio signals, but the resulting complication increases both the first and maintenance costs, and has the serious disadvantage of reducing the reliability of this most important element of the carburetted engine. For the present comparison it may be said that only a very small percentage of the smaller airplanes will be equipped with radio, either for communication or direction-finding purposes, so that this advantage of the Diesel is of small importance in that case. The large airplane will most certainly be equipped for two-way communication as well as direction finding, and no interference with good radio reception or transmission can be tolerated. The inherent advantage of the Diesel in this respect is therefore of most value in the large aircraft.

For reasons similar to those governing in the item of fuel costs, the small airplane derives little real advantage from the increased radius of operation made possible by the Diesel. The cruising-load fuel consumption of the 200-hp. unit may be taken at 0.55 lb. per hp-hr. for the standard gasoline engine and 0.42 lb. per hp-hr. for the Diesel, as shown in Table 1. This increased radius of around 30 per cent is not of very great value to an airplane which ordinarily makes only short flights. The reduction of fuel load is also unimportant, even for flights of two or three hours' duration.

The case is very much different, however, for the 2000-hp. unit in transportation service. The specific consumption of such an engine at cruising load would be about 0.50 lb. per hp-hr. for the gasoline engine and 0.40 lb. per hp-hr. for the Diesel. The increased radius of 25 per cent would be an important item in this large unit, as the resulting increase of pay-load capacity would yield immediate returns of economic importance. For a four-hour trip the actual increase of pay load would approximate 600 lb., the equivalent of three or four passengers.

The foregoing comparative figures are based upon more or less ideal conditions, and it is probable that the comparison of actual service results would be somewhat more favorable to the Diesel engine. The modern carbureted aircraft engine can be made to show excellent fuel economy on the test stand, but it is difficult if not impossible to realize such economies under actual service conditions.

Considerations of reliability operate to compel the use of mixture ratios in service which are considerably richer than those accompanying the low fuel consumption under test conditions. Easy starting, prompt warming up, smooth idling and running, cool operation, snappy acceleration, decreased fire risk from backfire, and other desirable attributes are accentuated by operating with a mixture ratio richer than that which gives maximum fuel economy. The carbureted engine which can be made to show a specific consumption of 0.45 lb. per hp-hr. on the test stand probably has an actual service consumption rate between 0.55 and 0.60 lb. per hp-hr. The limited experience accumulated to date with high-speed Diesels coming within the general requirements of the aeronautic type tends to show that the fuel consumption of the Diesel will show very little increase under service conditions. It is believed that the aeronautic Diesel which will yield a consumption of 0.35 lb. per hp-hr. on the test stand will give a normal service consumption rate not exceeding 0.40 lb. per hp-hr.

The preceding discussion serves merely to indicate the order of the fuel-economy advantages offered by the Diesel in air-transport service. Considered from the economic standpoint only, the prospect of carrying three or four additional passengers on a four-hour trip, and at a saving in fuel cost more than equivalent to the fare of still another passenger, can hardly fail to interest the large commercial organization faced with the problem of creating or increasing operating profits.

The power output of any given engine at a given speed being a function of the product of the b.m.e.p. and the frequency of power strokes, it is apparent that the power may be increased by doubling the number of power strokes, even though a lower b.m.e.p. is incurred in so doing. This advantage of two-stroke cycle operation has long attracted the efforts of the aircraft- and automotive-engine designer, but the complication and reduced efficiency accompanying all solutions so far proposed in connection with the carbureted engine have prevented the realization of the anticipated advantages. The case is quite different with the fuel-injection engine, however, and the two-stroke Diesel offers a definite means of realizing this increase of power output.

The effective stroke of the two-stroke Diesel piston is less than that of the four-stroke piston by some fraction of the height of

the ports, the i.m.e.p. is slightly lower by reason of the lower combustion efficiency resulting from the less effective scavenging, and the mechanical efficiency is lowered to the extent of the approximate difference between the energy required to move the scavenge air in the two-stroke engine, and the energy to drive the four-stroke engine during the exhaust and suction strokes, so that the two-stroke engine cannot be expected to equal the b.m.e.p. of the four-stroke engine. For any specific application, the factors involved may be so adjusted that the most favorable combination of the b.m.e.p. and specific fuel consumption is obtained. The performance of modern Diesel engines in land and marine service shows that as low a specific fuel consumption can be expected from the two-stroke engine as from the four-stroke type, if a reduction of about 10 per cent in b.m.e.p. is accepted. This loss may possibly be doubled for the higher operating speeds required of aeronautical Diesels.

Experience in the laboratory and with actual production high-speed automotive four-cycle Diesels shows that at the present time there can be secured and maintained in ordinary practice a full load b.m.e.p. of 100 lb. per sq. in., with a fuel consumption of 0.40 lb. per hp-hr. and an overload b.m.e.p. of 120 lb. per sq. in. at 0.48 lb. per hp-hr. at reasonable maximum pressures. Those figures would be reduced to perhaps 80 and 95 lb. per sq. in. for the two-cycle engine at the same fuel-consumption rates, so that the power output per cylinder would be increased by about 60 per cent. This increased power will be obtained at a cost of perhaps as much as 25 per cent increase in engine weight, represented mainly by the scavenge blower, so that the weight per horsepower should be reduced by about 20 per cent. As the space occupied by the engine is reduced by a greater percentage, the advantageous possibilities of the two-stroke aeronautical Diesel are most interesting.

This favorable comparison does not hold for most of the two-stroke Diesel mechanisms now in use, however, as many of these are inefficient in their scavenging or combustion functions or involve too high friction losses in complicated mechanisms. High-percentage scavenging must be secured with a minimum volume of air supplied at a minimum pressure. Exhaust ports must be of minimum height consistent with good scavenging, in order that the effective expansion stroke may not be unduly shortened. The fuel spray must be of correct characteristics and form. The combustion-chamber shape must suit the fuel spray form, and effective turbulence must be secured. The scavenge air pump must be efficient, and the engine structure itself must be of simple design involving those features making for high mechanical efficiency.

Our consideration of the merits of the Diesel as an aircraft engine having confirmed the original statement that this engine is of particular advantage for large units, let us now take up the disadvantages cited. It has been frequently stated that the Diesel must be much heavier than the carburetted engine because of the inherent limitations imposed by the lower b.m.e.p. and the much higher combustion pressure. Frequent repetition of this opinion has caused it to be generally accepted, notwithstanding the disproving accomplishments of Beardmore, Maybach, Junkers, and several other organizations. Since weight of engine structure is roughly proportional to the magnitude of the maximum forces handled, and power output is proportional to b.m.e.p., the realization of a light engine is dependent upon a low ratio of maximum to mean cylinder pressure. This is equivalent to saying that the material of the engine structure must be usefully worked at stresses as uniform and high as possible. The exhaust and inlet strokes of the four-cycle engine are direct violations of this requirement, since only useless inertia loads are incurred. An extension of this reasoning leads to the conclusion that double-acting two-stroke cylinders would be ideal from this standpoint of engine-structure weight utilization, although the gain from this last step would be limited by material fatigue induced by the reversed high stresses.

A few writers have contended that the attainment of acceptable b.m.e.p. and fuel economy in high-speed Diesels depends upon such an early injection as to insure combustion of all or nearly all of the fuel under constant-volume conditions. The sharp-topped or Otto cycle card resulting in that case has a very high maximum pressure value of the order of 1200 lb. per sq. in., and this high pressure involves the serious disadvantage of a correspondingly heavy engine structure. The constant-volume combustion-requirement theory has been disproved by laboratory and development tests of sufficient number to be conclusive, so that advanced workers in this field know that it is not unduly difficult to secure a reasonably high b.m.e.p. combined with a low fuel consumption at maximum pressures as low as 600 lb. per sq. in. That this accomplishment is not particularly new is indicated by data in Proc. I.A.E., Vol. XXII, 1927–28, showing that the R.A.E. 20T experimental Diesel engine had some time previously yielded 107 lb. per sq. in. b.m.e.p. at 1200 r.p.m. on a fuel consumption of 0.443 lb. per hp-hr. with a maximum pressure of 650 lb. per sq. in. The main requirement is to control the injection so that the combustion follows the desired dual cycle, part at constant volume and the rest at constant pressure. A maximum pressure of 600 lb. per sq. in. is not much higher than is regularly encountered in the carburetted aircraft engine, so that an engine structure of similar dimensions should suffice.

The best modern four-stroke carburetted aeronautic engines regularly yield a maximum b.m.e.p. of 140 lb. per sq. in. as compared to the stated value of 120 lb. per sq. in. for the high-speed four-stroke Diesel, but the two-stroke Diesel may show an equivalent value of $2 \times 95 = 190$ lb. per sq. in. On this basis, the weight of the two-stroke Diesel could be some 35 per cent greater than that of the four-stroke carburetted engine for the same cylinder displacement, and this would take care of such slightly higher maximum pressures as would be experienced with proper combustion control. It is to be expected, however, that more difficulty will be experienced in securing a satisfactory weight per horsepower in small units than in large ones, so that here again the Diesel is of most interest for large-powered units.

Owing mainly to the extra high quality of materials and workmanship required in the construction of fuel pumps and injection nozzles, the Diesel engine is more expensive to build than the carburetted engine for an equal number of cylinders. For this reason it is desirable to limit the number of cylinders and to use the two-stroke cycle. This urge toward the use of fewer and larger cylinders operates to advantage in large Diesel engines and tends to equalize their cost as compared with carburetted engines of the same power. It is believed that the continual normal development and simplification of fuel-injection systems will soon reduce the cost of this part of the Diesel engine to such an extent that the two-stroke Diesel in large powers will cost no more than the carburetted engine of equal power and torque uniformity. In the meantime, any extra cost of the Diesel must be amortized from the fuel-cost saving which it makes possible.

As the 200-hp. Diesel must have not less than five four-stroke cylinders or three two-stroke cylinders in order to give an acceptable balance and torque uniformity, the use of large cylinders is prohibited. Not less than three to five injection-system units are therefore required for this small engine. The 2000-hp. Diesel on the other hand may have as few as five large two-stroke cylinders requiring only five injection units, so that the large engine is here again seen to be the most promising field for the aircraft Diesel.

Maintenance of the Diesel has been stated to be more difficult and expensive than that of the carburetted engine. This is

probably true for engines using complicated and delicate fuel-injection systems which require frequent inspection, cleaning, and adjustment. The more advanced types of fuel pumps and injection nozzles, however, will show a surprising ruggedness and long life, so that there is every reason to believe that these seemingly delicate mechanisms may soon be normally expected to be at least as free from need of attention as is the modern carburetor or magneto. Perhaps the best available data bearing upon this point is the record of the Beardmore rail-car engines in regular operation on the Canadian National Railway, where an average of 1200 hr. of operation is secured from the fuel pumps without service attention, while the injection nozzles require only that the carbon be removed from the nozzle tip every 250 hr. or so of operation. In any event, whatever attention is required is more readily assured to the large engine in transport service than to the small engine with its necessarily more casual care, again pointing to the large unit as the legitimate field for the aircraft Diesel.

No one will deny that Diesel fuel oil is less cleanly than aviation gasoline. The fuel oil has a distinctly disagreeable odor, and serves as a medium for the accumulation of a dirty deposit when spilled and not wiped clean. The clean evaporation of aviation gasoline is in sharp contrast, and is a point against the use of fuel oil in the small engines used for sport flying. Special cleanliness provisions are not difficult to secure, however, in the case of large engines in regular commercial-transport service. The further fact that fuel oil is not generally obtainable would be a real difficulty for the small sporting plane which may frequently refuel at other than its home field. The large airplane in commercial-transport service covers a definite route, and private fueling facilities are usually provided at the terminals by the operating company. Emergency supplies of fuel oil may readily be stored at important intermediate fields, the storage deterioration loss being naturally very small with this fuel. The use of fuel oil therefore appears to present no difficulties to the operator of the 2000-hp. Diesel.

It is true, or nearly so, that airplanes are not at present large enough to require power plants as large as the 2000-hp. Diesel. This depends upon what may be considered the best power-plant arrangement for large commercial aircraft. We have been frequently told that the airplane with only a single engine is a poor risk, because, having only one engine to fail, failure meant an immediate forced landing. Proponents of this argument have assured us that the three-engine airplane represented the acme of safety, on the basis that failure of an engine merely necessitates that the airplane land at the next convenient field on its route. This supposed safety might be realized if any two of the engines would sustain the airplane in level flight in ordinary weather and still permit adequate control and maneuverability. In many of the three-engine airplanes so far offered this performance is not realized, so that a failure of one engine generally necessitates an immediate landing. The public has been so thoroughly sold on the supposed supersafety of the three-engine airplane by interests engaged in its exploitation that it will pass by as unsafe a modern overpowered single-engine airplane, and then unhesitatingly climb into any sort of airplane that may be equipped with three engines. The number of recent serious crashes of three-engine airplanes of reputable make, however, has had the effect of weakening the public confidence in this type of safety. Many engineers in the industry believe that the three-engine airplane which will not give a reasonable performance on any two of its engines is really only one-third as safe from the standpoint of power-plant failure as the properly powered single-engine airplane. In many cases the correct ratio is more nearly one-fourth or one-fifth, since each of the three small engines cannot well be as perfect in construction, installation, and maintenance as the single large engine, so that the risk of individual failure is therefore very much greater than for the single large engine.

Moreover, the general practice followed in this country of hanging two of the engines below the wings of a monoplane results in an aerodynamically inefficient monstrosity. The wings interfere with the flow of air through the propellers, while the propellers and engines ruin the smooth flow of air past the wing at that point. This is proved by the fact that two-thirds of the total power of the three-engine combination will yield a superior performance when concentrated in one properly mounted engine. This means that a single engine of power equal to the sum of the three will give the same maximum performance when throttled to two-thirds full power, with resulting increased reliability and fuel economy. The single-engine installation will inherently be considerably lighter, so that its reliability may be still further increased by the strengthening of structural elements, by the addition of duplicate or standby accessories, or by other means of compensating for the failure of those elements in which experience has shown the greatest weakness to lie. If reasonably full advantage is taken of the possibilities of this opportunity, the single-engine airplane will prove more reliable than even the three-engine machine which will really fly on any two of its engines.

As airplanes increase in size, the problems of satisfactory power-plant installation become more serious. Looking forward to the really large commercial aircraft of, say, 100-passenger capacity, one is struck with the obvious conclusion that the power-plant solution is not to be reached by mere multiplication of engine units of sizes now available. Present reports of the new Dornier X seaplane now under construction state that it will be powered with 12 Jupiter engines. This hanging of engines all over a large airplane is assuredly most undesirable from the standpoint of control and accessibility, however valuable it may appear to the airplane designer as a means of minimizing the structural stresses by distribution of weight and thrust. The larger the commercial airplane, and the more importance is attached to reliability, the more certain it is that the power plants will be few in number and so placed that they will be under constant observation and care while in flight. There seems to be no fundamental justification from the load- and thrust-distribution standpoint for using more than four engines on even the largest airplanes now proposed.

Designers are now providing from 60 to 90 hp. per person for the better class of two-engine and three-engine transports. The 100-passenger high-speed transport may therefore require about 6000 hp., and this is given by three 2000-hp. Diesels. As a matter of interest, the Dornier X (also designated "100 Passenger Boat") with its 12 Jupiter engines has a rated total power of just that amount. On the same basis, the 50-passenger transport which may be said to be just around the corner might be powered with two 2000-hp. Diesels or one 3000-hp. unit. The authors believe that the logical development of such large airplanes cannot be expected until engines of correspondingly large size become available.

The modern aeronautic engine is a remarkably reliable power plant. The majority of such failures as do occur originate in the accessories rather than in the structural parts of the engine itself. Of course, crankshafts, connecting rods, valves, and other major parts do break occasionally, but failures of this kind are generally looked upon as evidence of faulty construction, and will soon be considered as inexcusable. The minor elements have not, however, reached as high a state of perfection, the so-called plumbing of the power plant being perhaps the greatest source of weakness. The rise of the air-cooled engine has eliminated the weaknesses of the water-cooling system, but the fuel and oil systems still remain to plague us. The ignition system is perhaps of the same

order of weakness, with the carburetion system close behind.

The comparative reliability of the Diesel in respect to these several features may be at least roughly estimated. The main structural parts may evidently be made just as reliable as for the carburetted engine by adhering to similar stresses, load factors, and temperature gradients. The two-stroke Diesel makes possible the elimination of failures of valves and their operating gear. The fuel supply system up to the engine may be of the same general character for both oil and gasoline engines, but the higher viscosity of the fuel oil reduces the loss through small leaks, while the relatively slight inflammability eliminates the necessity of immediate landing because of fire hazard. The lubricating-oil system would be quite similar, so that the reliability of this element might be said to be unaffected by the engine type. As the Diesel has no ignition system, all failures due to this source are eliminated. The carburetion system is replaced by a number of fuel-injection pumps and spray nozzles, both the total number of parts, and the apparent complication being thereby increased. While this at first glance would seem to increase the chances of failure, there is reason to believe that the comparative ruggedness of the moving parts and the large forces available to operate them will tend to make this system as reliable as the carburetor. Furthermore, failure of an injection pump or nozzle need affect only the one cylinder served thereby, so that the remaining cylinders could continue to function normally, an advantage which could be secured with the carburetted engine only by having a separate carburetor for each cylinder. The scavenge and supercharge blower required for the two-stroke Diesel will offset and be of the same order of reliability as the geared centrifugal-mixture distributor and supercharger now incorporated in many of the carburetted engines. Taking all these points into consideration, and remembering that the two-stroke Diesel need have only one-half as many cylinders or pistons as the four-stroke carburetor engine to secure the same torque uniformity, it is concluded that the two-stroke Diesel especially offers a wide margin of advantage from the standpoint of service reliability.

Failure of important structural members of modern aircraft engines is very rare, and is being further diminished to the vanishing point by constant improvements in design and material. Most power-plant failures arise from causes which in themselves are relatively minor, and which in the majority of cases could have been easily corrected or temporarily repaired if it had been possible to work on the engine during flight. The bulk of future air transportation will be carried in large airplanes or airships for the same reasons that the bulk of ocean transportation is carried by large ships and not by motor boats. To place the engines of such aircraft in an inaccessible position is as fundamentally unwise as it would be to lock the engine-room crew of an ocean liner out of the engine room during the whole voyage. This fact is recognized in airship design, and the engine eggs or nacelles are made large enough to permit a mechanic to work on the engine at any time. As the airplane is more dependent upon the regular operation of its engines than the airship, it is rather peculiar that the large-airplane designers have so seldom made provision for this attendance. A partial explanation applying at least to conditions in this country may be based on the perfection and popularity of air-cooled engines of excellent reliability.

The air-cooled engine can be placed in an engine compartment or nacelle only with the greatest difficulty, and even then the necessary air ducts or shields would prevent effective detail attention by a mechanic. The water- or steam-cooled engine on the other hand can be readily enclosed and made generally accessible. Attendance on such enclosed water- or steam-cooled engines is certainly no more difficult or disagreeable than in the case of the submarine or the destroyer. The use of surface condensers and cabin-heating radiators of small size takes care of the final heat disposal by condensing the steam, thus allowing the greatest latitude in placing the engines.

The difficulty of dissipating the heat from the air-cooled cylinder increases with the cylinder size, probably about in proportion to the increase of the ratio of volume to surface (or directly as the diameter) so that there is some indefinite limit beyond which it is not practical to increase the air-cooled cylinder size. Large engines of that type will therefore involve a large number of cylinders, with resulting great increase of complication and risk of failure. The liquid-cooled cylinder can, on the other hand, be satisfactorily operated in large sizes, so that a large power output can be secured with a relatively simple engine involving the minimum number of cylinders necessary to secure satisfactorily smooth operation. The use of a cooling fluid having a much higher boiling point than water, such as ethylene glycol with which the Army Air Corps has been experimenting, will undoubtedly operate to still further reduce any apparent advantages offered by direct air cooling.

Despite the important advances made in the development of the high-speed oil engine in the last few years there still remains a great deal to be learned. It may be of interest to consider the various problems that need solution before the engine can be considered to be in a position to displace the gasoline engine. These include (*a*) efficiency (thermodynamic and combustion), (*b*) power output, and (*c*) reduction of engine weight.

Thermodynamic Efficiency. Despite the fact that the Diesel cycle is the most efficient one at present in use, it is considered that a great improvement in the thermodynamic efficiency has yet to be made in order to secure the maximum power output that the cycle is capable of delivering. Since so much has been written under this heading in the last year or so, it is not proposed to discuss this in great detail, but rather to set out the divergence of the actual cycle from theory, and consider its effect upon the design, power output, etc.

Experience with the high-speed oil engine has shown that for an efficient unit it is necessary to depart from the Diesel cycle, where combustion occurs at a constant pressure equal to that of the compression pressure, and operate on what is known as the dual cycle where heat addition occurs at both constant volume and constant pressure. This cycle is practically generally used now, whatever the engine speed, but the amount of heat addition at constant volume has been increased considerably in high-speed work. This of necessity involves increase in the maximum cylinder pressure, and the engine design must therefore be modified to take care of the resultant stresses. This increase in the amount of constant-volume combustion results in a more efficient cycle, and therefore a greater power output for a given fuel supply. The maximum efficiency would be reached if all the fuel was burned at constant volume, this cycle being that of the gasoline engine, but possessing a higher efficiency due to the greater expansion ratio possible. In practice it is not possible to utilize constant-volume combustion only, as it is at present considered that the resultant cylinder pressures would be higher than can be handled safely for long periods. The efficiency of the actual dual cycle used falls short of this maximum value, but is still much higher than that of the gasoline engine, which means that the oil engine will deliver more work than the gasoline engine for a constant air charge and given heat supply if combustion is completed. Theoretical reasons thus indicate that the output of an oil engine should exceed that of the gasoline engine by 10 to 15 per cent.

In practice the use of partial combustion at constant volume results in a greater gain in economy than simple theory indicates, since the higher temperatures involved, coupled with the slightly increased time available for combustion, result in a very substantial reduction in the fuel consumption. There is, however,

a limit to the amount of the constant-volume portion of the cycle, and this depends to some extent upon the engine speed. It is estimated that at 800 r.p.m. there is little gain in exceeding 950 lb. per sq. in. maximum pressure, while at 1000 r.p.m. 1100 to 1200 lb. per sq. in. represents approximately the most economical figure. For still higher speeds the limit has yet to be determined, the authors having used 1500 lb. per sq. in. without any sign of the limit being reached. These higher figures are outside the range that can be recommended for present practice, but fortunately the economies obtainable at lower pressures are such that the high-speed oil engine ranks as the most efficient prime mover yet constructed. So far as the authors' experience is concerned the following figures represent closely the most suitable maximum pressures at which excellent all-around economy and reliability can be obtained:

Speed, r.p.m.	Maximum pressure, lb. per sq. in.
600	700
800	750
1000	800
1400	900

Further gains result in actual practice, principally due to the fact that it is not necessary to operate the engine on a constant-mixture strength. At reduced loads a full air charge is still drawn into the engine cylinder, the power output being regulated by the quantity of fuel injected into this air at the end of the compression stroke resulting in low mean-cyclic temperatures and therefore reduced heat losses. In the gasoline engine the mixture strength must be maintained within very narrow limits, and the maximum temperatures of the cycle vary but little with the power, resulting in proportionally increased heat losses at reduced throttle openings. In the case of aircraft full power is ordinarily required for only a very short portion of the time that the engine is in operation. The average output is of the order of 70 per cent of full power at about 85 per cent of normal full speed, and it is at such loads and speeds that any comparison should properly be made. Fig. 1 shows the usual form of power curve for a propeller load, which applies equally well to Diesel and carburetted engines. Propeller-load fuel-consumption curves for both types of engines in two sizes also are shown, the data having been taken from the best available information. Note that the full load specific consumption of 0.41 lb. per hp-hr. for the 2000-hp. Diesel is 16 per cent better than the corresponding value of 0.49 lb. per hp-hr. for the carburetted engine, but that at 70 per cent load the Diesel advantage has increased to 20 per cent. The Diesel advantage is considerably greater in the small 200-hp. engine. Note also that the Diesel will carry a 25 per cent overload on considerably less fuel than the carburetted engine will carry its normal load.

Efficiency of Combustion. The previous paragraphs indicate the possibilities of the oil engine from a theoretical standpoint. In actual practice, it has not yet been possible to run these engines at a mean pressure approaching the theoretical limit. The maximum mean pressure is a function of the efficiency with which the combustion process is carried out. It is here that present knowledge is deficient. Until improvements are made in injection and combustion control, the maximum power output will be much less than that theoretically possible. The gasoline engine runs at approximately correct-mixture strength, and due to the intimate mixture of the fuel with the air, the combustion process is nearly 100 per cent efficient. The whole of the oxygen available in the air charge takes part in the combustion process, and the indicated efficiency approaches very closely to the theoretical limit. The efficiency of combustion is to a certain extent outside the control of the designer of the gasoline engine, so that a definite result is quite certain if the combustion-chamber shape, position of spark plugs, etc., fulfil the necessary conditions for efficiency. But in the case of the oil engine, even though the chamber shape and position of atomizer be ideal, the results obtained will depend upon the manner in which the injection process is carried out. Seemingly slight changes in the sprayer or pumping system often produce a great variation in the combustion process.

If continued progress is to be made, it will be necessary to determine the laws controlling the combustion, to the end that the whole of the oxygen available in the charge can be made to take part in the oxidation process. The fuel has to be sprayed into and distributed throughout the combustion chamber during the firing period, and at present it has been found impracticable to utilize the whole of the air available for the combustion process, an excess of something like 30 per cent usually existing in the exhaust. By the addition of extra fuel it is possible to cause some of this extra air to take part in the combustion process, but only at the expense of a rather rapid increase in fuel consumption, say from 0.38 to 0.48 lb. per hp-hr. for an increase of b.m.e.p. from 100 to 120 lb. per sq. in. This additional consumption is not in itself of great importance since the fuel used is still less than that of the gasoline engine, the greatest objection being the black smoke that is emitted from the exhaust. Provided that this can be tolerated there is no reason why, in the case of an airplane, this maximum output could not be used for the take-off. This smoke is obviously due to imperfect distribution and combustion.

How well this problem of air utilization is to be solved is not easy to say, but the result of years of experience in the testing of all types of oil engines has indicated that at least a partial solution is at hand in the present engine. It is necessary to give very careful attention to the combustion chamber, spray valve, and injection system, analyze their functions, and determine the correct relationship of one to the other in order to secure increased efficiency of combustion. Very low fuel consumptions have been obtained at different times, and when an engine is operating under such conditions (say 0.35 lb. per hp-hr. at 100 lb. per sq. in. b.m.e.p.) the maximum output is of the order of 130 lb. per sq. in. b.m.e.p. This begins to compare favorably with a gasoline engine, but is considered to be capable of material improvement. The attention that is now being given this type of engine should result in such mean pressures being easily available at an early date on four-stroke engines, with a maximum of say 110 lb. per sq. in. b.m.e.p. in the case of two-stroke engines.

Power Output. In judging the relative merits of various prime movers it is necessary not only to consider the thermodynamic efficiency, but also the power output per unit of air consumed, weight of material involved, and the relative costs of construction. It is not proposed to consider the item of costs, since so much depends upon the number of engines under construction. Most gasoline engines are manufactured on a production basis which is conducive to low costs, while the high-speed oil engine is being constructed only in very small numbers at present. Undoubtedly the oil engine will be slightly more expensive to manufacture, due principally to the high-class workmanship required for fuel pumps and atomizers.

Of the technical difficulties that have to be overcome in the development of the Diesel engine, the question of its power output is the most important, and at present it is necessary to install a slightly larger engine for a given horsepower. The usual rating of the four-cycle oil engine is of the order of 100 lb. per sq. in. b.m.e.p. against an average of say 130 to 140 lb. per sq. in. for a gasoline engine. It must be remembered that the latter engine is usually rated at practically its maximum output for any given speed, increased power being obtained only by increase in the speed of revolution, while in the case of the fuel-oil engine the rated output is usually that at which a reasonable fuel economy

and clear exhaust are obtained. The oil engine thus rated has a capacity of at least 10 to 20 per cent greater power for long periods while even 25 per cent could be carried for, say, 15 min. with perfect safety, much depending upon the injection system. The actual maximum power therefore is quite near that of the gasoline engine. However, the best interests of the industry will be served by keeping the maximum output of the four-stroke oil engine within, say, 110 lb. per sq. in. until greater experience is obtained, which means that the oil engine will be rated at 85 to 90 per cent of the power of the corresponding gasoline engine running at the same speed. The maximum output of the two-stroke oil engine should be limited to 100 lb. per sq. in.

The power output from any internal-combustion engine is limited by the following: (*a*) The air drawn in per stroke, (*b*) the efficiency of the cycle, (*c*) the efficiency of combustion, (*d*) mechanical losses, (*e*) speed of revolution, and (*f*) cycle of operations. Since (*b*) and (*c*) have already been considered, attention will be concentrated on the other items.

Air Supply. Little need be said on this point other than to distinguish the differences existing in the two types of engine. The fuel and air are drawn into the gasoline engine cylinder together, and the evaporation of the fuel causes a reduction of air temperature, and results in a higher volumetric efficiency than would otherwise exist. This action does not occur in the oil engine, resulting in a slightly lower volumetric efficiency. Against this has to be set the fact that the exhaust gases left in the clearance space of the gasoline engine are at a higher temperature, and roughly $2^1/_2$ times the volume of that of the oil engine, thus reducing slightly the advantage due to the cooling action of the fuel. On the whole, the volumetric efficiency of the oil engine is slightly less than that of the vapor engine, which, other things being equal, will result in reduced output.

Mechanical Losses. The mechanical losses of an engine determine how much of the work developed in the engine cylinder is actually delivered for useful purposes. These losses depend chiefly upon design, and are affected but little by the cycle employed, and hence the engine losses do not depend upon whether gasoline or fuel oil is used as fuel. Due to the fact that the pressures of the oil-engine cycle exceed those of the gasoline engine, there is possibly a slightly higher piston friction in the former. There is also the loss of power due to the fact that the high-pressure injection pump absorbs more power than the magnetos of the gasoline engine. This effect is very small, however, as the total power absorbed by the fuel pump is little over 1 per cent of the engine output.

Engines of the high-speed mechanical-injection type, with plain white-metal bearings throughout, have shown a mechanical efficiency of 85 per cent. This value is very little different from that of a gasoline engine of similar design, so that differences of mechanical efficiency are of slight importance when considering the relative merits of the two prime movers.

Speed of Revolution. Not many years ago the oil engine was considered to be essentially a slow-speed engine and its application to automotive and aircraft purposes almost an impossibility. With the increasing knowledge that has become available on the injection and combustion of liquid fuels, piston speeds have increased to figures which compare very favorably with the gasoline engine, and excellent test results have been secured at the same time. So far as the authors' experience is concerned there appears to be no limit to the speed of rotation that can be employed, the limiting speed of combustion being far from reached.

It has been stated that the time lag of combustion remains constant, and that therefore satisfactory results can be secured only by using the constant-volume combustion cycle. If this was the case the high-speed engine would be an impossibility. There is a lag of something like 15 deg. of crank angle between injection and combustion, no matter what the speed may be. Consider the case of an engine running at 300 r.p.m. Tests have shown that the lag is of the order given, and this represents a time interval of $^1/_{120}$ sec. Now in the case of an engine running at 1800 r.p.m., the engine rotates through 90 deg. in this same time interval, so that, if no change occurred in the time lag, injection would have to begin something like 100 deg. before top dead center in order to secure ignition at the desired point. Under such conditions detonation would be very severe. Actual measurements have shown that the beginning of injection when operating at 1400 r.p.m. is something like 15 to 20 deg. before ignition occurs, which shows that time lag is not a constant quantity, but depends upon the degree of atomization, compression pressure, etc. There seems to be no reason why the oil engine should not run at any speed that may be necessary. Test results of engines at various speeds have shown that the economy is unaffected by high speed, and therefore there appears to be no factor in the combustion process which will limit the operating speed of the Diesel engine.

Cycle of Operations. Present practice in the aeronautic Diesel engine is solely in the direction of four-stroke engines, but there seems to be no reason why this type of engine should not put the two-stroke principle upon a satisfactory basis due to the use of air only for scavenging purposes, thus eliminating the loss of fuel that occurs in the gasoline type, together with the other objections resulting from the employment of an air-fuel mixture for scavenging.

A two-stroke Diesel engine of the opposed-piston type, experimented with by the authors, developed indicated mean-effective pressures of the order of 135 lb. per sq. in. with a fuel consumption of 0.32 lb. per i.hp-hr. and maximum pressure of 650 lb. per sq. in. when running at 500 r.p.m. Now if this performance can be obtained in an aeronautic engine with a mechanical efficiency of 80 per cent, a maximum output of 108 lb. per sq. in. b.m.e.p. at 0.40 lb. per b.hp-hr. will be possible, and this is equivalent to 216 lb. per sq. in. b.m.e.p. in a four-stroke engine. Even allowing for the higher engine weight mainly due to scavenge blower, there would still be a considerable drop in weight per horsepower over a corresponding type of four-stroke engine.

A four-stroke oil engine weighs approximately 10 per cent more than a gasoline engine of the same type and size, and it should be possible to design a two-stroke engine for a further addition of, say, 10 per cent to the weight. Now if such an engine was conservatively rated at an equivalent output of 170 lb. per sq. in. on a four-stroke basis, the resultant weight per horsepower would be of the order of that of the gasoline type, and the Diesel is at once placed upon a new footing altogether. It is considered that the two-stroke principle when correctly applied presents the best possible power output for a given engine weight, and therefore should be the logical prime mover for aircraft.

Reduction of Engine Weight. It has already been stated that the oil engine weighs approximately 10 per cent more than a corresponding gasoline engine. This additional weight is due to the fact that higher combustion pressures are encountered with this type of engine, and at present there seems little chance of reducing this weight without sacrificing fuel economy to such an extent that the additional fuel required for a long flight will more than offset the reduced engine weight. The research work now in progress may result in lower maximum pressures than those used at present, under 800 lb. per sq. in. being possible, but as aeronautic work requires maximum power and economy consistent with satisfactory service a material reduction of maximum pressure is very doubtful.

Test Results. There are so many variables in engine testing that it is almost impossible to vary them one by one, so that it becomes necessary for very complete data to be available when

tests results from various engines are being compared. The suppression of one or more facts that might appear unimportant may result in very different conclusions being reached. This is so well known that further stressing of this point is unnecessary, and it has been mentioned only because of the new variables that are encountered in considering the compression-ignition engine for aircraft. The essential conditions of engine testing for the gasoline engine and fuel-oil engine are similar, but there is one point that is very important in the case of the latter, and which does not enter into the case of the gasoline engine. This is the maximum pressure of combustion.

The maximum pressure of the gasoline engine is determined by the compression ratio employed, the mixture strength, spark advance, etc., and for engines of the same compression ratio the maximum pressure varies but little when correctly tuned up.

This does not apply in the case of the oil engine, the maximum pressure being one of the most important figures to be included in the test results. If this figure is omitted, as is frequently the case, the power output and fuel consumption are valueless for comparative purposes. The reason for this is the fact that, unlike the vapor engine, the whole fuel supply is not in the cylinder at the moment of ignition, and therefore considerable control of the combustion process is possible. If all the fuel is injected as rapidly as possible, the cycle approaches the well-known constant-volume type, while by distributing the injection period over a considerable crank angle the constant-pressure cycle of the slow-speed Diesel engine is approached. In the first case the maximum combustion pressure will be of the order of 1500 lb. per sq. in. while in the latter the pressure approaches that of compression, say 500 lb. per sq. in. for air-injection and 400 lb. per sq. in. for mechanical-injection engines. By adjustment of the injection device all intermediate conditions can be obtained, i.e., there will be some definite proportion of the fuel burned at constant volume, while the remainder will be burned at approximately constant pressure. This cycle is known as the dual cycle, and it can be shown theoretically that this cycle is more efficient than the constant-pressure cycle, and that the efficiency gradually increases until the constant-volume condition is reached, though due to variable specific heats of the gases of combustion the increase in efficiency gradually reduces.

It is therefore seen that the normal Diesel cycle is the most inefficient type of combustion considered and that the good fuel economy achieved with this type of engine is due to the high expansion ratio employed. Increased economy is obtained by burning some of the fuel at constant volume, but incurs the disadvantage of an increase of maximum pressures. In the case of the slow-speed engine of, say, 100 to 200 r.p.m. the amount of constant-volume combustion possible is strictly limited because of the fact that combustion proceeds very slowly, mainly due to the low value of turbulence. If the speed of injection be increased in an attempt to accelerate combustion, detonation results. Experience with engines ranging in speed from 100 to 1400 r.p.m. has proved that the degree of constant-volume combustion that can be usefully employed is a function of engine speed, increasing as the speed of rotation is increased, until at something like 1800 r.p.m. it is thought that the whole of the combustion process can be carried out at constant volume with advantage despite the resulting high maximum pressures. By this means maximum economy is possible, provided that the engine will withstand the stresses involved. The present state of the art is such that it is considered inadvisable to go to this extreme.

Extended experience with engines of all sizes and speeds has amply demonstrated that when the high-speed engine is under consideration, the maximum pressure is of vital importance. The following representative test figures will illustrate this: When operating at 1000 r.p.m. an increase in full-load maximum-combustion pressure from 750 lb. per sq. in. to 1100 lb. per sq. in. resulted in the fuel consumption being reduced from 0.40 lb. per hp-hr. to 0.355 lb. per hp-hr. while the maximum engine output was increased. Many other similar instances can be quoted to illustrate this, so that a statement of fuel consumption alone without its corresponding maximum pressure is of no use for strictly comparative purposes. Very few of the available test reports include this fundamental information, hence the relative merits of the different injection and combustion systems cannot be ascertained with any degree of accuracy.

TABLE 2 PERFORMANCE OF VARIOUS TYPES OF HIGH-SPEED OIL ENGINES

Engine	No. of cyls.	Bore and stroke, inches	Speed, r.p.m.	B.hp.	B.m.e.p., lb. sq. in.	Max. pres., lb. sq. in.	Fuel cons., lb. b.hp-hr.	Weight, lb. b.hp.
Beardmore.....	6	8.25 × 12	1400	700	103.0	850	0.43	3.36
Beardmore.....	6	8.25 × 12	1007	424	92.0	900	0.365	4
Beardmore.....	12	12 × 12	900	1500	81.0	800	0.42	22.8
Packard.....	9	5 × 5.5	1800	200	90.7	1200	0.44	3
Maybach.....	6	5.5 × 7.125	1300	150	90.0	...	0.49	17.65
M.A.N.	6	4.53 × 7.09	1000	68	78.4	...	0.437	22
Dornier.....	4	3.73 × 6.30	1000	35	100	...	0.46	
Junkers	2	3.15 × 11.82	1200	54	96.5	...	0.38	16
McLaren-Benz .	6	5.32 × 7.87	800	90	84.7	...	0.42	42.3

Table 2 records the performance of various types of high-speed oil engines at their rated outputs. Examination of these results reveal some very excellent performances, and the figures serve to bring out why the type of engine under discussion will be a very serious rival to the gasoline engine as used at present. The maximum pressures are recorded where available. It is seen in lines 2 and 4 that the abnormally high maximum pressures as used in the Packard engine are not essential for high efficiency, since the Beardmore engine has an economy far superior to the Packard engine while only utilizing a maximum pressure of 800 to 900 lb. per sq. in. against 1200 lb. per sq. in. in the latter. It is reported that the Packard engine has given a fuel consumption of 0.35 lb. per sq. in. at approximately 55 per cent propeller load, the maximum pressure being still 1200 lb. per sq. in. (See *S.A.E. Journal*, vol. XXIV, Feb., 1929.) The Beardmore engine also shows lower fuel consumptions if operated at reduced loads. The impression exists in certain quarters that high maximums are essential for success. It is seen that this is not the case, high efficiency being secured with normal maximums. In case it is necessary to obtain maximum economy and power output, the use of higher pressures can still be resorted to.

There is little information available regarding the results of two-stroke engines operating at high speed. The Junkers engine is the only example in Table 2, but its performance ranks among the best. This is attributed to the excellent scavenging of this type of engine, the opposed piston arrangement having the advantage of unidirectional gas flow, and resulting in very complete sweeping of the cylinder free of exhaust gases. The engine also operates at a comparatively high m.e.p. for a two-stroke engine, again due to its excellent scavenging. Results from the slow-speed engines of this type confirm these views. The employment of this system should make the aircraft two-stroke engine a practical proposition.

It is seen that only two of the high-speed engines shown can be considered to be of the aeronautic type, viz., the Packard and Beardmore, each of which weighs approximately 3 lb. per hp. Now some qualification of the weight per horsepower is necessary when considering the oil engine, since, unlike the gasoline engine, a maximum power up to 20 per cent greater than the rated load is possible, if the engine is an efficient one, and therefore the load assumed in calculating the specific weight must be stated. In the *S.A.E. Journal* already referred to, the Packard engine is given as weighing under 3 lb. per hp., but no mention is made as

to whether this applies to its rated load or overload capacity. The Beardmore engine weight is based upon the rated load, and therefore this figure represents what the engine will weigh for each horsepower of continuous output. The specific weight would be decreased considerably if computed for the temporary maximum load.

Detonation. The most important factor controlling the output of the gasoline engine is the limit imposed on the useful compression ratio due to the occurrence of detonation. The phenomenon of detonation is also encountered in the Diesel engine, but is so seldom met with in practice that little is known of the laws governing it. It was not until the high-duty engine appeared that this characteristic was observed at all. The conditions of compression, speed of injection, etc. usually encountered in the slow-speed engine are such that true detonation does not occur. The metallic ring identified with detonation in the gasoline engine is not met with in the Diesel, so far as the authors' experience is concerned, until loads and speeds far in excess of normal practice are attempted.

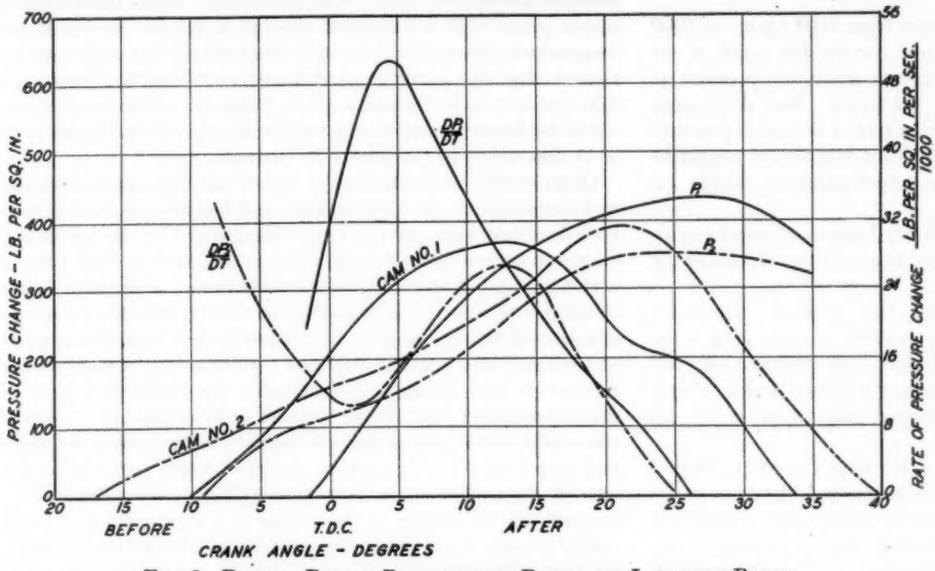


FIG. 2 RATE OF RISE OF PRESSURE FOR DIFFERENT INJECTION RATES

The effects of detonation upon the operation of a Diesel engine were first encountered by the authors while experimenting on a high-speed Doxford opposed-piston engine in which high b.m.e.p. was carried, and what appeared as excessively heavy explosions were encountered, though the indicator diagrams failed to show abnormal maximum pressures. These observations resulted in a close watch being kept on the combustion process while various changes were made in the factors controlling the combustion. A numerical value was given to each set of conditions to indicate the intensity of the resulting detonation as judged by the ear alone. This of course was only a rough estimate, but it served to give some sort of order of comparison.

The indicator diagrams obtained from the engine during this series of experiments were analyzed in the following manner. Combustion diagrams were plotted on a time base, and on the same base a compression and re-expansion curve for no fuel addition was added. The difference in pressure between these two curves represents, therefore, the heat addition due to combustion. This is not strictly true, because work is being done on the piston, but in the early stages of combustion with which this phenomenon is associated the piston is practically stationary and little work is being done compared to the heat added.

The indicator cards suggested that the rate of rise of this pressure increased as the detonation increased, the maximum pressures reached being maintained the same in all cases by adjustment of the time of injection. To examine this characteristic, the ordinates between the curves were plotted on a time base and differentiated, resulting in a curve showing the rate of rise of pressure. Then by plotting the maximum rate of rise of pressure against the detonation number assigned to the test, it was seen that these were related, which indicated that detonation was a function of speed of combustion, and not maximum pressure.

One series of such curves for the Doxford engine is shown in Fig. 2, the fuel cam shape, change of pressure, and rate of change of pressure curves being recorded. Comparison of a large number of such curves indicated that the change of pressure or heat addition follows very closely the cam shape, while the rate of change of pressure increased with the detonation. The results indicate that the rate of injection, other things remaining constant, has considerable influence upon detonation. Further tests with varying compression pressures and rates of injection showed the effect of each of these changes upon the resulting rate of change of pressure, and it was firmly established that detonation is eliminated in the injection engine by increasing the compression pressure, instead of reducing it as with the gasoline engine; also, that a reduced rate of injection eliminated detonation, if the compression was maintained at a constant figure.

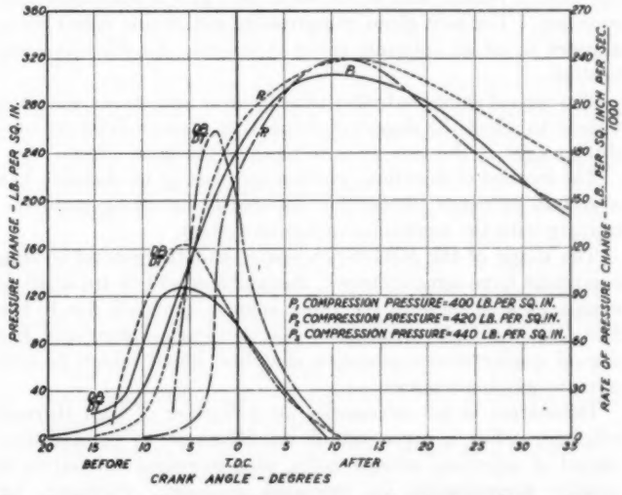


FIG. 3 RATE OF RISE OF PRESSURE FOR DIFFERENT COMPRESSION PRESSURES

These conclusions have been confirmed in subsequent work on engines of various makes and at speeds up to 1400 r.p.m. Fig. 3 indicates the effect of compression pressure upon rate of change of pressure in a high-speed mechanical-injection engine, there being slight detonation with a compression pressure of 400 lb. per sq. in., while there was none in either of the other cases. The only change made during these tests was in compression pressure, the rate of injection and the maximum pressure being

kept constant. It is seen that a slight change in compression is all that is necessary to eliminate detonation, in this case 20 lb. per sq. in. reducing the rate of change of pressure from 193,000 lb. per sec. with 400 lb. per sq. in. to 124,000 lb. per sec. with 420 lb. per sq. in. in compression, and eliminating all traces of detonation. This figure was further reduced to 97,000 lb. per sec. by an additional increase in compression to 440 lb. per sq. in.

The effect of change in engine speed from 1000 r.p.m. to 1300 r.p.m. is shown by the following test results, the point of injection only being altered so as to secure maximum pressure at approximately the same point in the cycle. Bad detonation was occurring at the lower speed with a rate of change of pressure of 270,000 lb. per sec., while the detonation was almost negligible at the higher speed, although the rate-of-pressure change increased to 410,000 lb. per sec.

Considering the results of both air-blast and mechanical-injection engines together, detonation depends upon the following variables:

Compression Pressure. Increasing this pressure eliminates detonation in the injection engine, instead of producing it as with the petrol engine. This result has been observed on two-stroke and four-stroke engines with both air-blast and mechanical-injection systems, and is considered characteristic for all engines of this type.

Speed of Injection. For economical working, the charge should be injected at the maximum possible rate which does not cause detonation or an undesirably high maximum pressure. Injection at a slower rate may give inferior results, but not necessarily so.

Engine Speed. Detonation is a function of the engine speed, the two being related in some inverse manner.

These are the three leading facts, and they work together to a certain extent. Detonation may be eliminated by increasing the compression, by reducing the rate of injection, or by increasing the engine speed. A point is reached, however, with low-compression pressures where no value of injection rate will produce the desired results, and at the same time give a reasonable fuel economy. For any given compression and engine speed there appears to be an optimum speed of injection for efficient combustion.

The type of engine, whether two-stroke or four-stroke, does not appear to affect the degree of detonation, though different conditions exist in the two types to produce the same effect.

The method of injection, whether air-blast or mechanical, has a certain influence, detonation apparently occurring more frequently with the mechanical-injection system.

The shape of the combustion space, and the general cooling conditions have some influence, though these effects are small as compared with those in the vapor engine; probably due to the fact that the high-performance injection engine invariably has a good symmetrical combustion chamber, which cannot be said of many gasoline engines.

Detonation is not necessarily an indication of high thermal efficiency. For a given set of conditions as to compression, period of injection, sprayer holes, etc., increased detonation is usually accompanied by increased economy. However, by changing one or the other of these variables, severe detonation can be produced coupled with a decreased efficiency, and usually with increased maximum and mean temperatures.

It is difficult to say at present whether detonation of the type under review is dependent upon pressure and temperature. The following facts have, however, been observed: Detonation is increased by advancing the injection. Both pressure and temperature are thereby increased, but only so to any marked degree near maximum pressure, which seems to have little effect on the speed of combustion. The correct interpretation of these observed results appears to be that it is the initial stages of combustion that determine whether the charge will detonate or not. By advancing the ignition, the fuel is injected into the combustion chamber earlier in the cycle, and therefore the initial stages of combustion are controlled by a lower compression pressure and temperature. Advancing the point of ignition by 5 deg. means that the first portion of the injected fuel encounters a compression pressure lowered by about 45 lb. per sq. in. Since the diagrams shown prove that a reduction of even 20 lb. per sq. in. in the compression pressure will produce detonation, it is reasonable to assume that the lower compression encountered by the earlier injection will have the same effect, hence the detonation is produced by lower pressures and temperatures, and not higher ones as at first might be thought to be the case.

Observations of the manner in which this phenomenon occurs, and examination of all the results and factors influencing it at both low and high speeds, has confirmed the above reasoning. It is well known that ignition is not instantaneous, that there is a certain lag between injection and ignition. During this time interval fuel is being continually fed to the cylinder at a rate determined by the pump and injection system as a whole, while at the same time vaporization and the formation of combustible mixture is proceeding, and eventually the initial fuel injected ignites and produces a rise in pressure and temperature. Now if the rate of fuel supply during this period has been too rapid, there will be a localized rich mixture produced which under the influence of the conditions in the engine cylinder may detonate in a manner exactly similar to the charge of a vapor engine. If the rate of supply of fuel exceeds some limiting value set by the rate of ignition, detonation may occur due to spontaneous ignition throughout the whole mixture supply. A detonation wave may also be set up in the following manner. The fuel first injected is vaporized and ignited. While this flame is traveling across the combustion space, fuel is being continually supplied, and the combustion chamber behind the flame could, if the supply is sufficiently rapid, be filled with a rich but unvaporized mixture which under certain circumstances could be partially vaporized so that the rise of temperature and pressure due to the combustion of the first fuel sprayed might cause a detonation wave to form in this mixture. This is supported to a certain extent by the observation that detonation usually occurs when a considerable time lag takes place in the engine, due, say, to inefficient atomization.

The effect of injection period and compression pressure are easily explained. Increasing the period of injection reduces the rate of fuel inflow, and prevents the formation of a mixture strong enough for combustion before ignition has occurred, while increasing the compression pressure causes the vaporization and combustion of the initial fuel injected to be accelerated, and rich pockets are again prevented.

It has also been observed that heating the fuel supply produces detonation. This seems to be opposed to the foregoing theory, since it would be presumed that ignition would occur earlier, and reduce the amount of combustible mixture formed. The actual sequence of events is thought to be as follows. The fuel being heated vaporizes very readily, and much of it is converted to a gas soon after entering the combustion chambers, in which state all penetration is lost, so that instead of traveling across the cylinder, the fuel is concentrated near the spraying device, and thus forms a rich mixture in a shorter time interval. This conclusion receives support from the fact that reduced combustion efficiency accompanies heating of the fuel, showing that either the distribution has been upset or penetration is lacking. Since the distribution remains unaltered, so far as it is affected by the direction determined by the nozzle and the turbulence is concerned, there is left only lack of penetration to account for the increase in fuel

consumption, which would also help detonation, and blackening of the exhaust.

This discussion is not intended to convey the impression that detonation is considered a serious problem in the development of the high-specific-power Diesel. The power output and maximum combustion pressure are almost unaffected by detonation, while the fuel consumption may be slightly decreased when the detonation is slight, but will be increased if excessive detonation occurs. The principal objections to detonation in the Diesel arise from the noisier operation and the destructive shock loads on bearings and structural elements due to the greatly increased rate of gas-pressure rise. The importance of detonation as a problem will be further decreased as the operating speeds continue to increase, but may again assume some importance if means are developed for utilizing the whole of the oxygen in the air charge, in which case the mean pressures will be considerably increased.

As the result of his own work and that conducted by the Royal Aircraft Establishment during the past few years, the eminent British research engineer Harry R. Ricardo recently expressed the opinion that the speed of the Diesel or compression-ignition engine is not limited by the combustion processes, that the pumping and metering of the fuel present no serious difficulties at speeds under 2000 r.p.m., that the high efficiency is little diminished by unskilled handling in service, and that the two-stroke Diesel can be built about 10 per cent lighter than the carburetted engine.

Conclusions

In conclusion, we may summarize our views in the following terms. The Diesel or fuel-injection engine in some of its forms is well adapted to the requirements of airplane and airship propulsion. Its first and most important commercial application will be in relatively large units on large aircraft. The increased fuel economy, in both weight and cost, will be an important factor in the progress of commercial air transport. The greatly reduced fire hazard will accelerate the growth of passenger business. The absence of radio interference will simplify the problems of navigation and communication. The potential advantages of two-stroke operation will at last be realized. In the large airplanes, from one to three engines of large power will be used, and these will be so mounted and enclosed as to allow of attendance during flight. Such engines will be fluid cooled, either by circulation or evaporation. The service reliability will greatly exceed that of carburetted air-cooled engines inaccessibly mounted. The present problems of weight and cost are less serious than generally supposed and will very soon reach commercial solution.

Discussion

C. R. Alden.[3] The authors state that the trend of the fuel-injection system owing to the high cost per cylinder of fuel pumps is toward fewer and larger engine cylinders for the Diesel engine as compared with the carburetted engine. Judged by present standards it is granted that this statement is true with at least two additional influences contributing to its truth:

1 For fuel pumps of contemporary design it may be even more difficult and expensive to build a satisfactory fuel pump for small cylinder sizes than for larger cylinder sizes. Engines of the smaller cylinder sizes are called upon to perform at higher rotational speeds introducing the necessity for still further refinements in the design and construction of the fuel-injection system

2 Each increase in the number of engine cylinders increases not only the number of pumps required, but the number of adjustments required to divide properly the load between cylinders, with attendant increase in maintenance costs.

Operating against the logic of decreased numbers of cylinders and increased cylinder sizes, there is the tendency toward increased specific engine weight due to the decreased rotational speed obtainable from engines of larger cylinder sizes, often with attendant increase in engine vibration.

However, the fact that two well-known systems of airless fuel injection have held the center of the stage since the idea of airless fuel injection was first conceived (at least 25 years ago) is not sufficient reason for being tied to the limitations of these two systems forever.

To some who have given much thought to the application of specialized equipment, to the production of precision fuel-injection details, not to mention the possibilities of accomplishing the fuel-injection objective by methods differing widely from the commonly accepted practice of the present day, there remains a confidence that the first cost, weight, and freedom from delicate adjustments of the fuel-injection system can be brought down at least to a level comparable to the present respective attributes of the carburetors, and magnetos eliminated from the aircraft engine by the use of the compression-ignition cycle. This conviction holds without regard to the number of cylinders or size of cylinders likely to be chosen by the aircraft-engine designer.

Philip Lane Scott.[4] There is reasonable ground for saying that the aircraft Diesel engine, the injection engine, will equal or exceed the four-stroke cycle carbureting engine in b.m.e.p., and that a two-stroke cycle injection engine can be built of nearly the same weight as the four-stroke engine of same bore, stroke, and r.p.m. The Diesel will have about twice the power for equal size, and will be simpler and more reliable. Work comparable to that spent in developing the beautifully functioning valve mechanism, and the astonishing ignition and carbureting system of the four-cycle engine will be done in developing the scavenging and fuel-injection systems of the two-cycle injection engine, and the scavenging and fuel systems are inherently simpler elements than their counterparts in the present engines.

Use of the two-stroke cycle may well be emphasized as one of the two major reasons for the development of the injection engine. Why has four-stroke cycle been used almost exclusively? Because feeding a pre-mixed charge and driving out burnt gases with it meant, inherently, serious losses.

Four-stroke cycle engines can be charged with two major breaches of sound design. Such an engine acts only as a low-pressure air pump during two complete strokes, but the parts are designed to carry loads 4000 per cent or more in magnitude. A single low-pressure light-weight unit serving a multiplicity of cylinders would do just as well. The four-cycle engine has a multiplicity of valves and driving mechanism. A few holes in the cylinder wall would suffice if some loss of the scavenging and recharging medium were permissible.

The injection engine at once permits the use of a two-stroke cycle since air alone is used for scavenging and recharging, and a little loss is permissible. Ports in the cylinder wall are substitued for the valves and their drive. A low-pressure air blower replaces the engine itself as an air pump. In many cases four-cycle engines now have air blowers for pressure carburetion and for altitude work, though still compelled to keep the entire valve system.

Aircraft-engine designers have become so wedded to the four-cycle engine that the inertia against change is enormous. When as much effort is spent on the simpler mechanism of the two-cycle

[3] Ex-Cell-O Tool and Mfg. Co., Detroit, Mich. Mem. A.S.M.E.

[4] Consulting Engineer, San Mateo, Cal.

engine as has been on the four-cycle engine a type will result which will work full time, not half time, and in which the extra weight of the air blower may not greatly exceed the weight of the valve mechanism discarded.

The authors, conservatively, place limits unfavorable to the two-cycle engine in the matter of combustion for the present, and also make the point that injection is at present substantially behind carburetion in b.m.e.p. produced even with four-cycle operation.

For equally good combustion the working pressures in a two-cycle engine as against a four-cycle engine depend solely on the amount of oxygen in the cylinder. An opposed piston two-cycle engine (Junkers type) has shown 95 per cent fresh charge. The four-cycle engine has always a clearance volume, some 10 per cent or more of burnt gas to mix with the fresh charge. The two-cycle engine always charges above atmosphere, and the four-cycle below unless it uses pressure carburetion in which case the disadvantage of the extra weight of the air blower on the two-cycle engine is cancelled, for the four-cycle engine must have the same device. A two-cycle engine charging at two pounds above atmosphere as against a four-cycle engine charging at one pound below has a 22 per cent advantage in amount of oxygen, potential power output. This is a power factor of 244 against 100 on a basis of power strokes per minute.

The authors show that the working pressures with fuel injection are some 20 per cent to 40 per cent lower than with carburetion on a basis of rated load for injection, and about maximum for carburetion. The maximums of the two methods are about the same. It is interesting to remember in this connection that many oil engines have been successful commercially with powers based on 30 lb. b.m.e.p., and some as low as 20 lb. Not so long ago a 75 lb. b.m.e.p. was considered good for a gasoline engine. The war development has brought the 140-lb. pressures which injection must strive to exceed.

Combustion with injection of fuel is the phase of the matter about which least is known. The high effective pressures must be obtained without excessive maximum pressure or the familiar black smoke so common at "overload." There are several kinds of undesirable burning found in the case of fuel injection which are not present when a pre-mixed charge is fired though the burn of a pre-mixed charge has its special bad tricks also. A group of several series of indicator cards taken over a wide range of pressure and temperature variation showed quite clearly detonation and rough burning occurring at several distinct places in the runs. Gasoline is made by subjecting the rather unstable molecules of the heavier-oil fractions to various pressure and temperature combinations. It is reasonable to think that the same sort of thing may occur in the engine cylinder in a detrimental way. The physical chemistry of oil should prove a valuable source of information, and aid in avoiding the undesirable combinations. The matter of bringing all the oxygen in the cylinder in contact with the fuel is strictly a test of the skill of the designer in dealing with the spray nozzle and combustion chamber.

The great advantage that injection of fuel has over carburetion and electric ignition is that combustion is substantially under control by the rate of injection, while the flame once started in a pre-mixed charge does pretty much as it pleases. With the carburetor-electric ignition system engineers have learned to prepare and distribute an elastic material, and to handle an almost intangible substance at some 15,000-volts pressure. With fuel injection an inelastic material and only one is dealt with which can be more easily metered and controlled.

A. L. PARKER.[5] The authors assume for the Diesel engine certain probabilities of development for comparison with carburetted aircraft engines of present-day development, with the supposition that the latter has reached its ultimate efficiency, whereas the Diesel engine only has potential possibilities. With this supposition the writer does not agree.

Citations of actual performance of Diesel engines in aircraft service are limited, and there is scant performance data other than pertaining to fuel economy, the first advantage claimed for the Diesel engine.

The authors fail specifically to classify Diesel fuel. Their statement that Diesel fuel costs one-quarter as much as aeronautical gasoline indicates the usual low-grade fuel oil used for stationary Diesel engines. In the aircraft carburetted engine it has been considered advisable to use a fuel more costly than the fuel ordinarily used in carburetted engines, the expensive fuel being desirable because of improved acceleration, reliability, and reduction in service requirements, as well as for the increased mileage obtained. High-test fuel has contributed materially to the performance of aircraft carburetted engines, and it seems likewise probable that development in Diesel fuel may contribute to the performance of the aeronautical Diesel engine in a like manner. Improved vaporization and distribution with extension of the limits imposed by the occurrence of detonation may result from the use of a higher grade fuel, and it seems quite logical to assume that such improvements will be made. Data indicating the effect on performance of Diesel fuels of various degrees of volatility, density, and viscosity would be of interest. Apparently the high-speed Diesel aircraft engines are identical with the stationary commercial engines of this type except that by careful design, weight has been eliminated, and maximum working pressures have been reduced to some extent by departure from the constant-volume cycle with the extension of the injection period over a considerable crank angle.

The crank-angle limits for injection of fuel are of course irrevocably established so that the design problem hinges about control of maximum internal pressures, within the limits imposed by structural strength, and the improvement of fuel distribution in the combustion chamber. Increased maximum pressures necessarily demand increased weight; therefore, attainment in this direction has its limitations. Although injection nozzles with suitable turbulence created may be designed for a fixed speed of any particular engine, it will be necessary to vary the injected charge for different speeds, and the control of the injection conditions to coordinate with varying speeds will probably constitute a serious problem. Speed changes, acceleration, and retardation will be accomplished only by control of fuel rather than by control of fuel-air mixtures. This control of fuel with as simple facilities as characterize the carburetted engine controls seems unlikely as an injection nozzle setting is so exceedingly sensitive, as reflected in the shape and character of the spray, that manipulation will be difficult to accomplish with efficiency and simple means. Change of fuel injected by change of injection-line pressure will always result in varying spray and fuel distribution with partial combustion resulting at variable speeds.

This inherent difficulty of efficiently controlling Diesel-engine speeds is reflected in the general opinion that the Diesel engine is primarily a constant-speed unit. A variable-pitch propeller may, if properly developed, be the answer for the Diesel aircraft motor, as such a propeller may be operated with efficient results without unusually wide range of motor speed. One outstanding advantage of injection engines is their ability to carry increased load without increased speed. The variable-pitch propeller seems to be essential fully to capitalize this valuable Diesel characteristic in air service.

Improvement in distribution of fuel in the combustion chamber is dependent upon refinement of injection nozzles. Although combustion chambers may be designed to create sufficient turbu-

[5] The Parker Appliance Co., Cleveland, Ohio

lence for thorough vaporization and distribution of fuel, no doubt the greatest problem confronting the designer of high-speed aircraft Diesel engines is the construction of a reliable and efficient injection nozzle.

Experiments with injection nozzles seem to indicate that super-accuracy is required, and dimensional limits are so very close that the slightest corrosion or clogging due to carbon or soot is very apt to disrupt performance and decrease efficiency. Efficient injection nozzles will, no doubt, require so accurate design and adjustment to attain the desired fuel spray and distribution throughout the combustion chamber during the limited firing period imposed by high speeds that exceedingly delicate adjustment will be unavoidable. Wear or distortion of these delicate adjustments will require so accurate service that it will be impractical for the average mechanic to service parts of this character. It is doubtful if turbulence alone can promote satisfactory fuel atomization and distribution throughout the charge with a simple rugged nozzle construction.

If, in commercial low-speed Diesel engines, such as are in operation on the Canadian National Railway, it is necessary to remove nozzle tips for the removal of carbon every 250 hours or so of operation, it is quite natural to assume that in aircraft motors of high speed and efficiency with more delicately adjusted injection nozzles, this operation will have to be performed more often. Greater service attention will thus be needed than in the carburetted gasoline engine.

Efficient spray injection without great turbulence is, no doubt, a function of nozzle design, and suitable penetration and distribution of the sprayed fuel will require unusually high injection pressures for high-speed engines with accordingly difficult fuel-distribution problems.

Manufacturers of industrial Diesel engines experience considerable difficulty with fuel-injection lines. Such lines must invariably be of exceptionally heavy construction to withstand possible backfire which occasionally occurs in these engines. These lines must also withstand a severe fluid hammer or vibratory effect as fuel is injected at rapid intervals into the cylinder. As the speed of Diesel engines is increased, and the fuel injected at a higher rate of speed, this vibratory or fluid hammer effect will, no doubt, be materially increased. This will require careful and strong construction of fuel-injection lines.

If the spring-loaded injection nozzle is used, a very small air bubble will quickly disrupt the performance of the motor, which, in aircraft service, would be most undesirable.

Excessive air accumulation in the fuel-injection lines may cause backfire. With the open type of injection nozzle, it is questionable whether fuel pressures could build up rapidly enough to give the desired spray and distribution of fuel. It has been found that an efficient spray requires very high injection pressures, and the trend of design in the Diesel engine is to increase fuel-injection pressure to the very high limits of 4000 to 5000 lb. per sq. in. To carry from 800 to 1500 lb. fluid pressure in any conduit, particularly with rapid periodic impositions of these pressures, creates a plumbing risk far in excess of any plumbing risk to be found in the gasoline-powered ship of today. Should pressures of from 3000 to 4000 lb. per sq. in. be found necessary for really efficient fuel spray and distribution, this plumbing problem is correspondingly complicated.

The authors classify so-called plumbing of the power plant as "one of the minor elements." No part of an airship should be classified as a minor element, and surely such an important part as fuel-distribution equipment may not be so classified. A casualty accruing from plumbing is just as deplorable as one from engine failure, and plumbing is as important a part of the engine as is the crankshaft, piston, or other unit. Tendency on the part of both power-plant and aircraft designers to classify aircraft plumbing as a minor element and to give too little attention to the design and installation of this important part of aircraft construction is responsible for many failures, both in power plants and in ships. Fuel-distribution plumbing is vitally important, and a most essential portion of a ship. It must of necessity spread throughout the ship whether carburetted gasoline or Diesel engine is used, and upon the plumbing is dependent the safe and efficient distribution of fuel for both gasoline or Diesel engines. The plumbing of the Diesel engine is subjected to far greater strains than the gasoline engine, and therefore it is safe to assume that with the adoption of high-speed, high-pressure Diesel engines, the power-plant plumbing must be given the same care and design as is given to fuel pumps, distribution nozzles, and other vital and important units of the Diesel engine.

These plumbing problems may be solved in the same way that aircraft gasoline-plumbing problems have been and are being solved. Exception is taken to the statement that aircraft power-plant plumbing has not reached the state of perfection comparable with the perfection of other aircraft units comprising a ship. It is true that many ships in service are deficient in structural detail (plumbing included), but performance records indicate that during the past several years there has been a most marked and indisputable improvement in aircraft plumbing, quite comparable to improvements in other aircraft structural units.

The one great problem confronting the aircraft industry today is reliability and safety. Safety exceeds cruising range, carrying capacity, comfort, economy, or any other factor. Complete public confidence awaits safety, and is to be acquired only from demonstrated safety. Absolutely dependable power plants that nothing can stop but the pilot is the basic problem of the industry. The place of the high-speed Diesel engine for aircraft service rests upon proof of inherent reliability characterized in air service rather than upon economy. It is true that there is a popular opinion that the Diesel engine is far more reliable than the gasoline engine, but has this been satisfactorily demonstrated for the high-speed light-weight engine? Stable performance of the industrial Diesel engine of heavy construction in uniform temperatures, protected from the weather, with constant attendance may not be considered proof of superior reliability in the air.

The carburetted gasoline engine has been established upon a very elaborate production schedule. Service is established throughout the world, and unless the high-speed Diesel engines have unusual stability and reliability, it seems questionable if a change from present practice will be warranted purely on the grounds of improvement in economy and cruising range. The Diesel engine, no doubt, requires far higher standards of production than the carburetted engine. Fuel injection and pump parts must be machined with an accuracy seldom required in commercial production. Although great improvement in performance may be anticipated, it is quite possible that limitations are inherent, and that such perfection in fuel injection and distribution, as anticipated, may be unattainable just as in the gasoline engine inherent characteristics have prevented attainments anticipated years ago.

The Diesel engine is, no doubt, more adaptable to the construction of large units in comparison to the air-cooled radial engine. With our present knowledge of propeller design, and with available materials, it seems that propeller sizes and tip speeds have reached a maximum for efficient performance. New developments in propeller construction and design are, no doubt, required for the efficient absorption of 2000 hp. in a single propeller.

If huge slow-moving propellers are to be used with power units of this capacity, gear reduction of some kind will be required, with a corresponding decrease in efficiency. It may be that the future will develop means for transmitting power from a single large

power-plant unit into several propellers disposed about the ship. In view of recent developments in hydraulic power transmission, it may be that hydraulic or other fluid power-transmission mechanisms may serve this purpose, but some means for distribution of this large power output from a single unit will be necessary.

The authors have made no reference to such motor characteristics as starting quality, acceleration, pickup, effect of temperature and climatic condition on performance, and similar characteristics which affect the performance of a ship. Military ships require the best of acceleration or pickup qualities. It is true that for the most part flying is accomplished with comparatively uniform motor speed, but as air lanes become crowded, pickup and acceleration will be most important motor characteristics.

Judging from the fuel curve shown by the authors the imposition of overload on the motor brings a sharp rise in fuel consumption with soot and smoke development, so that it is logical to assume that, at least for the time being, no appreciable output above the rated capacity should be counted upon for high-speed Diesel engines, as operation under overload conditions for any considerable length of time or at frequent intervals would probably create deposits in sufficient quantities to be detrimental to fuel injection and require frequent overhauling and servicing.

The decreased fire risk claimed for the Diesel engine seems to be disputable. The text indicates that this advantage is of a limited extent, and that Diesel fuel apparently has at least a tendency to ignite from a hot surface more readily than gasoline, as proved by laboratory experiment. Inasmuch as it is and probably will be compulsory to equip ships with suitable fire-extinguishing facilities, no doubt the slight advantage accredited to the Diesel type of engine will not be of material consequence as fire-extinguisher equipment will still be necessary. As stated, Diesel fuel is unquestionably less cleanly than gasoline, and due to the fact that it will not evaporate as readily as gasoline, it may be in some instances a greater fire risk than gasoline, so at least the two types of engines seem to be about of a parity in this respect. In view of progress in the efficient shielding of high-tension ignition in the system from radio interference, it is quite possible that an entirely efficient and satisfactory shield equipment may be established in the course of further development of the gasoline engine inasmuch as such shielding equipment is quite as new as the high-speed Diesel engine, and it may be that the development of such equipment will provide valuable protection for the ignition system, and contribute to more dependability rather than to detract from its efficiency.

Two-cycle high-speed engines are a problem of the future in both carburetted and Diesel style. The Diesel cycle offers the advantage of intake of pure air as compared with a fuel-air mixture. Scavenging pumps are, however, a requisite of both carburetted and Diesel two-cycle engines. Superchargers as developed to date can hardly be termed dependable for two-cycle high-speed service capacities. The question of ample and proper scavenging of two-cycle engines will be the problem for designers to solve before success is assured. With proper scavenging equipment available, engines of intermediate type may develop; that is, a two-cycle engine burning a volatile fuel such as gasoline with injection of fuel over a wide crank angle with low maximum pressures, and with electrical ignition maintained for combustion. In such an engine fuel distribution throughout the charge could be obtained readily with low-pressure fuel-injection piping, comparatively simple injection nozzle, rugged and inexpensive lightweight fuel pump, and with performance comparable with two-cycle high-pressure Diesel performance in the same ratio as the four-cycle engines now compare.

In the advantages claimed for cruising range, consideration of the 10 per cent greater weight of the Diesel engines seems to have been ignored. This 10 per cent additional weight for a large unit (fuel costing 21 cents per mile) will approximate the weight of one passenger, which would, in turn, offset the double fare gain, due to fuel economy as cited in the text.

Is not the advantage of working on an engine in flight overestimated? Particularly if dependence is placed entirely upon a single prime-mover. So little can be accomplished in the way of servicing a running engine in flight that it is highly improbable that any but the most minor adjustment could be accomplished unless multi-power plants are so disposed as to permit quite extensive shutdowns during flight. Claims for Diesel aircraft engine superiority are well taken, but performance data to date are too meager to warrant final confirmation of such superiority.

HAROLD C. GERRISH.[6] It is gratifying to note the interest that is being shown in the development of the high-speed oil engine for aircraft use. As pointed out by the authors, the "dual cycle," a combination of both the Otto and Diesel cycle, is the cycle commonly used as the operating cycle for high-speed oil engines. It not only reduces the theoretical weight per horsepower and increases the reliability in comparison with the Otto cycle for the same power output, but is especially adapted to the oil-injection engine.

Several valuable contributions on the ideal cycle efficiency of internal-combustion engines made during the past ten years have not been brought out definitely by the authors, as follows: T. B. Morley (1)[7] developed an approximate chart for both constant-volume and constant-pressure combustion cycles and found in many cases "that the chart gives results for efficiency and mean effective pressure within three per cent of the correct values;" R. W. Bailey (2) derived algebraic relations and presented graphs and tables necessary for the graphical and arithmetical method of solution; W. J. Walker (3) obtained a general expression for the efficiency of the "dual cycle," a combination of constant-volume and constant-pressure combustion; G. A. Goodenough and J. B. Baker (4) analyzed two leading cycles of internal-combustion engines, giving the ideal efficiencies under various conditions, while using several fuels; F. O. Ellenwood, F. C. Evans, and C. T. Chwang (5) considered the working fluid to be actual cylinder gases, and gave results of calculations for ideal efficiencies for the Otto and Diesel cycles in the form of curves and tables; A. E. Hershey (6), using the method developed by G. A. Goodenough for calculating internal-combustion-engine efficiencies and temperatures, showed the effect of compression ratio on the maximum temperature of combustion, the temperature of exhaust for the Otto and Diesel cycles, and the thermal efficiency for various amounts of excess air.

Two of the main factors which must be decided upon in the design of high-speed oil-injection engines suitable for aircraft are compression ratio and maximum cylinder pressure. The effect of these factors on the performance of an oil engine is impossible to obtain accurately by actual experiment, because of the influence of several factors difficult to control, such as penetration and atomization of the fuel spray, chemical composition of the mixture, and the rate of combustion.

The National Advisory Committee for Aeronautics, Langley Field, Va., has therefore, during the past two years, conducted an investigation to determine the effect of compression ratio, maximum cylinder pressure, and excess air on the cycle efficiency, indicated mean effective pressure, and fuel consumption of oil-injection engines working on the "dual cycle." This investigation covered a range of compression ratios from 8 to 18, maximum cylinder pressures from 700 to 1200 lb. per sq. in., and excess air from 0 to 244 per cent. The working fluid was considered to

[6] Assistant Mechanical Engineer, National Advisory Committee for Aeronautics, Langley Field, Va.

[7] Numbers refer to bibliography at end of discussion.

be actual cylinder gases, the mixture to vary at different points on the "dual cycle," the specific heats of the gases to vary with the temperature involved and the complete combustion of the fuel. A complete report of this work is soon to be published and will include a sample calculation of the method employed.

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W. L. Dempsey.[8] The authors have given a complete picture of the development of the aeronautical Diesel engine up to date. The state of the art has been clearly recorded and will be of historical value in the future. However, the term "aeronautical Diesel engine" has not been defined. It is assumed that it means an engine adapted to meet the requirements of aviation and has about the following characteristics:

(1) An engine capable of burning the heavier hydrocarbons, ranging from 30 deg. Baumé downward.

(2) Since the r.p.m. is the function of the stroke or crankarm, to meet satisfactorily the requirements of an aviation engine, it should have a piston velocity at three-fourths maximum speed of not less than 1500 ft. per min.

(3) It will have the Diesel characteristic of autoignition.

(4) It will have variable speed characteristics over a wide range.

If this be a correct definition of an aeronautical Diesel engine, I fear that it is but another noble and costly experiment, another rosy dream at twilight that becomes a fading fallacy at dawn.

Every natural process under the sun functions on a fixed and immutable time schedule. In spite of learned theses and abstract and subtle mathematics dealing with the relativity of time and space, the physicist is still keenly conscious of the existence of both time and space, and each is infinitely beyond the puny power of man to affect.

The electronic beams of light and energy pouring forth from the sun travel on an exact and unchangeable time schedule, first arriving at Mercury, then Venus, Earth, Neptune, and the infinite wastes beyond the Polar Star, arriving at each with such precisely timed exactness that, if we but knew the curvature of this grand cycle, we could determine the exact second when they would return again to rekindle the fire of a perhaps slowly waning sun. Even at the terrific velocity of 186,000 ft. per sec., we can calculate to the second the return of a comet that departed from our skies centuries ago.

It will be as easy for the genius of man to accelerate or decelerate the flight of a comet through stellar space as it will be for him to change the time schedule for the burning of the molecules of the homologous petroleum series. Methane (CH_4) will always burn in a shorter length of time than pentane (C_5H_{12}); therefore, if only the heavier molecules are to be utilized as fuel, the piston velocity will inevitably be determined by the size and boiling point of the molecules of the fuel used. Piston velocity is necessarily controlled by the time schedule of the burning fuel, and this fixed and immutable law of nature stands between the high-speed aeronautical Diesel engine and success. The Diesel engine is of incalculable economic value, but it requires a firm foundation on solid earth rather than the unstable foundation of the upper air.

In ballistics muzzle velocity is strictly dependent upon the time schedule of the explosive, and in gas engineering piston velocity is a function of the fuel used. The combined efforts of science, engineering, and genius will never be able to build an incubator that will hatch out a goose egg in the same length of time that it will hatch out a chicken egg. It is axiomatic that large bodies move slowly, and it is equally axiomatic that large molecules of fuel burn slowly. The highest efficiency in steam power plants has been reached by using powdered coal. The humblest housewife knows that she cannot cook small potatoes and large potatoes in the same length of time at the same temperature, hence she halves and quarters them to uniform size.

This requires an expenditure of energy and also an expenditure of time. This in effect is exactly what our refineries are doing in cracking processes. The refineries are efficient because of mass production and the availability of cheap fuel for generating power to reduce the molecules of the fuel to approximately uniform size. Why should this process be changed from mass production to individual treatment, from the solid earth to the hazy atmosphere? It would seem wise that aviation be supplied with a uniform, finely pulverized fuel so that maximum power would always be available without using a portion of the energy developed for preparing the fuel in flight.

Another factor opposing high piston velocity in the Diesel engine is the manner of introducing the fuel, which greatly lengthens the time required for diffusion of the fuel in the almost impenetrable density of the compressed air within the cylinder. Present practice utilizes pressures of from 600 to 1200 lb. per sq. in. to drive a fine pencil of fuel into the combustion chamber. The force required would easily be sufficient to drive a tenpenny nail into a seasoned oak beam.

It must be remembered that the force applied is applied in practically a straight line and that gases within a closed vessel exerted equal pressure in all directions, that there are no areas within the cylinder where the air is of unequal density, and that there are no dynamic forces within the cylinder capable of creating diverse currents. Air under a pressure of 500 lb. per sq. in. becomes a very solid mass, and a thin stream of fuel injected into it under a pressure of from 600 to 1200 lb. per sq. in. forces the corpuscles of the fuel so close together that "no air can come between them," hence only the peripheral surface of the stream of fuel can come into contact with the oxygen of the air, so that instead of the fuel mixing freely with the air it is literally eroded away, and this process of diffusion is necessarily slow.

There can be no turbulence within the cylinder under these conditions to scatter and diffuse the fuel because turbulence is the result of kinetic energy applied; therefore, until the temperature rise due to compression and the injection of the fuel is sufficiently high to ionize the gases, distribution of the fuel cannot take place. This, in my opinion, will forever prohibit high piston velocity in an engine using low-grade fuel.

It has been suggested in this discussion that the building of a high-speed Diesel engine should not be considered doubtful and that the tremendous amount of research and experimentation in this field will ultimately succeed. The speaker used a very pretty, but to me a very inapt, simile when he said all that is necessary to do is to bring into close proximity the atoms of oxygen, hydrogen, and carbon and, like a lovelorn boy and girl when drawn together, a marriage was the inevitable result. It is thought that the simile did not go quite far enough.

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It is well known to scientists that there are very few philandering bachelors among molecules, that they are all polygamous and polyandrous, and that they are as gregarious as sheep; and both civil law and natural law require that before a marriage can take place the contracting parties must be single and legally competent; so in the act of combustion molecules do not unite with each other to form new compounds, but the molecules must be disintegrated, the atoms must be divorced and made free to seek other conjugal felicities.

Again, the simile falls down in that the union of two atoms of hydrogen and one atom of oxygen does not release energy; in fact, energy is absorbed by the union, and when once two atoms of hydrogen and one atom of oxygen become united, their honeymoon beginning perhaps in the cylinder of an engine, they float out through the exhaust and drift away arm in arm to the upper air and wander forever through ethereal space until some vagrant lightning shaft dissociates them. At no time during the union is a single dyne of energy released.

Sir J. J. Thomson, Dr. Millikan, Dr. Bohr, Lewis, Langmuir, and others have experimentally demonstrated that every individual atom is a storage battery composed of a positively charged proton and a negatively charged electron. Nature has exactly balanced these forces, and the only energy that can be unleashed in the cylinder of a gas engine is these two electrical charges, and this energy cannot be made available until some external force disrupts and destroys the perfect balance between these two charges, whether it be the energy of high compression or the jumping of an electric spark.

The molecules within the cylinder, composed of millions of atoms joined together by the resultant forces of the electrical charges in the atoms, would forever remain in a quiescent state, like a nest of hibernating ants or a hive of bees, until disturbed by an external force. It is then and only then that ionization takes place, that the bees begin to swarm forth, and the ants suddenly awaken into active energy; so the released energy of the atoms impels the piston on its working stroke.

In other words, combustion takes place in two phases or stages: the first stage is that of disintegration or ionization of molecules and atoms; the second stage is that of integration or the forming of new compounds. The first is a stage in which the stored energy of the molecular and atomic batteries is being released, when millions of negative and positive charges, like angry hornets, bombard the piston head to impel it on its working stroke, and it is only in this first stage of combustion that the dynamic energy in the fuel is transferred to the moving piston.

The second stage is a mating period when the atoms are being joined into harmonious unions, thereby reestablishing perfect static and dynamic balance between the positive and negative charges inherent in the atom.

In order to release the maximum potentiality of the fuel, not only must there be sufficient oxygen in the cylinder to supply one atom of oxygen for every two atoms of hydrogen and two atoms of oxygen for every atom of carbon, but in addition to the atoms being in sufficiently close proximity to each other to bring them within attractive range of the electrical charges, there must exist sufficient temperature to disintegrate or ionize the largest molecule present.

For these reasons I do not believe that the paper is prophetic of the success of a high-speed aeronautical Diesel engine.

Authors' Closure

The extensive discussion accorded the paper is most gratifying as indicating the general interest in the subject. The many points raised cannot be treated as completely as their importance warrants, but we would like to reply to some of the more controversial features in the interest of clarification.

Mr. Alden cites the tendency toward decreased rotational speed with increase of cylinder diameters, but we believe that this will not be a serious handicap to the large engine, especially with opposed-piston two-stroke engines. We may assume that piston speeds will remain about constant, regardless of stroke. Whether the scavenge function in two-stroke engines will prove to be the speed-limiting factor is debatable. It is of interest in this connection to study the favorable results of the analysis presented by Dr. Otto Holm in *Motorwagen* for Jan. 20 and 31, 1928. The simplification of injection systems as mentioned in the paper will eliminate adjustments of excessive accuracy, and the injection mechanism will stand more abuse than the ignition system of the gasoline engine. We are glad to note the opinion of one so experienced in fuel-pump and injection-valve construction to the effect that the manufacturing problems involved are not serious.

Mr. Scott has done much work with the two-stroke fuel-injection engine, so that his favorable comments are authoritative. He evidently anticipates no serious difficulties in the fuel-injection system. The high scavenging efficiency stated for the Junkers opposed-piston two-stroke engine shows that this operation can be performed at least as well in the two-stroke as in the four-stroke engine.

Mr. Parker errs in assuming that we consider the carburetted engine to have reached the limit of its development. Our comparisons were based on results already achieved in both types of engines, but it must be remembered that the potential development possibilities are all in favor of the Diesel. We agree that fuels of special characteristics may give more power, lower consumption, or better operation that the ordinary good quality of industrial Diesel fuel oils. Neither the literature of the subject nor our own work, however, indicates that much is to be expected in that direction.

It is possible that the variable-pitch propeller will prove more desirable for the Diesel engine, but at the same time we invite attention to the fact that the value of this device to the present carburetted engines also has been definitely proved, and its adoption, at least in the larger powers, merely awaits the development of a satisfactory construction. However, it must not be assumed that the Diesel is less flexible than the carburetted engine, for the superior acceleration of the Diesel is already considered one of its strongest points for automotive applications. Its torque is much better maintained than that of the gasoline engine at low speeds. The Diesel will not idle as slowly, though the difference is small with properly regulated injection.

The problem of control of injection at various speeds is nonexistent. It is a much simpler operation than the control of a gasoline-air mixture to the very close limits that are necessary. Change of injection pressure does adversely affect spray formation, but the reduction of spraying efficiency at reduced loads is more than offset by the increased combustion efficiency due to the additional air available.

As stated in the paper, the maintenance difficulties are difficult to evaluate. The evidence at hand leads us to expect that they will not be serious. Certainly, one cannot object to giving the injection pumps and nozzles the same amount of attention as is now required by magnetos and spark plugs. Operating reliability is undoubtedly of prime importance. The Diesel engine has in this respect a great advantage, as the injection system may be composed of a number of complete and entirely separate units, the failure of any one of which affects only the one cylinder served thereby. An eight-cylinder engine will have a total of from 40 to 60 holes for the injection of fuel, so that the clogging of one or two holes has but little effect upon the engine performance, particularly when the overload capacity and the flat shape of the fuel curve at the usual operating load are considered. The partial choke of a carburetor jet immediately affects all

cylinders served thereby, and in addition weakens the mixture ratio with resultant danger of backfire. Very careful filtration of fuel is desirable in both types of engine.

Fuel lines must be heavy, of course, but Mr. Parker's statement that they must withstand "backfire" is based on an incorrect understanding of Diesel functioning. Contrary to general belief, air bubbles in the line or nozzle of the modern injection system have no bad effect. In fact, air is so readily dissolved in the fuel at high pressures that it is impossible to use an air chamber as an accumulator for the purpose of damping out pressure fluctuations in common rail systems. No difficulty has been experienced in starting oil engines with the system empty and without priming of pumps.

Peak pressures in some types of fuel-injection systems for aircraft engines are likely to approximate 10,000 lb. per sq. in., but no difficulty is experienced in holding these pressures. It must be remembered that the lines are of small diameter and short length. The fuel system up to the primary pump is the same as on the carburetted engine, but the greater viscosity of fuel oil reduces the comparable leakage, and the high flash point greatly minimizes the danger from such leakage.

The form of propeller system to be used with very large engines is, of course, as yet undetermined, but it appears that single suitably geared direct-mounted propellers will be applicable to engines up to, say, 3000 hp. A four-blade propeller 18 ft. in diameter will efficiently absorb 2000 hp. when turning 900 r.p.m. at an air speed of 140 m.p.h. Such a propeller is quite reasonable for the large airplanes which will use engine units of this power.

With regard to comparative starting ability, acceleration, and effect of atmospheric temperature, it is well known that the Diesel will take full load immediately after starting, that its acceleration is much superior to that of the carburetted engine, and that atmospheric temperature changes have no effect other than to vary the maximum power output proportional to the change of density. The overload capacity of the Diesel is an important advantage, especially in automotive applications, since such overloads are ordinarily required for only very short periods, during which the increased fuel-consumption rate and darkened exhaust are not objectionable.

The decreased fire risk is doubtless most valuable immediately after a crash, when the fire-extinguishing equipment has probably been put out of commission. We doubt whether Mr. Parker would in such case be as happy if the tanks were filled with aviation gasoline instead of high-flash fuel oil. The two-stroke gasoline-injection engine mentioned has some interesting possibilities, especially for military use, but cannot meet the safety and economy requirements of commercial aircraft. It is being investigated by the N.A.C.A. laboratory.

Mr. Parker has misunderstood the reference to the Diesel engines on the Canadian National Railways. These sets can be considered as in the high-speed class, since they operate at speeds up to 1000 r.p.m. with a 12-in. stroke, or a piston speed of 2000 ft. per min. The gasoline engine does not perform equally well without attention for 250 hours.

If Mr. Parker will refer to the paper, he will see that we estimated the specific weight of the two-stroke aeronautical Diesel at 20 per cent *less* than that of the carburetted engine, not 10 per cent *more*. The Diesel overload capacity further increases this advantage. The advantage of working on an engine during flight is not now generally appreciated, mainly because present installations of air-cooled engines almost prohibit such attendance. It is worth noting, however, that the present airplane endurance record was made possible by reason of special provisions for servicing during flight; spark plugs being changed, rocker arms lubricated, etc. Also, nearly all airship engines are regularly serviced during flight, with or without stopping them.

As to fouling of the combustion chamber, the Diesel will continuously carry an overload of 10 to 15 per cent without smoke in the exhaust or accumulation of carbon in the cylinder. One may say that the Diesel rejects its carbon, while the gasoline engine accumulates it inside the cylinder. Decarbonization is determined by the life of the valves, rather than by detonation or preignition from carbon as in the gasoline engine. Incidentally, the exhaust-valve temperature is much lower in the Diesel.

Mr. Dempsey lists four characteristics which he thinks the aeronautical Diesel should possess. These are correct in the main, but it is quite possible that an oil lighter than 30 deg. Baumé will prove best for such engines and service. There is no more reason to require an aeronautical Diesel to run on the cheapest fuel oil available than that the carburetted aeronautical engine should run on cheap commercial gasoline.

We are in doubt whether Mr. Dempsey expects the remainder of his remarks to be taken seriously. Perhaps the best reply is to direct attention to the fact that all of the requirements for the aeronautical Diesel have been separately met, so that the remaining problem is to secure the same results in a single engine mechanism. Satisfactory b.m.e.p. and economy have been obtained at 3000 r.p.m. and piston speeds over 2000 ft. per min., so that no present limitations in this direction are anticipated.

Mr. Gerrish draws attention to several valuable items in the literature of the subject, a perusal of which will be of interest to any one seriously studying the development of the high-specific-performance Diesel. We take pleasure in drawing attention to the comprehensive research being conducted into all phases of the general problem by the National Advisory Committee for Aeronautics at its Langley Field laboratory, and which work is not as well known or utilized as its importance warrants. This Government organization is conducting its investigations with a thoroughness far beyond the possibilities of any industrial organization in this field.

Since our paper was written some months ago, the published performances of the Junkers aeronautical Diesel and the unpublished results secured by several organizations working on the problem have furnished a basis of definite accomplishment which substantiate our statements regarding the possibilities of the light-weight high-speed Diesel. News reports state that recent Junkers airplanes are designed for Diesel engines and sold with the understanding that such engines will shortly be supplied to replace the original equipment of gasoline engines.

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Factors in the Design of Commercial Airplanes

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A steady and profitable market may be expected for existing equipment in remote and inaccessible places, where other forms of transportation are not available, but the real problem is to make the air transport compete successfully with the extra-fare trains. The real commodity that is being sold is transportation, and the factors involved are speed, comfort, and cost, assuming of course that safety and reliability have been demonstrated. The combination of these three factors must equal that of the fast train. The cost of air travel is of necessity high; therefore the speed-comfort combination must exceed that of the railroads. In general, the comfort of the train at least equals that of the airplane, so the latter must be sold on the speed factor alone, without sacrifice of a certain minimum requirement for comfort and at a cost that is justified by the increase in speed.

Existing air transports have a high speed of 135 to 145 miles per hour, a capacity of ten to twelve passengers, and a power output of 100 hp. per passenger. The speed has been obtained by increasing the power per passenger to nearly double that formerly thought sufficient. High power means high fuel capacity and decrease in percentage of pay load.

The author advocates the development of a 30 to 40 passenger airplane that will equal present speed requirements with a reduction of power to 50 to 60 hp. per passenger. To obtain this result he advocates abandoning direct-drive air-cooled radial engines and replacing them by geared engines, air-cooled or liquid-cooled, placed preferably inside the wing, with a remote drive to the propellers. The airplane itself should approach the flying wing, the wing loading should be high, and the landing gear should be retractable. Due to increase in landing speed the brakes should be much larger, should be operated by mechanical power, and should be capable of utilizing the full friction between the wheels and ground. The material from which the airplane is made will depend largely on a depreciation factor which is not yet clearly defined and cannot be set up until large airplanes have been in operation for a longer period of time.

THE history of American aviation during the last ten years has been as romantic as it has been remarkable, and the men in charge of the technical development find themselves in a most extraordinary position. Plodding along during the lean years, between 1918 and 1927, when their sole support came from the military authorities, the technical leaders of the industry find themselves suddenly supported by great financial resources and told to go forth and earn returns on large sums of money. This situation has its difficulties as well as its elements of strength. Probably no industry in its infancy has been in a stronger financial position to take advantage of whatever development the future may hold in store, but at the same time there is every reason to believe that the financial structure is stronger than the technical foundation. Hence a heavy responsibility rests on the aeronautical engineers to justify the confidence of the public in their chosen profession. The time has come to take careful stock of the situation and to control the direction of development in order that the resultant product may fill a definite economic need and produce a profit for both manufacturer and user.

The title of this paper was chosen to limit the discussion to large airplanes used for transportation. The term commercial airplane has in the past referred to any airplane that was not solely military in character, and this fact is a tribute to the extent to which the industry has in the past depended on the military services. In future this term should carry the same significance that the term commercial automobile conveys, and should be limited to those airplanes that are used in supplying transportation or some other business commodity.

The large commercial airplane really supplies transportation and is of value only when the commodity thus supplied can be marketed at a lower price for service rendered than similar service offered by the older and more established methods. This leads to an examination of the factors entering into the service rendered and their relative importance. These factors are speed, cost, safety, and comfort—and, most important of all, existing competition. This latter factor sets the standard that must be met or bettered to make operation an economic possibility. The author has designed airplanes for operation over the Andes and in the Peruvian hinterland where the only competition was small river steamers, connecting with Indian dugouts and mule pack trains. Although the airplanes were single-engined open-cockpit machines with a cruising speed of 85 to 90 m.p.h., the service rendered was faster, cheaper, safer, and much more comfortable than anything the competitor could supply, and hence was bound to be a success. The same condition will apply where the competition is supplied by small vessels making infrequent sailings. In fact, a steadily increasing and profitable market for air transportation, using equipment as now developed, may be expected from the more remote and inaccessible portions of the world where quick and cheap boat and rail transportation has not yet been provided. However, this type of market will not supply the demand that the industry needs, and the aeronautical engineer must produce a product that can equal the speed, cost, and comfort combination of the extra-fare train, with its dining, sleeping, and club-car accommodations.

Unfortunately, these three factors are mutually antagonistic. At present the airplane cannot compete either in price or in comfort with the extra-fare train; hence, its only reason for being is speed. Therefore, this factor must be improved upon and given priority over every other consideration. Cost, however, increases very rapidly with speed and also with comfort and may become prohibitive if both these other factors are exaggerated. However, it is probably safer for the designer to assume that, if a fast and reasonably comfortable service is rendered, the public will pay the bill.

The hardest problem for the designer is to get the proper relation between comfort and speed, assuming that the cost factor remains constant. Obviously, the maximum speed can be obtained by crowding a given number of passengers into as small a space as possible and having them "stay put," without giving them the various little accessories that contribute to their enjoyment. However, the class of passengers that can afford the high cost of air transportation is one that can afford and will demand all the conveniences of life, and will look askance at a journey, no

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[1] Keystone Aircraft Corporation. Mem. A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

matter how swift, if it embodies any real bodily discomfort. Therefore, there is a certain minimum of comfort, and hence size and weight, which must be supplied, and the designer must start with this requirement and at the same time maintain his high speed.

It is generally claimed by air-transport operators that a cruising speed of 110 to 125 m.p.h. is necessary for successful competition with railroads. The term cruising speed is elastic and depends upon the revolutions per minute of the engines at which the pilot chooses to operate; hence, it is better to set up a criterion based on maximum speed. This figure may be set for the present at 145 to 155 m.p.h.

Having set up a high speed that must be equaled or bettered, the designer thus turns his attention to the size of body that must be provided. This will depend on the length of journey, but in general it may be assumed that flights of three to four hours will be the shortest in duration for which airplane use is justified. Each passenger will want a comfortably upholstered seat at least 20 inches wide, which should be adjustable at will. At the same time, he will require a certain freedom of movement, such as is furnished by the railroad train. The relief afforded by getting up and walking around and, in general, changing location is so great that some eminent authorities believe that in future we shall have 40-passenger airplanes carrying a maximum of 20 people.

This requirement of mobility of action points to the development of large airplanes with a total capacity of 30 to 40 passengers. Several developments of airplanes of this type are now under way, and the author believes that this size probably represents the economic limit at the present time.

Before, however, going too far with the determination of the proper passenger capacity, considerable thought must be given to the relation between size and cost of operation per passenger mile. The direct cost of fuel, oil, salaries, maintenance, and overhead is easily obtainable, but the proper factor for depreciation, due to normal wear and tear, accident, and obsolescence, is not known with any degree of certainty and cannot be known until large transports have been in operation for some years. Obviously an airplane of large capacity can be operated at less cost per passenger mile than a small one, but when depreciation on capital cost is set up, will the resultant cost be a minimum for the very large airplane or for one of intermediate size? So important is this relatively unknown depreciation factor that it may control both the size of airplane and the materials of which it is made.

Let us look briefly at the large transports now in operation and take stock of the situation, and then play the dangerous game of endeavoring to predict the line along which future developments will follow.

The present type of big transport is almost standardized as a high wing monoplane, employing three air-cooled engines, one in the nose and two below the wing. This type has been a very logical development. Three engines are the minimum number on which satisfactory flight can be maintained with one engine out; hence, this number will probably remain the minimum for subdivision of power required. There is now, however, a campaign being launched to do away with the nose engine in order to reduce fuselage vibration. To accomplish this the designer either will place his three engines entirely above the wing, or he can raise his wing above the body and mount his engines in the leading edge, or he can revert to a biplane with underslung body and the engines between the wings. The more logical development will be the use of four engines in pairs of tandems, with corresponding increase in size of airplanes.

The present transports have a high speed of 135 to 145 m.p.h., with cruising speed about 110 m.p.h. This is probably a little low for transport requirements, but even this speed had been obtained only by the use of 100 hp. per passenger, a figure that the author believes is too high for the result obtained. The ratio of useful load to gross load that may reasonably be expected with high-powered ships is 33 to 40 per cent. Out of this must come the weight of fuel and oil, and of course the more power per passenger the more of the useful load will be absorbed by the weight of fuel. The old rule for passenger ships was 50 hp. per passenger. Modern ships have doubled the power, and for this increase have given more speed, more reliability (three engines), and more comfort. The present cabins have two rows of comfortable seats and sufficient aisle space to give a reasonable amount of comfort and freedom of motion, but there is need of additional space for rest rooms or smoking rooms. This latter room should be so located and ventilated that the smell of smoke will not annoy the other passengers, especially those on the verge of airsickness.

The wing loading of modern transports, 14 to 16 lb. per sq. ft., is all that can be used with the present Department of Commerce ruling of a landing speed of 60 m.p.h. This rule, the author believes, is open to very serious objection. In an admirable paper entitled, "The Economic Relationship of Speed and Weight in Air Transportation," Mr. E. P. Warner evolves a most interesting relationship between economical cruising speed and landing speed. His conclusions are that cruising speed should not exceed 1.8 times the landing speed and that the cost of transportation at any given speed decreases rapidly as the wing loading is increased. This of course substantiates Dr. Rohrbach's theories, expounded many years ago, that in high wing loading lies the solution of the very large airplane.

In reviewing existing airplanes from the standpoint of materials used, a great variation of thought is at once apparent. There has been much talk of the advantages of all-metal construction. This is as ridiculous as to talk of all-metal houses or all-metal rowboats as having some fundamental advantage over wood. In building houses, as in shipbuilding, there is some economic size limit where metal has universally superseded wood, and this also may apply to airplanes, but the engineer should select his materials for each case for the best combination of safety, durability, cost, and performance; and while wood generally has been abandoned for body construction for reasons of safety and cost, the case of wood for wing beam construction is still very strong, and in any new project the use of wood should be investigated before definite decision is made to use other materials.

Present transports have a variation in capital cost of from $4000 to $6000 per passenger. Present indications are that no very great reduction in these figures is likely as the size is increased. It is well established that the larger the airplane the greater the attention required to detail to keep the established pay-load ratio, and hence an increasing cost. The large ship must justify itself on its apparent reduced operation costs, apparent because of the unknown depreciation factor.

Two new transports with capacity of 18 passengers have been constructed and show great promise, although they have not yet been in service long enough to prove whether they can increase the economy of operation. One of them designed under the author's direction is a high wing, externally braced monoplane of 16,000 lb. gross weight, using three 525-hp. radial air-cooled engines. The wing loading is 17 lb. per sq. ft. This ship has an excellent performance with a power loading of 87 hp. per passenger, and this should be a strong candidate for supremacy in the intermediate field. The other development is a remarkable piece of engineering work. The ship is a large biplane lightly loaded and using two 620-hp. geared water-cooled engines, which give a horsepower per passenger of 67. In fact, this airplane, compared with the older types, gives an equal performance with the same power and an increase in pay load of 50 per cent. It is open to the objection which may be applied to all twin-motored ships, that with one engine dead a satisfactory performance on the

remaining engine at altitude is difficult, if not impossible, and also the objection that the lower wing impairs the vision of the passengers, but otherwise the airplane marks a real advance in aeronautical design. One significant feature is the entrance of the liquid-cooled engine into the large transport field. This point is one of great interest and may mark a turning point in our ideas of engine-propeller efficiency.

Looking ahead, the author believes in the development of the 40-passenger airplane, but believes that present efficiencies must be improved and high performance obtained with an expenditure of 50 to 60 hp. per passenger. The first and most obvious step is to "clean up" the airplane and to increase the present propulsive efficiencies of the power plants. Present practice of stringing a number of direct-drive radial engines along the wing must be abandoned, or else the cowling must be much improved. The use of geared radials in this country has been very slow, but this change is a partial remedy at best. If the engines are to be hung in the relative wind, it would seem natural to expect the use of in-line air-cooled or the water-cooled type with Prestone cooling. The most elegant solution of the engine problem is to house all the power plants in the wing itself and to have them geared in groups to shafts which have large propellers located at some distance from the engine room. The similarity to marine practice is evident. While the development of clutches, gearing, and shafting of light weight will present quite a problem, the advantages are sufficient to make this a most promising direction for development.

Next in line will come the increase in landing speed, and at the same time the development of very effective power-operated brakes. The pilot should have at his control sufficient braking force to develop the full friction between wheels and ground at all times. This braking force with improved long-stroke landing gears and possibly a spoiler system to kill the lift of the wing should make landing at 80 m.p.h. a safe operation on any prepared landing field. The probability of forced landings should be greatly reduced by multiplicity of power plants.

Another line of endeavor should be the working out of a satisfactory retractable landing gear on large airplanes. The resistance of this unit is out of all proportion to its size and weight, and it is time it was removed from the relative wind.

Space requirements for rest rooms and sleeping accommodations seem to indicate the building of bodies with larger cross-section area, and this points to that dream of the airplane designer, the flying wing with everything inside except the propellers. This type of design, or a modification of it that improves the visibility of both pilots and passengers, seems to be a goal toward which to aim and would seem to offer the best speed-comfort combination.

A word should be said about large flying boats. The problem here is rather different. The competition moves at slower speed, and hence a reduction in top speed may be allowed without losing relative advantage as compared with the land transport. This is fortunate, as the flying boat may have to land in the open sea, and hence a lower limit on landing speed is advisable. This idea is contrary to modern European practice, where the highest wing loadings have been used on flying boats, but to the author the probability of landing a large boat in the open sea seems much greater than the necessity of putting an airplane down on an unprepared field.

After landing in rough water a flying boat should have reasonable seaworthiness and have a strong wing-tip float anchorage. This makes a strong argument for the biplane, where large area and rugged construction can be obtained for less weight than in the corresponding monoplane.

To summarize the foregoing remarks, the airplane designer, to be successful in competition with the extra-fare trains, must supply a combination of speed and comfort that will be sufficiently more appealing to the general public to justify the higher cost of air transport. In spite of the phenomenal advances of the last two years, air transport is not yet sold to the people at large. With increase in number of landing fields and improvement in the efficiency of the airplanes, there is no doubt that this sale will be consummated and that air transportation will play a prominent part in the economic life of the country.

Discussion

ARCHIBALD BLACK.[2] Perhaps the most interesting feature of the paper is the frankness with which the author faces the issue and recognizes the basic problems confronting the designer of transport airplanes. Too many designers have, in the past, found it impossible to get into their heads any semblance of appreciation of the importance of airplane-operating costs. I am particularly gratified to note how the author objects to the recent trend toward more power. This development is one of very questionable value. Increase in speed must come by the more laborious method of refining the design. Merely increasing horsepower is a thoroughly inefficient expedient, which only increases operating costs, making it a still greater problem for the traffic salesman. We must squarely face the fact that we are competing with the railroad, and if we are to carry any substantial volume of traffic, we will have to get our costs down closer to those of the railroad. This is possible if we keep down unit operating costs and increase volume of traffic simultaneously. It is indeed refreshing to observe that one designer of airplanes fully appreciates the importance of this.

B. C. BOULTON.[3] This paper, while entirely general in its character, nevertheless emphasizes several fundamental points which the writer believes are most important. First, the horsepower required per passenger in many of our present-day high-speed transports is undoubtedly too great. Although the disposable load for an airplane may amount to 40 per cent of the gross weight, yet only the passengers and baggage or the mail and express are pay load, and this pay load is only 20 to 22 per cent of the gross weight. The fuel and oil form entirely too large an item. In themselves they constitute a weight equal to approximately 65 per cent of the pay load. It is evident that a great increase in efficiency may be obtained through a reduction in the horsepower per passenger. This gain is threefold: first, the fuel and oil bill is decreased; second, the size and, hence, cost of the power-plant units are less; and third, the saving in fuel and engine weight may go directly into pay load; and inasmuch as the pay load is roughly one-fifth of the gross weight, a saving of 5 to 10 per cent in the gross weight due to reduced fuel and decrease in engine size means an increase in pay load of 20 to 40 per cent. The horsepower per passenger can be reduced by aerodynamic refinement, which decreases the resistance, by the use of geared propellers, which increases the efficiency of the power plant, and by the adoption of variable-pitch propellers, when these are perfected, which makes possible a decrease in the excess horsepower necessary to insure safety in case of failure of one of the engines. The variable-pitch propeller permits the remaining engines to continue to operate at full efficiency and also, by feathering the blade of the propeller on the dead engine, decreases its resistance. A further means of reducing the horsepower per passenger is through improved structural design—in other words, by reducing the structural weight per passenger. In this connection, it is quite possible that by employing metal monocoque construction

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to large fuselages, both the resistance and structural weight may be reduced. A further structural refinement that may soon be possible is the use of very high strength steels in thin gages for wing spars and certain other structural parts.

It is also believed that for several years to come it will be uneconomical to design airplanes for a capacity of more than 30 to 40 passengers. There are several sound reasons for this, and all of these reasons may also be used as arguments against the development of giant dirigibles. Until the safety factor is considerably further increased, it would appear unwise to invest so large an amount of capital in a single unit or to assume a responsibility for the lives of too many passengers. Another argument against the very large airplane is the lack of economy in operation due to the problem of partial loads. It is most expensive to operate a giant airplane unless nearly filled to capacity. Furthermore, it is usually highly desirable to increase the number of trips, providing more frequent service, since much of the time gained by using the faster service afforded by an airplane is lost if the passenger has just missed the morning plane and must wait until the one leaving in the afternoon.

In conclusion, it is felt that the author's remarks about flying boats are sound and also that their use, especially when equipped with amphibian gear, is going to increase steadily, particularly along our coasts and great inland lakes. The greater safety afforded by such aircraft, which practically at all times have beneath them a natural landing field, is sure to appeal to the public and will probably outweigh the somewhat lower speed of such airplanes as compared with land types.

AER-52-7

Seaplane Design

By T. P. WRIGHT[1] and G. A. LUBURG,[1] GARDEN CITY, N. Y.

The design of the seaplane is given in detail, and the advantages of its use are described. The seaplane can generally find a landing place nearer to the center of large cities than can the land airplane. The flying boat, it is predicted, will ultimately be the method of air travel across the oceans. The market for the flying boat and the amphibian is waiting for development by the manufacturer and operator.

ALONG the waterways and coastlines of the world are thousands of miles of air routes already prepared, with nearly continuous emergency landing fields, lighthouses, landing docks, and shorelines that afford ample guide signs at all times. These facilities are available for the flying boat and the seaplane.

To this may be added the inland lakes, a large proportion of which affords landing facilities for amphibians.

From a transportation standpoint, for mail, express, and passengers, the advantages of land airplanes are most strongly demonstrated in long-distance operations, where time lost in transportation between the heart of the city and the flying field forms a small proportion of the total. The greater majority of the large cities and towns lie along a waterway. Cities are founded on the shore or bank of the river or lake, and the heart of them always lies close to the waterfront. A seaplane, therefore, has its landing field close to the center of the city, whereas a land plane must take on and discharge cargo and passengers far out where adequate facilities can be available.

From a cost standpoint the free landing field for the seaplane afforded by the water is an item not to be overlooked.

From the passengers' standpoint there is a feeling of greater safety, knowing that there is always a landing field below. Also due to its being unnecessary to fly at high altitudes in order to glide to a landing in an emergency or to cross mountains, the seaplane can be flown at a low enough altitude along the waterway to afford a close-up view of the scenery for the passengers.

Safety being the cardinal factor in air transportation, the advantages of seaplanes over land planes may be briefly listed, as follows:

(1) Existence of practically continuous emergency landing facilities. Most bodies of water have sufficient size for this purpose.
(2) Existence of terminal landing fields of practically unlimited extent.
(3) Presence of already marked routes as line of travel, especially along the seacoast, which is thoroughly lighted by lighthouses and light ships and which has shorelines which afford ample guide signs.
(4) Presence of less hazardous landing fields. A pilot, when landing in a fog, for example, would much rather land in a large bay than try to locate a comparatively small landing field.

History of the Seaplane

The seaplane was born January 25, 1911, when Glenn Curtiss at San Diego flew from and landed on the water in a land plane fitted with a pontoon. It developed quite rapidly, especially during the war, after which, particularly in this country, very little progress was made. Nations having colonies, notably England, to a certain extent foresaw the advantages of the seaplane, especially from a military standpoint, and have developed a rugged, reliable flying-boat type particularly adapted for their needs.

Since the war, airplane development in the United States, except by the Navy, has been practically entirely concerned with land planes. Within the past year, however, some attention has been paid to the design of seaplanes, with very satisfactory results.

It may be of interest to note that what was probably the first commercial seaplane route was operated between Key West and Havana in converted Navy flying boats.

General Design

Seaplanes may be divided into four classes:

(*a*) Single float
(*b*) Double float
(*c*) Flying boat
(*d*) Amphibian.

Classes (*a*) and (*b*) consist essentially of the mounting of a land plane on one or more floats instead of a wheel landing gear.

In class (*c*) the flying boat, the pontoon, and the fuselage are combined, or, as its name implies, it is a boat with wings.

The amphibian is a seaplane either with floats or a hull to which a retractable wheel landing gear is added to permit landing on the ground as well as on the water.

From an aerodynamic standpoint, the seaplane design is worked out exactly the same as a land plane and need not be further discussed. Landing load factors must be increased, due to the higher shock loads that may occur when landing on water with the absence of shock absorbers, and construction must be designed accordingly. Control surfaces must be of a size, shape, and location to take care of the center of thrust, resistance, vertical center of gravity, and yawing moments. These can be closely determined by wind-tunnel tests. Except for the amphibian, which is required to fly over land, the seaplane normally does not require the initial climb or ceiling of a land plane, thereby usually allowing an equal pay load even when giving consideration to the increased construction weight over the land plane. Take-off speed, which will be increased, will have its effect on the ability to get off from the water and must be accounted for in designing the hull.

The new problems lie in the float or hull design for operation on the water and for minimum resistance in the air.

Requirements for float or hull design should include:

(*a*) Seaworthiness at rest or at anchor. This is not for the boat alone, but for the entire machine.
(*b*) Seaworthiness and stability when under way. There must be no danger of nosing under. The tendency to "porpoise" must be kept at a minimum, but at the same time permit longitudinal control by the elevators at least above planing speed. Longitudinal and lateral stability must be inherent until the air controls become effective. Propeller thrust and torque must be taken into account.
(*c*) Low water resistance and absence of suction that would tend to prevent the hull leaving the water.
(*d*) Maneuverability, being the ability to taxi in any direction in a wind.
(*e*) Clean lines that will have a minimum of spray and disturbance of the water.
(*f*) Proper freeboard and excess buoyancy.
(*g*) Aerodynamic cleanness.

The displacement type of boat is unfitted for seaplanes due to

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

its high resistance and to the high suction tending to prevent its leaving the water. The single-step hydroplane has been found to be generally the best type and is most generally used in this country.

The design of pontoons or hulls is based on well-known principles that apply to ordinary planing boats or hydroplanes. It is affected, however, by the additional factors of wing lift varying as the square of the speed and by the high location of the thrust line and center of gravity. The design must provide for the keeping of the angle of incidence fairly low to cut down wing resistance until the machine is up to flying speed, at which point the tail can be dropped to get the necessary angle of attack for take-off.

The single-step hydroplane fills this requirement very nicely. Its main features are a step located a short distance aft of the center of gravity and the bottom aft of the step slanted upward relative to the bottom forward of the step at a minimum angle of about 5 deg.

In take-off the boat is headed into the wind and power is applied. At first the bow is depressed due to the high thrust, but as speed is increased, the bow rises and the stern is depressed. Planing action then commences. The flow at the step starts to jump across to the bottom aft, and air to fill in the space is admitted either at the chines or by vent tubes. As the speed increases the point of contact moves farther and farther back, until, at the point of planing, the bottom aft of the step is entirely out of the water. The planing speed is about one-third take-off speed, and at this point the air controls are starting to function. Inasmuch as the resistance increases with the speed up to planing speed, at which point it drops sharply, assistance should be given by the elevators to level off and get on the step. The peak resistance just before planing is known as the "hump" resistance and is the point past which heavily loaded machines in smooth water and no wind sometimes fail to get. Wing lift has but little effect at the lower speeds. With a 100 per cent lift at take-off speed, the lift would be 25 per cent at half take-off speed and only 1 per cent at 10 per cent take-off speed.

Due to decreased water resistance and increasing lift of the wings, speed increases rapidly from planing up to take-off. Usually the elevators are depressed to keep the machine on the water until well above take-off speed, so that there will be no liability of stalling. By raising the elevator and dropping the tail, the machine will take off with the tail end of the floats or hull remaining clear of the water. It is sometimes advisable to take off at stalling speed when the wind is strong and the sea is rough.

Landing can be made at stalling speed, which is usually done in rough water. The tail of the float touches first. This type of landing requires skill and care as the machine is likely to pancake and the controls are losing their effectiveness. Preferably, the machine is flown in at above stalling speed, so that the stern of the float and the step touch at the same time. The air controls still operate effectively, and the landing is made with a minimum of vertical component of velocity.

Float Design

Let us see how a float design as described in the foregoing will meet design requirements. These points should be as follows:

(*a*) *Seaworthiness at Rest or at Anchor.* The boat is comparatively narrow and long. Flying boats will average a beam length ratio of 5, while twin floats will run from 6 to 8. The proper longitudinal stability is easily attained. Lateral stability must be obtained by separating the twin floats to a sufficient degree or by the use of wing-tip floats in the cases of single floats and single-hull flying boats. Due to the low meticentric height and the high location of the center of gravity, inherent stability is either low or negative. Dornier overcomes the difficulty by the use of short, thick wings extending from the sides of the boat at the water line. These provide necessary buoyancy for lateral stability and clear the water when the boat starts to plane. Rohrbach's flying boats have the wing-tip floats located relatively close in. They are made larger and, in fact, support part of the load. It is really a triple-float system.

(*b*) *Seaworthiness and Stability When Underway.* When applying power there is a tendency to bury the bow. There must be sufficient displacement at the box to prevent this happening. Porpoising has a tendency to occur. This is a tendency for the boat to rock fore and aft at certain speeds. It may be due to the center of gravity coming too close to the step so that the boat balances longitudinally at that point. Moving the step aft a short distance often eliminates the tendency. Both longitudinal and lateral stability should to a great extent be controlled by the surface controls at speeds above planing. At this speed the machine rises so that the entire wing-tip floats and main float back of the step are entirely out of the water. The ailerons take care of lateral stability, and inasmuch as the usual size of machine is being supported on only about 3 to 5 ft. length of water surface, the stabilizer and elevators must provide the necessary stability to longitudinal movements. The angle of incidence of the wings can also be controlled for take-off.

(*c*) *Low Resistance and Absence of Suction.* The desirability of low resistance is self-evident. If the "hump" resistance is greater than the thrust, the machine will not plane and therefore not take off. Curved surfaces will suck down in the water at high speeds. Old-fashioned fan-tail stern boats, when run above a certain speed, are good illustrations. The bow rises and the stern squats nearly level with the water. A flat-bottomed hydroplane with clean-cut steps will plane with a fixed unit pressure as related to the speed, so that when the speed is up to take-off, the unit pressure is zero and there is no suction action.

(*d*) *Maneuverability.* It can readily be seen that a seaplane in a breeze of wind and a seaway is not the easiest thing to handle. When at anchor or taxiing head into the wind, the tail surfaces form a good weathercock. When traveling cross-wind or with the wind. it is a different matter. The single-float seaplane and the flying boat are easiest to maneuver. Twin floats are more difficult. In flying boats where the tail end is above the water and the keel length in the water is comparatively short, a water rudder is sometimes installed and improves maneuverability to a marked degree. The water rudder must either be operated independently of the air rudder or must be interconnected by springs strong enough to carry the steering forces, but with sufficient elasticity to yield to the shock load if the rudder is not in line when landing.

(*e*) *Clean Lines.* A boat traveling over the water with least disturbance and spray will normally be the one with the least resistance. Spray must be kept away from the propellers and passengers. A V-bottom with hollow flared sections at the bow will cut down the amount of spray and will throw what there is down and away from the boat.

(*f*) *Freeboard and Excess Buoyancy.* There must be proper excess buoyancy for longitudinal and lateral stability and for seaworthiness. Floats normally have a total buoyancy of approximately twice the load-water-line displacement. The proportion for flying boats runs much higher, and the freeboard must be sufficient to keep the cockpits clear of water and spray.

(*g*) *Aerodynamic Cleanness.* It is of course desirable to reduce air resistance of floats to a minimum, so as to improve performance of the seaplane when off the water without injuring in any way its handling properties while on the water. This can be accomplished by streamlining the nose of the float. There are instances of higher speeds and therefore more aerodynamic refinement in a given plane when equipped with a single float than when equipped with wheel landing gear.

Alternate Arrangements

Alternate arrangements would be as follows:

(*a*) *Single Float Seaplane.* This, as well as the twin float arrangement, is practically a land plane with floats in place of the wheel landing gear. This type (or that next described) must be employed where a land plane is to be equipped for water use. The single float is placed on the center line below the fuselage and due to its size provides adequate longitudinal stability. In a single-engine installation the propeller will come directly above the float which is located just to clear it by a few inches. The float itself protects the propeller from spray. Lateral stability is accomplished by floats attached to each lower wing. Small floats placed near the wing tips are more numerous, but in large monoplanes the wing-tip floats are sometimes made larger and brought in nearer to the main float,

carry part of the load, and make a triple-float system. Wing-tip floats get hard usage and must be securely braced and fastened.

(*b*) *Twin Float Seaplanes.* This arrangement is usually most easily substituted for the wheel landing gear. The floats can be located approximately in place of the wheels, and the struts can attach in the same fuselage fittings. This float system is especially desirable where the space below the fuselage must be clear, such as in military machines where bombs and torpedoes are dropped out of the fuselage or for commercial machines fitted for photograph work. Lateral stability is automatically taken care of by the spacing of the floats themselves. If float proportions are the same as for the single-float arrangement, the floats will be shorter in proportion to their smaller size, and longitudinal stability will be reduced. When the floats are short a tail float is sometimes added. A preferable and more general arrangement is to make the floats longer and narrower to attain proper longitudinal stability.

(*c*) *Flying Boat.* This arrangement is proving itself to be the best for general commercial use. The float and fuselage are combined and give more commodious accommodations for passengers as well as giving a feeling of greater safety over the float arrangement with its comparatively small excess buoyancy. The boat is also very much easier to enter or leave. There must be a clear passageway to the box of the boat. Beaching is always made bow to the dock or on the beach. Small single-engined open boats therefore must be of the pusher type, with the propeller well aft of the passengers. In larger boats, with either a deck or cabin top over the passengers, tractor propellers may be used. Where the stern of the boat is carried back and the tail planes are attached to it, there is ample longitudinal stability. Some designs shorten the stern of the boat and set the tail planes out on a boom structure. In this design care must be exercised to see that there is sufficient stability to preclude tipping over backward when drifting backward in a seaway. Lateral stability is accomplished by wing-tip floats the same as with a single-float arrangement. Dornier employs short wing-shaped sponsons, as already explained.

(*d*) *Amphibian.* This arrangement has a wheel landing gear as well as being a seaplane or flying boat and can operate from both land and water. As now designed it is practically a water machine in which a retractable landing gear is incorporated. This type to date has had the least attention given to it, yet will have the brightest future. If properly designed, it should have a high enough ceiling to go practically wherever a land plane can go and in addition be able to land on rivers, bays, and the innumerable inland lakes that are closed to the land machine. The amphibian naturally cannot be as efficient as the other type machines. It must have the higher structural weight of the seaplane and in addition the weight of the landing gear. It must have high enough initial climb to get out of landing fields and a low enough landing speed to land in them safely. These conditions require a reduction in the wing and power loading ratios, with a corresponding reduction in useful load. To more than overbalance, however, the amphibian can land and take off anywhere that there is a body of water of any size and can land in an emergency any where that a land plane can. If it is to be used exclusively over water, the landing gear can be removed and the useful load increased. An additional advantage is the ability to run an amphibian up on a shelving beach or runway on its own landing gear, under its own power, without outside assistance.

Detail Design

Details of the design would be as follows:

(*a*) *Step Location.* A seaplane when planing is supported by water surface that extends only a few feet forward of the step. The location of this supporting surface relative to the center of gravity must be such that there is longitudinal stability. The step location is a matter of good practice and is usually from 18 to 24 in. aft of the center of gravity. If the step comes too nearly under the center of gravity, the machine is balanced on this point rather than on the water surface as a whole and becomes unstable, usually showing a tendency to porpoise. Floats and boats, especially those with long tails, sometimes have an additional step. This step helps to break the water away from the bottom aft of the main step. It also prevents dropping the tail for take-off before the machine is up to take-off speed.

(*b*) *Bow.* The bow must have sufficient lifting power to prevent nosing under in a seaway or when power is first applied. It must not be too full, however, to have the water flow clean at high speeds. Bow lines follow high-speed boat practice, the V being fairly sharp with considerable flare so that the chine line in plan view is very full. This gives good lifting power when the bow is depressed or going into a wave and throws the spray away from the boat.

(*c*) *Stern.* The bottom lines must be straight and break cleanly at the chine line to prevent suction and leave the water easily in getting up to planing speed. In floats and boats with the tail planes carried on booms there must be sufficient buoyancy at the stern to give the proper amount of longitudinal stability to prevent tipping over backward. A transom stern will accomplish this requirement, but the increase in aerodynamical resistance prevents its use. If the stern comes to a point with a vertical sternpost, the chine and deck lines must be full to obtain the necessary displacement. If the stern comes to a horizontal V, the deck line inside elevation should be curved down to meet the bottom at an abrupt angle so that when drifting backward there is no tendency for the stern to dig under the water.

(*d*) *Bottom.* Hydroplane design teaches that, in planing, a flat bottom has least resistance. Hydroplanes now generally have more or less sharp bows with concave sections and a large flare which gradually changes to a flat bottom at the step. Flying boats and seaplanes, however, must leave and land on the water, and in taking off in a seaway or in landing there may be a contact between the flat bottom and the water that will pound the bottom severely. To lessen this pounding the seaplane bottom is given a flat V-section. Sometimes the sections are flared so that at the chine the lines are nearly horizontal. This gives the effect of a partial flat bottom with the advantages of the V preventing pounding. Also spray is thrown downward and away from the float or hull. Where the bottom is of straight V-sections a spray strip is attached to the chine line to throw the spray downward.

(*e*) *Deck.* Training seaplanes, where high performance is not necessary, sometimes have pontoons that are in plan view straight-sided with a V-bow and square stern. The deck is flat, making it easy to stand and walk on. Seaplanes requiring high performance usually have the deck lines approximately semicircular in section. Flying-boat superstructure follows boat practice altered to meet flying-boat conditions.

(*f*) *Hydrovanes.* This type of design has not been discussed, although reports of tests by Graham Bell and Baldwin, together with the successful boats built in France and Italy, show that it is more efficient than the hydroplane. Hydrovanes consist of airfoil-section wings attached to a framework below the bottom of the boat, and their action is the same in water as a wing in the air. They are not on the surface of the water like a hydroplane, but operate beneath it. Due to the great difference in density between air and water these planes are very small, and due to their high lift will lift the boat clear of the water at fairly low speeds. Practical difficulties arise, such as inability to get longitudinal control by the tail planes and danger of fouling mooring lines, floating debris, seaweed, etc. It also cannot be beached readily. The idea, however, presents advantages that warrant very serious study.

(*g*) *Structural Arrangement, Wood, Metal, Composite.* Structural design must be made to conform to the loads imposed. At anchor in a seaway the floats or hull must stand hogging and sagging stresses the same as any boat. These stresses are a maximum when the wave length equals the length of the boat. The bow in addition must be of sufficient strength to take water loads when under way. The bottom forward of the step receives the highest stresses, although the bottom aft of the step may receive high stresses in a stalled or pancake landing. Tests made by the U. S. Navy on a V-bottom seaplane under various conditions show maximum loads on the forward bottom to be 7 lb. per sq. in. at the step, gradually decreasing to 3 lb. per sq. in. at the keel and $^1/_2$ lb. per sq. in. at the chine line at a point well forward near the bow. The after-bottom loads decrease from a maximum of 6 lb. per sq. in. at the sternpost to 2 lb. per sq. in. at the step. Float design for strength is usually based upon a water pressure on the bottom whose total is the gross weight of the machine multiplied by a load factor of approximately 8 and the center of pressure of which passes through the center of gravity of the machine. This condition imposes forces in the pontoon struts and brace wires that can be easily calculated. The attaching fittings of these struts and wires are designed to take these loads and are attached to the pontoons so that the forces are carried into and distributed in the float structure.

A flying boat carries the equipment and useful load directly within itself and must be designed to carry these loads with the proper load factor, and in addition the concentrated loads of the struts, wires, and hinges supporting the wings and engines. It must also be designed to be lifted by the wings and to sustain the tail loads, the same as a fuselage in a land machine.

The skin, bottom, sides, and deck supply the larger part of the strength and rigidity as in other monocoque construction. The main trussing usually consists of a backbone running the entire length with transverse bulkheads attached to it. Secondary supports for the skin consist of transverse ribs and longitudinal stringers to prevent its buckling. The bottom in addition must take the

water loads. The backbone, sides, and bulkheads form the main supports for these loads, and the secondary supports either consist of longitudinal stringers supported by the bulkheads or by trussed transverse ribs supported by the backbone and chine. In float construction the backbone usually extends from the keel to the deck and is a longitudinal bulkhead dividing the pontoon in halves throughout a large proportion of its length. In flying boats the backbone is below the floor line and in many cases is replaced by the flat sides of the boat reinforced by a truss construction. Where holes are cut in the skin (in the deck for the cockpit, for instance), reinforcements must be incorporated to make up for the reduced strength.

Pontoon and hull construction is of three types:

(*a*) Wood
(*b*) Metal
(*c*) Composite.

Wood construction usually consists of mahogany-plywood skin and bulkheads and spruce struts and framing. The mahogany plywood is usually two thicknesses of narrow strips laid at an angle (or at right-angles) to each other with a thickness of cloth between, laid in marine glue and is fastened together with clinched copper nails or tacks. Three-ply mahogany veneer glued together with a waterproof glue is sometimes used on flat surfaces or those having a simple curvature, but unless the gluing is done perfectly, the veneer is likely to open up after a time.

Wood construction has the advantage of cheapness for experimental machines or for small production. Its greatest drawback is the absorption of water. The weight increase due to it very appreciably affects the performance of the machine.

Metal construction has been almost entirely in duralumin, although experiments are now being made with stainless steel. Corrosion due principally to electrolysis must be overcome, and especially joints between different metals must be carefully insulated by paint to prevent electrolytic action. The type of construction follows closely that of wood. There appears to be a preference for supporting the bottom between main bulkheads by longitudinal stringers rather than by transverse ribs. Metal construction will not absorb water. Its cost, however, for experimental machines or small production is nearly prohibitive. Metal pontoons are now being specialized in by some manufacturers who are standardizing them and with the increased production are able to produce for a reasonable price.

Composite construction is that employing wood framing and a metal skin. It has the advantage of the low cost of wood construction for the frames and the absence of water absorption by the metal skin. Several very successful designs have employed this type of construction, and for small production it has a decided advantage from a cost standpoint.

Conclusions

Seaplane operation, although confined to the sea and to waterways, commercially can compete to a great extent with the land planes. A large proportion of air routes can follow or can be arranged to follow waterways. Landing fields are not needed, and terminal facilities need only consist of a dock or landing stage along the waterfront in the heart of the city. Hangars and servicing facilities can be located a comparatively short distance away where land is cheap. For pioneering, transportation over, and exploration of undeveloped territory its advantages are outstanding. Expeditions using flying boats have explored the upper Amazon many miles beyond civilization. Seaplanes have flown the length of Africa, following the Nile and the chains of lakes. In the gold rush that occurred in Canada a few years ago seaplanes flew a route in a few hours that required over a week by land.

Within a few years flying boats will be designed with sufficient seaworthiness, reliability, and comfort to establish the ultimate method of air travel across the sea. With a slight sacrifice in pay load, the amphibian is well adapted for routes a greater proportion of which is over the water, the landing gear being available for an emergency.

For the private owner, the amphibian is the ideal machine. He can operate it anywhere, either over land or water, and there is nothing to compare with the thrill experienced when taking off from, or skimming over, the surface of the water.

The market for the flying boat and the amphibian is here and only waiting for development by the manufacturer and operator.

A Study on Airplane Ceiling

By LOUIS BREGUET[1] and MAURICE ROY,[1] PARIS, FRANCE

The useful torque exerted on the propeller shaft depends principally on the altitude, on the throttle setting, and on the r.p.m. of the engine. While the French Section Technique d'Aeronautique postulates that the variation of torque with altitude depends mainly on the density, the authors believe that, given good altitude adjustment and a carburetor heated to a constant temperature, the variation of torque depends directly on the relative pressure at altitude.

Having established formulas for power variation with altitude, it is possible to calculate the ceiling theoretically, provided the characteristics of the plane have been determined by tests in horizontal flight at any altitude and at any load whatsoever.

To obtain the greatest possible ceiling it is essential that:

1. **The engine should turn at full throttle the greatest possible r.p.m.**
2. **The weight of the machine per horsepower at the ground should be as low as possible.**
3. **The span loading should be as low as possible.**
4. **The propeller should be designed to have maximum efficiency at the condition of minimum power.**

With a specially designed plane and an unsupercharged engine the authors estimate that an altitude of 13,000 meters can be reached, beating previous records by some 2000 meters.

A few simple tests at one altitude with a careful record of all the elements involved can be made to give all the plans and propulsive group characteristics. The theory and practice of such experiments are fully developed in the paper.

To make full use of supercharging it is apparently necessary to change not only the pitch but also the diameter of the propeller. An alternative proposition would be to employ a variable-speed gear between engine and propeller. This fascinating idea is in some respects easier to realize than the propeller of variable pitch and diameter.

Both supercharging with altitude and undercharging at the ground offer possibilities of increasing ceiling.

Torque of an Unsupercharged Engine

THE torque Q (in kg-m.) of an aviation engine depends principally on the altitude h, on the throttle setting, which we shall characterize by a certain parameter a, and lastly on the number of revolutions n (in r.p.m.) of the engine.

Let us consider an unsupercharged engine and suppose that the altitude adjustment, which maintains constant at all altitudes the air-gas ratio of the carbureted mixture, is properly effected.

Under these conditions, the useful torque on the propeller shaft is given by

$$Q_h(a, n) = \nu Q_0(a, n) \quad \ldots\ldots [1]$$

The coefficient ν, called the coefficient of reduction of torque with altitude, depends by hypothesis only on h, and the factor $Q_0(a, n)$ represents what is known as the equivalent torque at the ground ($h = 0$) of the engine under the condition (a, n).

For a normal engine Q_0 is the torque actually developed by the engine under the conditions (p_0, ν_0, T_0) of the standard atmosphere at sea level.

For an undercharged engine, Q_0 is a fictitious torque which the engine is not allowed to develop at sea level at any throttle setting, since on such an engine this is limited at sea level.

Law of Variation of Coefficient ν With Altitude

The law officially recognized by the French S. T. I. Ae, called the standard law, for the variation of the coefficient ν with h, is based on the hypothesis that ν depends only on the relative density (relation of specific weight of air at h altitude and at the ground) of the atmosphere, and that the torque of the frictional losses of the engine keeps a constant value with the altitude, for a fixed a and n.

The standard law is expressed by the relation

$$\nu = 1.11\ \delta - 0.11 \quad \ldots\ldots [2]$$

This law, based on a few debatable experiments, justifies certain reservations. One can criticize especially its principle by noting that in considering an engine whose carburetor is heated to a constant temperature, independently of the altitude, the coefficient of such an engine ought to depend only on the relative pressure μ of the atmosphere (ratio of air pressure at h altitude and at the ground) and not on the relative density δ.

Moreover it should be noted that, in principle, the coefficient ν ought to depend on a and on n as well as on all the characteristics of the engine (operating at 2- or 4-cycle, volumetric compression, etc.), and the various arrangements effecting carburation lubrication, and cooling.

Thus, strictly, one ought to reserve for each type of engine a special law appropriate to the coefficient ν. Various forms of the law for ν have been proposed, each different from the standard law [2] and each quite debatable.

Certain recent experiments seem to indicate that, up to altitudes of the order of 10,000 meters (32,800 ft.), for engines supplied with a good altitude correction and a carburetor heated to a constant temperature, one can formulate an approximate law, both simpler and more nearly true than the standard law [2], namely,

$$\nu = \mu \quad \ldots\ldots [3]$$

To compare the laws [2] and [3] the authors have summarized in Table 1 the corresponding values of ν calculated by both of these laws, for standard altitude increasing from 1000 meters by 1000-meter steps.

TABLE 1

Standard altitude, h Meters	(Feet)	Relative density, δ	Values of coefficient ν from Standard law [2]	Approximate law [3]	Difference in per cent of [2]
0	0	1.000	1.000	1.000	—0.000
1,000	(3,281)	0.905	0.895	0.887	—0.010
2,000	(6,562)	0.820	0.803	0.785	—0.020
3,000	(9,843)	0.743	0.713	0.692	—0.029
4,000	(13,123)	0.669	0.630	0.609	—0.035
5,000	(16,404)	0.601	0.556	0.534	—0.040
6,000	(19,685)	0.539	0.486	0.465	—0.044
7,000	(22,966)	0.4815	0.425	0.405	—0.047
8,000	(26,246)	0.429	0.366	0.352	—0.039
9,000	(29,527)	0.3805	0.311	0.3035	—0.024
10,000	(32,808)	0.337	0.262	0.261	—0.004
11,000	(36,089)	0.297	0.215	0.2235	+0.035

[1] Breguet Aviation Works.

Presented at the Third National Meeting of the A.S.M.E. Aeronautic Division, St. Louis, Mo., May 27 to 30, 1929.

This table, as well as the curves of Fig. 1, shows that the simple law [3] gives, up to $h = 10{,}000$, values of ν relatively little different from those of the standard law [2]; values, moreover, which are always lower as they result from the hypothesis of reheating of the incoming mixture to constant temperature.

Expressions for h as Functions of δ and μ in the Standard Atmosphere

Let us recall that the standard atmosphere, supposedly in hydrostatic equilibrium, is characterized by the following hypotheses:

For $h = 0$: $p_0 = 1.033$ kg. per sq. cm.; $v_0 = 1.225$ kg. per cu. m.; $T_0 = 273 + 15 = 288$ deg. cent. abs. (= 59 deg. fahr.)

For $h = 0$ to 11,000 m. (36,090 ft.): $T = T_0 - 0.0065\,h$.

For $h > 11{,}000$ meters: $T = 273 - 56.5 = 216.5$ deg. cent. (= —81 deg. fahr.).

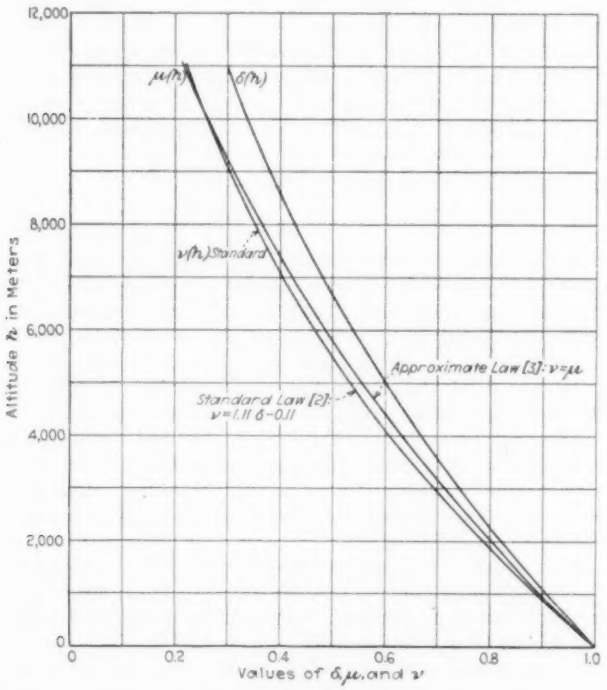


Fig. 1 Variation of δ, μ, and γ With h in Standard Atmosphere

It is easy to show that these hypotheses lead to the following laws of variation of δ and μ as functions of h:

$$\text{For } h = 0 \text{ to } 11{,}000 \text{ m., } \left\{\begin{matrix}\delta = (1 - 0.000026)^{4.255} \\ \mu = (1 - 0.000026)^{5.255}\end{matrix}\right\} \quad \ldots [4]$$

$$\text{For } h > 11{,}000 \text{ meters, } \log\frac{1}{\delta} = \frac{h - 3{,}300}{14{,}600}; \log\frac{1}{\mu} = \frac{h - 1{,}490}{14{,}600} \quad [5]$$

From these relations we deduce the following expressions for h as functions of δ and μ:

$$\text{For } h = 0 \text{ to } 11{,}000 \text{ meters, } \left\{\begin{matrix}h = 44{,}250\ (1 - \delta^{0.235}) \\ h = 44{,}250\ (1 - \mu^{0.191})\end{matrix}\right\} \quad \ldots [6]$$

$$\text{For } h > 11{,}000 \text{ meters, } \left\{\begin{matrix}h = 3{,}300 + 14{,}600 \log 1/\delta \\ h = 1{,}480 + 14{,}600 \log 1/\mu\end{matrix}\right\} \quad \ldots [7]$$

These relations permit us later to give an analytical expression to the ceiling.

Determination of the Theoretical Ceiling of an Airplane With an Unsupercharged Engine

Let us now dismiss the case of supercharged engines and occupy ourselves exclusively with normal and undercharged engines, for which we can say that the torque, at any altitude whatever, obeys the fundamental law [1].

It is known that the theoretical ceiling H or maximum altitude which an airplane can attain is characterized by the fact that, at this altitude, the plane can only fly horizontally at the single angle of incidence for which there is an exact balance between the power furnished by the engine[2] and that absorbed by the propeller, this balance permitting of but one solution.

Let us express this condition of equilibrium by indicating the effective torque of the engine. At the ceiling H let:

V_H = speed of plane (meters per sec.)
n_H = r.p.m. of propeller
a_H = throttle setting.
ν_H = coefficient of torque reduction
W_H = weight of plane (kg.)
S = wing area (sq. meters), and
D = diameter of propeller (meters).

The power supplied by the engine is (in kg-m. per sec.)

$$\nu_H \frac{2\pi}{60} n_H Q_0(a_H, n_H)$$

The power absorbed by the propeller is

$$W_H V_H \left(\frac{C_x}{C_z} \cdot \frac{1}{\eta}\right)_H$$

The condition of equilibrium between the two powers leads to

$$\nu_H \cdot \frac{2\pi}{60} n_H Q_0(a_H, n_H) = W_H V_H \left(\frac{C_x}{C_z} \cdot \frac{1}{\eta}\right)_H \quad \ldots [8]$$

which can be written

$$\frac{1}{\nu_H} = \frac{Q_0(a_\eta, n_H)}{W_H D \left(\dfrac{V}{\dfrac{2\pi}{60} nD} \cdot \dfrac{C_x}{C_z} \cdot \dfrac{1}{\eta}\right)_H} \quad \ldots [9]$$

Knowing that the efficiency of the propeller depends only on the parameter

$$\frac{V}{\frac{2\pi}{60} nD} = \gamma \quad \ldots [10]$$

and that, in horizontal flight, γ is bound to the incidence, it follows that the function

$$\xi = \frac{C_x}{C_z} \cdot \frac{\gamma}{\eta} \quad \ldots [11]$$

is, in horizontal flight, a simple function of the incidence which depends neither on the altitude nor the total weight of the plane.

The ceiling H is marked by the fact that in Equation [9] ν passes through its minimum, that is to say, the values of a, n, and i (or γ) in horizontal flight form a system rendering ν a minimum.

Moreover, one admits, with a very close degree of approxi-

[2] In this paper the authors will consider only the case of a single-engined plane. The case of the multi-engined plane would be treated in the same way and would result merely in slightly more complicated formulas, which, however, may be easily derived from those for the single-engined plane.

mation, for actual aviation engines, that the torque $Q_0(a, n)$ only depends on a, that is, for a fixed throttle setting, it varies very little with n, at least between the limits of speed utilized.

From this it can be seen from [9] that the term $1/\nu$ does not depend on n and that its numerator depends only on a, while its denominator depends only on i (the incidence). There results from this fact that at the ceiling H where ν_H is a maximum:

1 The admission of the gas is regulated to the value a_m which gives the maximum torque (case of full throttle)
2 The angle of incidence is that which gives a minimum value to the function

$$\xi = \frac{C_x}{C_z} \cdot \frac{\gamma}{\eta} \quad [11]$$

which characterizes the different possible cases of horizontal flight. In the case of incidence i_H at the ceiling, the condition of horizontal flight at the ground corresponds to the number of revolutions of the engine n_0 related to the number of revolutions n_H at the ceiling, by the expression

$$n_0 = n_H \sqrt{\delta_H} \quad [12]$$

Let us introduce in [9] the equivalent engine torque at the ground with full throttle and at n_0 revolutions by placing

$$Q_0(a_m \cdot n_H) = K_0 Q_0(a_m, n_0) \quad [13]$$

Equation [9] will then appear in the form:

$$\frac{1}{\nu_H} = \frac{Q_0(a_m, n_0) K_0}{D W_H \xi_H} \quad [14]$$

Let us put

$$X_H = \frac{Q_0(a_m, n_0) K_0}{D W_H \xi_H} \quad [15]$$

In this relation K_0 is a factor near unity and which may be taken as a first approximation equal to 1. The other elements are then known or can be determined by tests in horizontal flight at any altitude and any load whatever, as will be shown farther on, and without the necessity in any case of climbing to the ceiling, which solves in advance the problem of this determination. Having determined X_H, it is easy, on a graph giving h as a function of $1/\nu$ as established from the standard law [2], to determine H. One can also in this case, for a ceiling included between about 2000 and 7000 meters (6560 to 13,100 ft.), apply the following approximate formula:

$$H_{\text{meters}} = 325 + 18{,}090 \log X_H \quad [16]$$

from [2]. If one wishes to take account of the variation of Q_0 with the revolutions and add to the accuracy of H, he will proceed by successive approximations in first making $k_0 = 1$ in [15] and determining a first approximate value H' of the ceiling. Formula [12] will then permit an approximate calculation $n_{H'}$ of n_H which will lead to a value nearer to k_0 taking account of the relation of the definition [13]. By substituting in [14] this new value of k_0, one will get a new value for X_H which is satisfactory for the final calculation of H.

If it is desired to use the simpler law [3] proposed, up to 10,000 meters, H is given analytically by the following formula deduced from [3] and [6]:

$$H = 44{,}250 \left(1 - \frac{1}{X_H^{0.191}}\right) \text{ for } H < 10{,}000 \quad [17]$$

Other Formulas for Calculating Ceiling

The formula for determination of the ceiling H can be expressed in different ways. Let us express the equivalent power P_0 (in hp.) as

$$P_0(a_m, n_H) = n_H Q_0(a_m, n_H) \frac{2\pi}{60} \cdot \frac{1}{75} \quad [18]$$

in place of the equivalent torque Q_0 in [8] which expresses the equilibrium of the engine power and the power absorbed by the propeller. Then:

$$\frac{1}{\nu_H} = \frac{75 P_0(a_m, n_H)}{W_H V_H \left(\frac{C_x}{C_z} \cdot \frac{1}{\eta}\right)_H} \quad [19]$$

But the speed at the ceiling V_H is given by

$$V_H = \sqrt{\frac{W_H}{\frac{\rho_0}{2g} \delta_H S C_z}} = 4 \sqrt{\frac{W_H}{S}} \cdot \frac{1}{\sqrt{\delta_H}} \cdot \frac{1}{\sqrt{C_z}} \quad [20]$$

and Equation [19] can be expressed as

$$\frac{1}{\nu_H \sqrt{\delta_H}} = \frac{75 P_0(a_m, n_H)}{4 W_H \sqrt{\frac{W_H}{S}} \left(\frac{C_x}{C_z^{3/2}} \cdot \frac{1}{\eta}\right)_H} \quad [21]$$

In horizontal flight the function $\zeta = \frac{C_x}{C_z^{3/2}}$ and the efficiency η of the propeller depend on the incidence only. The function

$$q = \frac{C_x}{C_z^{3/2}} \cdot \frac{1}{\eta} = \frac{\xi}{\eta} \quad [22]$$

then depends only on the incidence.[3]

Let

$$K_H = \frac{75 P_0(a_m, n_H)}{4 W_H \sqrt{\frac{W_H}{S}} q_H} \quad [23]$$

Suppose that this quantity has been determined for a given plane. To find the ceiling it will be sufficient if one adopts the standard law [2] of variation of ν, to mark off on a graph giving h as a function of $\frac{1}{\sqrt{\delta}}$ from [2] the value H corresponding to K_H. One can also, if H lies between 2000 and 7000 meters, use the following approximate formula (from law [2]):

$$H_{\text{meters}} = 285 + 12{,}800 \log K_H \quad [24]$$

If we adopt the simpler law [3] of variation of ν for altitudes less than 10,000 meters, H could be calculated from the following formula deduced from [3] and [6]:

$$H_{\text{meters}} = 44{,}250 \left(1 - \frac{1}{K_H^{0.1345}}\right) \quad [25]$$

Note that in expression [21], K_H, which occurs in the preceding formulas, involves the number of revolutions n_H of the engine at the ceiling, a value which is not known directly, in

[3] The function q thus defined is equal to $4q'$, q' designating the function $\frac{K_x}{(K_z)^{3/2}} \cdot \frac{1}{\eta}$ that one of the authors (Louis Breguet) has named previously "moto-sustaining quality" and on which he has based the first classic French formula for ceiling.

general, and whose influence on the value of $P_0(a_m, n_H)$ is quite important. To avoid the occurrence of this value, one can show in the formula the number of revolutions n_0 of the engine in horizontal flight at the ground at the same angle of incidence as that at the ceiling. From [12] and [13] we can write

$$P_0(a_m, n_H) = P_0\left(a_m \cdot \frac{n_0}{\sqrt{\delta_H}}\right) = \frac{n_0}{\sqrt{\delta_H}} Q_0(a_m, n_0) K_0$$

$$= \frac{K_0}{\sqrt{\delta_H}} P_0(a_m, n_0) \ldots\ldots [26]$$

and [21] becomes

$$\frac{1}{\nu_H} = \frac{75 P_0(a_m, n_0) K_0}{4 W_H \sqrt{\frac{W_H}{S}} q_H} \ldots\ldots [27]$$

Comparing this with [14] and [15] it is seen at once that

$$\frac{75 P_0(a_m, n_0) K_0}{4 W_H \sqrt{\frac{W_H}{S}} q_H} = X_H \ldots\ldots [28]$$

This formula is identical with [15], and it is evident that one would have been able to pass directly from one to the other by substituting the equivalent of power $P_0(a_m, n_0)$ for the torque equivalent $Q_0(a_m, n_0)$. This comparison affords the opportunity to make an interesting observation. We have seen, in relying on the fact that the torque Q_6 is seemingly independent of the number of revolutions at full throttle admission a_m, that the incidence at the ceiling corresponds to the minimum of the function

$$\xi = \frac{C_x}{C_z} \cdot \frac{\gamma}{\eta}$$

Now, in relying on the relation [23] and neglecting in it the perceptible variation of P_0 with n, it has often been stated that the incidence at the ceiling is that which renders a minimum in horizontal flight, the function

$$q = \frac{C_x}{C_z} \frac{1}{\sqrt{c_z}} \frac{1}{\eta}$$

It is easily seen that this statement is not exact, for the minimum of q does not coincide with that of ξ, which alone corresponds truly to conditions at the ceiling. According to the type of plane and the characteristics of the engine-propeller group, the incidence at the actual ceiling can be less or more than that of the minimum of q.

As a rule for actual planes, it is the first case which is realized. One can, moreover, as will be seen further on, verify this fact by determining the coefficients ξ and q in functions of the incidence in horizontal flight, by trials in flight at one altitude and at any load whatever.

Conditions for Reaching the Highest Ceiling

Let us return to Equation [28] in which one can, as has been seen, equate k_0 to unity on a first approximation. To obtain the highest theoretical ceiling possible it is proper to combine with the ceiling the conditions which give to X_H the greatest value possible. This observation shows immediately that it is necessary at the ceiling that:

1 The engine shall turn at full throttle at the maximum number of revolutions, since n_0 is as much larger as n_H is larger.

2 The weight of the machine per horsepower at full power at the ground shall be as low as possible (power loading).

3 The weight of the machine per unit of wing span shall be as small as possible (span loading), and

4 The propeller shall be adapted so as to give in horizontal flight its maximum efficiency for the case of incidence corresponding to the minimum of $\xi = \frac{C_x}{C_z^{3/2}}$. In this case the minimum of q coincides with that of ξ and this case of incidence is, with the propeller considered, that of the ceiling. It is understood that these varying conditions are not independent, and it is not always possible to obtain them simultaneously, either because of a propeller of excessive diameter and bulk or with a relative pitch p/D too high to allow an easy take-off. The best practical solution, then, is to compromise judiciously in each case.

Possible Record of Altitude for a Plane With Normal Engine

It is interesting to see, by relying on the principles of construction actually realizable, to what height a plane can climb that is fitted with a normal engine and specially constructed with an altitude record in view. Let us establish a relation between the various weights which compose the whole. At the start, the total weight W_0 comprises:

1 Weight of the airplane structure represented by αW_0, α denoting the coefficient of lightness of the structure

2 Weight of the engine-propeller group represented by rP_0, P_0 designating the maximum engine power at the ground and r the specific weight of the group, that is, its weight per horsepower at maximum power

3 Weight of fuel which, it is assumed, corresponds to t hours of running at full power at the ground; weight represented by tmP_0, m designating the specific consumption of the engine in kilograms of gasoline and oil per effective horsepower-hour

4 The weight of the crew (1 pilot) and his equipment (instruments for flying, special clothing, oxygen apparatus, etc.) estimated at 120 kg. (265 lb.).

Under these conditions,

$$W_0 = \alpha W_0 + r W_0 + tmP_0 + 120 \ldots\ldots [29]$$

The weight W_H at the ceiling is equal to the weight at take-off W_0 less the stock of fuel and oil which is used up. Then:

$$W_H = W_0 - tmP_0 = W_0\left[1 - tm\frac{1-\alpha}{r + tm}\right] + tm\frac{120}{r + tm} \ldots [30]$$

To calculate the ceiling H, we shall apply the relation [21] in which we make $P_0(a_H, n_H) = P_0$, reverting to the supposition that the engine operates at the ceiling at the rated full-load speed which gives the maximum power at the ground.

Let us first try to render a maximum the relation

$$J = \frac{P_0}{W_H \sqrt{\frac{W_H}{S}}}$$

We have:

$$J = \frac{P_0}{W_H \sqrt{\frac{W_H}{S}}} = \sqrt{S} \frac{\frac{W_0(1-\alpha) - 120}{r + tm}}{\left\{W_0\left[1 - tm\frac{(1-\alpha)}{r + tm}\right] + tm\frac{120}{r + tm}\right\}^{3/2}} \ldots\ldots [31]$$

In this relation the coefficient of lightness α of the glider, for a given type of construction, is almost independent of W_0/S and depends only on the dimensions, increasing slightly with them. Thus one can regard α as dependent only on S for one type of construction and a given form of plane, and not on W_0.

Moreover, the coefficients r, m, and t are nearly independent of P_0 and, as a result, of W_0. In reality, nevertheless, one must consider that, the less the power of an engine-propeller group, the greater its specific weight becomes.

However this may be, let us assume S, α, r, t, and m fixed in the preceding relation [31] and seek the value of W_0 which renders this expression a maximum. It is easily found that this value is:

$$W_0 = \frac{120\ [3r + (2 + \alpha)\ tm]}{(1 - \alpha)\ (r + tm\alpha)} \quad \ldots\ldots [32]$$

for which

$$P_0 = \frac{240}{r + tm\alpha} \quad \ldots\ldots [33]$$

$$W_H = \frac{360}{1 - \alpha} \quad \ldots\ldots [34]$$

It is seen that for this optimum value of W_0, W_H is independent of the number of hours t of the fuel carried at the start. But the relation J considered is not independent, for its maximum is:

$$J_{max} = 0.385 \sqrt{\frac{S}{120}} \frac{(1 - \alpha)^{3/2}}{r + tm\alpha} \quad \ldots\ldots [35]$$

To fix these ideas, let us adopt the following hypothesis actually attainable by a plane and engine especially made for the altitude record:

S = 30 sq. m. (323 sq. ft.); α = 0.30; r = 1 kg. per metric hp. (2.2 lb. per hp.); m = 0.235 kg. per metric hp-hr. (0.518 lb. per hp-hr.); t = 2 hr.

Then

W_0 = 613 kg. (1351 lb.); P_0 = 210 hp. (metric); W_H = 514 kg. (1131 lb.).

Assume the following data:

	At start	At ceiling
Airplane structure, kg	184	184
Engine, kg	210	210
Fuel, kg	99	...
Load, kg	120	120
Total weight, kg	613	514
Load per hp., kg	2.93 (6.45 lb.)	2.45 (5.4 lb.)
Load per sq. meter, kg	20.45 (4.2 lb. per sq. ft.)	17.12 (3.51 lb. per sq. ft.)

and

$$J = 0.0985.$$

It is seen that this points to a single-seater plane with very light loading, analogous to a glider equipped with an engine having relatively great power and which represents in itself alone, despite its small specific weight (r = 1 kg. per hp.), nearly 35 per cent of the total weight at the start.

It may indeed be objected that, for so light a plane, a coefficient of lightness of 30 per cent (α = 0.30) would perhaps be difficult to realize, for one would be compelled to use material of insufficient thickness to adequately resist the secondary stresses. However, to show that such a conception is a mistaken one it may be said in passing that in working with other data, more like those corresponding to actual planes of medium weight, one would realize values of the relation J of the same order as that indicated above.

As an example, let us take a biplane of 50 sq. m. (538 sq. ft.) surface, loaded to 2000 kg. (4408 lb.) at the start, of which the structure represents 30 per cent. Assuming as before that r = 1 kg. per hp.; m = 0.235 kg. per hp-hr.; and t = 2 hr., would give P_0 = 870 hp.; W_H = 1590 kg. (3507 lb.). That is to say:

	At the start	At the ceiling
Airplane structure, kg	600	600
Engine, kg	870	870
Fuel, kg	410	...
Loading, kg	120	120
Total, kg	2000	1590
Load, per hp., kg	2.30 (5.08 lb.)	1.83 (4.04 lb.)
Load per sq. meter, kg	40 (8.2 lb. per sq. ft.)	31.8 (6.52 lb. per sq. ft.)

In this case, which does not correspond to the maximum of J for the values chosen of S, α, r, m and t, we have J = 0.0970. The maximum of this relation would be 0.125 and would correspond to the same weight (W_0 = 613 kg.) as in the preceding example, since in these examples the surface only is different and the value of W_0 giving the maximum of J is independent of this surface.

Let us now return to our first example, a light single-seater, and attempt to determine its ceiling.

For evaluating the value of q_H by substitution in [21] it is necessary to make certain hypotheses.

For the region of the ceiling we must take the minimum condition of $\zeta = \frac{C_x}{C_z^{3/2}}$ by assuming that the propeller is adapted to furnish its maximum efficiency (n = 0.75) in this region. Let us consider the parabolic polar diagram between the region of fineness (minimum of C_x/C_z) and that of the minimum of ζ. By designating by R the aspect ratio of the elliptic wing equivalent of the wing considered, the polar equation is:

$$C_x = a + \frac{1}{R\pi} C_z^2 \quad \ldots\ldots [36]$$

Let (C_{x1}, C_{z1}) be the region of fineness; then

$$\frac{C_{x1}}{C_{z1}} = \tan \varphi_1 \quad \ldots\ldots [37]$$

and

$$C_{z1} \tan \varphi_1 = \frac{2}{R\pi} C_{z1}^2$$

That is to say:

$$C_{z1} = \frac{R\pi}{2} \tan \varphi_1 \quad \ldots\ldots [38]$$

The limit (C_{x2}, C_{z2}) of the minimum of ζ is characterized by:

$$C_{x2} = \sqrt{3}\ C_{z1} \quad \ldots\ldots [39]$$

$$\zeta_2 = \zeta_{min} = 0.702 \sqrt{\frac{\tan \varphi_2}{R}} \quad \ldots\ldots [40]$$

This relation shows the influence of the extension equivalent R on the coefficient ζ_2. By increasing R, not only is the denominator of ζ_2 increased but also, in general, the numerator is reduced. Moreover, it is worth while to notice that if the aspect-ratio extension R becomes too large, the value of c_{z2} indicated above will exceed the maximum lift, c_{z1} becoming quite large and, as a result, the polar diagram can no longer resemble a parabola as far as the minimum case of ζ is concerned.

In this case, the minimum of ζ is found to be decidedly greater than the value indicated by [40].

To fix these ideas in the mind, let us assume that the airplane structure considered has a fineness $\tan \varphi_1 = 0.065$ and an aspect ratio $R = 7$. Under these conditions, the condition of fineness corresponds to $c_{z1} = 0.715$ and that of the minimum of ζ to:

$$C_{z2} = \sqrt{3} \times 0.715 = 1.235; \quad \zeta_{z2} = 0.0677$$

These values are reasonable. They correspond to those of an airplane structure having aerodynamic qualities which are exceptionally good but are nevertheless obtainable. It is to be noted, in fact, that certain tests of particularly fine wing-truss structures have shown values of the order of 0.055 for the minimum of ζ.

For the plane considered, by using the figures established above we have:

$$J = 0.0985; \quad \zeta_{\min} = 0.0677; \quad \eta_{\max} = 0.75$$

and consequently, from [21]:

$$\frac{1}{\nu_H \sqrt{\delta_H}} = \frac{75 \times 0.0985 \times 0.75}{4 \times 0.0677} = 20.5 \ldots\ldots [41]$$

According to the standard law [2] the altitude corresponding to this number is 13,560 meters (44,500 ft.), which is the theoretical ceiling of the plane considered. At this altitude the conditions of flight are characterized by:

$$\frac{W_H}{S} = 17.12 \text{ kg. per sq. cm.}; \quad V_H = 33.3 \text{ m. per sec.}$$

$$= 120 \text{ km. per hr. (74.6 m.p.h.)}$$

The effective power of the engine is reduced to about 23 hp., which is sufficient to drive a plane weighing 514 kg. (1130 lb.) at a speed of 120 km. per hr. at the region of the ceiling for which $c_{z2}/c_{x2} = 0.075$ and $\eta = 0.75$.

It is seen then that it is quite possible to reach and to exceed 13,000 meters, that is, to surpass the recent records of altitude in the vicinity of 11,000 meters, with a specially designed plane equipped with a standard engine.

It is understood of course that the preceding calculation assumes that every precaution is taken in order that, at the very high altitude contemplated, the carburation and ignition shall operate properly and without trouble.

Determination of the Aerodynamic Characteristics of an Airplane by Flying Tests

It is known that one can estimate, with a certain approximation, by calculation based on laboratory tests on small-scale models of plane and propeller, the aerodynamic characteristics of a full-sized plane.

But, in addition to the error due to the scale effect, which is yet impossible to foresee or to figure, this method of calculation involves numerous other sources of error. Among these are the airplane-propeller interaction or mutual effect and the differences in the equipment of the real plane and its model (guy wires, struts, radiators, nature and elastic properties of the wing surfaces, etc.).

Further, it is of the greatest interest to determine directly by flying tests the aerodynamic characteristics of an airplane-propeller system, and we shall therefore show how to proceed with tests in horizontal flight.

Let us first assume as correct the law of variation of the coefficient ν of reduction of the torque with the altitude.

Imagine that a plane attains a horizontal stage at a certain incidence i, a stage in the course of which one notes the following:

1 The relative pressure μ and the temperature T of the atmosphere, from which is deduced the corresponding standard altitude h
2 The aerodynamic speed V_{hi} (in meters per second) at the altitude h and incidence i
3 The speed of rotation n_{hi} (in r.p.m.) of the propeller
4 The throttle setting a_{hi}
5 The weight W (in kg.) of the plane, deduced from its weight at take-off and from the reduced load corresponding to the fuel consumed.

The power (in kg-m.) furnished by the engine is

$$\nu_h \frac{2\pi}{60} n_{hi} Q_0 (a_{hi}, n_{hi}),$$

and the torque $Q_0(a_{hi}, n_{hi})$ can be determined by tests on the testing block on the ground, during which the engine will be rotated at the revolutions n_{hi} by giving to the throttle the position corresponding to the value a_{hi} of admission.

Moreover, the power utilized for the flight considered can be written:

$$WV_{hi}\left(\frac{C_x}{C_z} \cdot \frac{1}{\eta}\right)_i \quad \text{or} \quad 4W\sqrt{\frac{W}{S}}\frac{1}{\sqrt{\delta_h}}\left(\frac{\zeta}{\eta}\right)_i$$

Equating these expressions to that of the power furnished by the engine,

$$\nu_h \frac{2\pi}{60} n_{hi} Q_0 (a_{hi}, n_{hi}) = WV_{hi}\left(\frac{C_x}{C_z} \cdot \frac{1}{\eta}\right)_i$$

$$= 4W\sqrt{\frac{W}{S}}\frac{1}{\sqrt{\delta_h}}\left(\frac{\zeta}{\eta}\right)_i \ldots\ldots [42]$$

It is particularly interesting to know, for horizontal flight, the values of:

$$\tan \varphi' = \frac{C_x}{C_z} \cdot \frac{1}{\eta}; \quad \xi = \frac{C_z}{C_s} \cdot \frac{\gamma}{\eta}; \quad q = \frac{\zeta}{\eta} \ldots\ldots [43]$$

which depend only on the incidence i, in functions of this, or better, of the lift coefficient C_z.

These quantities will be given by the formulas:

$$\tan \varphi' = \frac{\nu_h \cdot \frac{2\pi}{60} n_{hi} Q_0 (a_{hi}, n_{hi})}{WV_{hi}} \ldots\ldots [44]$$

$$\xi = \frac{\nu_h Q_0 (a_{hi}, n_{hi})}{\lambda WD} \ldots\ldots [45]$$

$$q = \nu_h \sqrt{\delta_h} \frac{\frac{2\pi}{60} n_{hi} Q_0 (a_{hi}, n_{hi})}{4W\sqrt{\frac{W}{S}}} \ldots\ldots [46]$$

As to the variable C_z, its value is:

$$C_z = 16 \frac{W/S}{\delta_h (V_{hi})^2} \ldots\ldots [47]$$

We have supposed that the law of variation of ν with h is known. In reality, the tests just indicated permit its exact determination.

It should be noted that the functions $\tan \varphi'$, ξ, and q of C_z are in horizontal flight, independent of W and h, that is to say, of the load on the plane and of the altitude of flight. One can then determine these functions by tests in flight near the ground ($h = 0$), for which $\nu = 1$.

By repeating these tests at any altitude and at any load whatever, the relations [44], [45], and [46], in which all the elements are known except ν_h, will serve

1 To determine ν_h
2 To determine which law gives the closest value for ν_h and especially if this coefficient can be expressed exclusively in terms of δ (or μ); and
3 To determine, eventually, if ν_h does not depend in a certain measure not only on h but also on a and on n.

It will be noticed that the tests mentioned are extremely simple and can be accomplished in the course of a single flight, comprising four or six runs of a number of kilometers at three or four different levels of altitude. Standard instruments only are required. It will be sufficient to check on their precision and to choose registering types with an arrangement for synchronizing their readings. For recording the variations of a, the most practical method will consist in mounting on the carbureter a simple and inaccurate arrangement for indicating the position of the throttle valve. Block tests of the engine should be made, using the same instruments and reproducing the indications registered in the course of the flight.

The preceding tests give, in terms of C_x, the functions tan φ', ζ, or q which pertain to the efficiency of the propeller. It must be recognized that these are overall coefficients of the plane-propeller system which alone determine the performance and are consequently the most interesting to measure and learn directly.

Nevertheless it might be desired to separate the influence of the propeller from that of the airplane structure. In that case it would be necessary to trace the polar diagram (C_x, C_z) of the airplane alone. This is possible, either from the model tests in the tunnel or from gliding tests with propeller stopped or rotating without producing traction, tests which permit the finding of $C_x/C_z = \tan \varphi$ in terms of C_z.

Determination of the Ceiling by Flight Tests

The method just indicated makes it possible to determine the function ζ, and therefrom, by some simple tests in flight, the angle of incidence at the ceiling.

Equation [14] gives then, directly, the theoretical ceiling (H) sought. But one can, in the tests considered, show a characteristic relation directly connected with the ceiling H of the machine.

We shall now develop this method of determining the various influences on the part which the power reserve plays in the possibility of a plane's reaching a high altitude.

Let us consider a test in horizontal flight at an altitude h, at the incidence i, and with weight W. Let V_{hi}, n_{hi}, and a_{hi} be the values of speed, number of revolutions, and throttle setting recorded in the course of this test.

The power furnished by the engine in this flight and run is

$$\nu_h \cdot 2\pi \frac{n_{hi}}{60} Q_0 (a_{hi}, n_{hi}) = WV_{hi} \left(\frac{C_x}{C_z} \cdot \frac{1}{\eta} \right)_i \quad [48]$$

which can be considered as known, if the law $\nu = f(h)$ is known.

At the incidence i, the plane can only fly at a certain altitude H_i which can be considered as the ceiling of the plane for this angle of incidence. The theoretical ceiling H that we have studied so far is the maximum of the different ceilings H_i and corresponds to a certain angle of incidence i_H.

The ceiling H_i is determined by the following equation, analogous to Equation [9]:

$$\frac{1}{\nu_{H_i}} = \frac{Q_0 (a_m, n_{hi})}{WD\xi_i} \quad [49]$$

Let us put:

$$X_{hi} = \frac{Q_0 (a_m, n_{Hi})}{Q_0 (a_{hi}, n_{hi})} \quad [50]$$

which can also be written:

$$X_{hi} = \frac{Q_0 (a_m, n_{i0})}{Q_0 (a_{hi}, n_{hi})} \cdot K_{i0} \quad [51]$$

by placing

$$K_{i0} = \frac{Q_0 (a_m, n_{Hi})}{Q_0 (a_m, n_{i0})} \quad [52]$$

$n_{i0} = \sqrt{\delta_{Hi}}\, n_{Hi}$ designating the number of revolutions of the engine in horizontal flight at the incidence, i, at the ground. The coefficient K_{i0} is in general approximately equal to unity, the equivalent torque at the ground at full admission $Q_0(a_m, n)$ being seemingly independent of the number of revolutions.

The coefficient X_{hi} thus defined is the ratio of the equivalent torque at the ground which the engine develops, on the one hand at full admission and at the number of revolutions n_{Hi} at the ceiling H_i corresponding to the incidence i, and on the other hand to the admission a_{Hi} and to the number of revolutions n_{hi} actually used on the test under the conditions (h, i). It is this that we call the coefficient of excess of power at the altitude h and at the incidence i.

From Equation [48] also,

$$\nu_h Q_0 (a_{hi}, n_{ai}) = WD \left[\frac{C_x}{C_z} \frac{1}{\eta} \frac{V}{\frac{2\pi}{60} nD} \right] = WD\xi_i \quad [53]$$

and consequently the coefficient X_{hi} defined by [51] can be written:

$$X_{hi} = \frac{Q_0 (a_m, n_{Hi})}{WD\xi_i} \nu_h$$

At the ceiling H_i corresponding to the incidence i from [49],

$$\frac{1}{\nu_{Hi}} = \frac{X_{hi}}{\nu_h} \quad [54]$$

From this it results that the relation X_{hi}/ν_h, equal to a number which depends only on i, is itself independent of h and depends only on the incidence i.

Thus it is seen that the coefficient of excess of power X_{hi} reproduces, by a nearly constant factor, $\frac{Q_0 (a_m, n_{Hi})}{WD} \nu_h$, the variation of $1/\xi$ in terms of i. For a given value of i, its value is proportional to ν_h.

If, following tests such as those examined, the curves X_{hi} are plotted in terms of i for various altitudes h, similar curves will be obtained which are related to each other in the same way as curves of the corresponding values of ν_h. The comparison of these curves will allow both the determination of the variation of ξ in terms of i (or, if one prefers, of C_z) and of ν in terms of h. This process corresponds, moreover, to that which has been indicated above under another form. Let us return now to the test on the level under the conditions (h, i) which furnish, through Equation [52], the value of X_{hi}, when k_{i0} is known. In fact putting $k_{i0} = 1$ will correct this value after having determined, with the value of X_{hi} thus obtained, a value near to the ceiling H_i.

The law $\nu = f(h)$ being supposedly known, the relation [54] will furnish the ceiling H_i corresponding to the incidence i. The following formulas can also be used:

1 If we take for ν the standard law [2], the ceiling will be determined by the following approximate formula, which holds when h and H_i lie between 2000 and 7000 meters.

$$H_i - h = 18{,}090 \log X_{hi} \text{ (for } h < 7000 \text{ m.; } 2000 \text{ m. } < H_i). \quad [56]$$

2 If we adopt for ν the law [3] proposed up to 10,000 meters, the ceiling H_i will be determined by the following accurate formula:

$$H_i - h = (44{,}250 - h)\left(1 - \frac{1}{X_{hi}^{0.191}}\right) \text{ (for } H_i \text{ and } h < 10{,}000 \text{ m.)} \quad [57]$$

The theoretical ceiling H or maximum of the various ceilings H_i corresponds to the case i_H of horizontal flight for which, at every altitude, X_{hi} is a maximum.

If one takes for altitude h the sea level, the maximum of X_{hi} becomes equivalent to the coefficient X_H defined by [15], that is,

$$(X_{hi})_{max} = X_H \text{ for } h = 0$$

In this case Equations [56] and [57] coincide with [16] and [17] which give H in terms of X_H.

The coefficient X_{hi} thus generalizes, for one altitude and any incidence whatever, the idea of the coefficient of power reserve introduced by the coefficient X_H for the particular case at sea level and at the incidence i_H corresponding to the theoretical ceiling H. Equations [56] and [57] indicate, in terms of X_{hi}, the additional altitude $(H_i - h)$ which the plane can attain starting from h level and for the case of incidence i. They show how this "relative ceiling" is connected with the coefficient of excess of power X_{hi} corresponding to the altitude at departure h and at the incidence i.

Variations of Ceiling Under Various Influences for a Plane With a Normal Engine or an Undercharged One

1—Influence of a Variation of Load

Consider two planes 1 and 2, differing only in the total load, and distinguish by the indices 1 and 2 the relative sizes of these two planes.

The ceiling H_1 of the first is given, from the relation [9] by

$$\frac{1}{\nu_{H1}} = \frac{Q_0\,(a_m, n_{H1})}{W_1 D \xi_{H1}} \quad [58]$$

The ceiling of the second, H_2, is in the same way given by

$$\frac{1}{\nu_{H2}} = \frac{Q_0\,(a_m, n_{H1})}{W_2 D \xi_{H2}} \quad [59]$$

But, the planes, the engines, and the propellers being identical, the conditions at the ceiling are the same for the two structures and $\xi_{H1} = \xi_{H2}$.

Let us place

$$\frac{Q_0\,(a_m, n_{H1})}{Q_0\,(a_m, n_{H2})} = k \quad [60]$$

This coefficient k is practically equal to 1 and can be taken, for a first approximation, equal thereto.

From [58] and [59],

$$\frac{\nu_{H2}}{\nu_{H1}} = k\,\frac{W_2}{W_1} \quad [61]$$

Two cases are to be examined, namely, whether the relation W_2/W_1 has any value whatsoever or lies in the neighborhood of unity.

(*a*) *General Case: When W_2/W_1 Has Any Value Whatsoever.* If the standard law [2] is taken for the coefficient ν, it is easy to see that there can be deduced from [61] the following relation between the relative densities δ_{H1} and δ_{H2} at the respective ceilings H_1 and H_2 of the planes considered:

$$\delta_{H2} = \delta_{H1}\,k\,\frac{W_2}{W_1} - 0.099\left(k\,\frac{W_2}{W_1} - 1\right) \quad [62]$$

If we take the approximate law [3] of the variation of ν, accurate up to 10,000 meters, the relation [61] permits the following relation between the ceilings H_1 and H_2:

$$H_2 = 44{,}250 - (44{,}250 - H_1)\left(k\,\frac{W_2}{W_1}\right)^{0.191} \quad [63]$$

To calculate H_2 from [62] or [63] it is necessary to proceed by successive approximations, first making $k = 1$ and then correcting the value of k from the approximate value of H_2 thus obtained.

(*b*) *Particular Case: W_2/W_1 Differing Little From Unity.* In this case the variation of the ceiling is evidently small, so that we can take k equal to unity.

Using the approximate law [3], accurate up to 10,000 meters, [63] is simplified and gives

$$H_2 - H_1 = (0.0191\,H_1 - 8440)\,\frac{W_2 - W_1}{W_1} \quad [64]$$

Fig. 2, which represents this formula, gives directly the variation of ceiling $(H_2 - H_1)$ which causes a relative variation determined by the load. This relation is accurate and a close enough approximation up to $H_1 = 10{,}000$ m. and $\frac{W_2 - W_1}{W_1} = \pm 10$ per cent.

2—Influence of a Variation in the Torque

Let us consider two planes 1 and 2 differing only in their engines, which are assumed to give at full admission the equivalent torques Q_1 and Q_2.

The ceiling of the first plane is determined by

$$\frac{1}{\nu_{H1}} = \frac{Q_1\,(a_m, n_{H1})}{W_{H1} D \xi_{H1}} \quad [65]$$

and that of the second by

$$\frac{1}{\nu_{H2}} = \frac{Q_2\,(a_m, n_{H2})}{W_{H2} D \xi_{H2}} \quad [66]$$

The planes and propellers being identical, we have, by hypothesis:

$$W_{H1} = W_{H2}; \quad \xi_{H1} = \xi_{H2}$$

Let us place

$$\frac{Q_2\,(a_m, n_{H2})}{Q_1\,(a_m, n_{H2})} = k \quad [67]$$

the revolutions n_{H1} and n_{H2} corresponding to the ceilings H_1 and H_2 being determined by

$$\frac{n_{H2}}{n_{H1}} = \sqrt{\frac{\delta_{H2}}{\delta_{H1}}}$$

The comparison of [65] and [66] permits us to write, taking account of [67]:

$$\frac{\nu_{H2}}{\nu_{H1}} = \frac{Q_1\,(a_m, n_{H1})}{Q_2\,(a_m, n_{H2})} \cdot \frac{1}{k} \quad [68]$$

We distinguish again two cases, according as the relation Q_2/Q_1 of the torques considered has any value whatsoever or a value differing little from unity.

(*a*) *General Case: When Q_2/Q_1 Has Any Value Whatsoever.* If we adopt for ν the standard law [2], we can deduce from [68] the following relation between the relative densities δ_{H1} and δ_{H2} at the ceilings H_1 and H_2:

$$\delta_{H1} = \delta_{H2} \cdot \frac{1}{k} \cdot \frac{Q_1}{Q_2} - 0.099 \left(\frac{1}{k} \frac{Q_2}{Q_1} - 1 \right) \quad \ldots\ldots [69]$$

If we adopt for ν the approximate law [3], accurate up to 10,000 meters, we can deduce from [68] the following simple relation between H_1 and H_2:

$$H_2 = 44{,}250 - (44{,}250 - H_1) \left(\frac{1}{k} \frac{Q_1}{Q_2} \right)^{0.191} \quad \ldots\ldots [70]$$

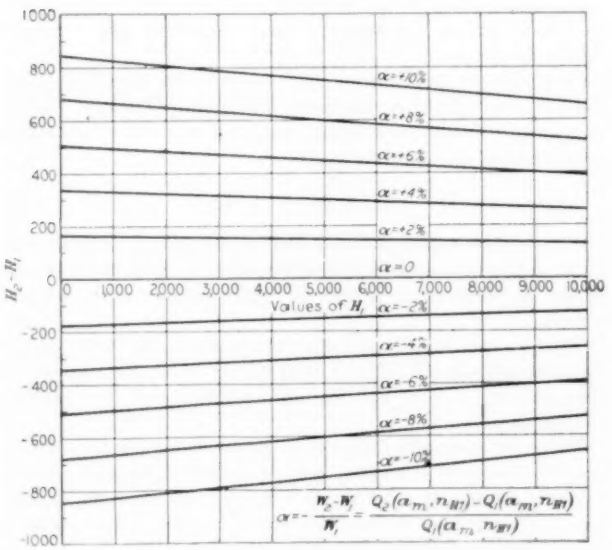


FIG. 2 VARIATION IN CEILING PRODUCED BY A SLIGHT VARIATION ($\leq$ 10 PER CENT) IN THE WEIGHT OF THE PLANE OR THE TORQUE OF THE ENGINE

To calculate H_2 by [69] or [70] it is necessary to proceed by successive approximations first making $k = 1$ and then correcting the value of k from the approximate value of H_2 thus obtained.

(*b*) *Particular Case: Q_1/Q_2 Differing But Little From Unity.* In this case the variation of the ceiling is evidently small and we can assume k to be equal to unity.

If we adopt the approximate law [3] accurate up to 10,000 m., the [70] becomes simplified and gives:

$$H_2 - H_1 = (8440 - 0.191\, H_1) \frac{Q_2 - Q_1}{Q_1} \quad \ldots\ldots [71]$$

Fig. 2, which expresses this formula, gives directly the variation of the ceiling ($H_2 - H_1$) which entails a relative variation determined from the torque. This relation is sufficiently accurate close up to $h = 10{,}000$ m. and $(Q_2 - Q_1)/Q_1 = \pm 10$ per cent.

3—*Influence of the Diameter of the Propellers and of a Reduction Gear Between Engine and Propeller*

Let us consider a plane which has been calibrated and examine the influence which a change of propeller diameter exercises on the ceiling when the engine is geared down, without changing the type of propeller—that is to say, its form and its relative pitch.

Let us utilize the ceiling formula [21] which we shall first put into a little different form. Assume the engine to be supplied with a reducing gear and let

N = number of revolutions of the engine crankshaft
P_{N0} = equivalent power of the engine on the crankshaft
η_G = mechanical efficiency of the reduction gear, assumed to be independent of the gear ratio
λ = gear ratio or ratio of the number of revolutions corresponding to n and N of the propeller and of the crankshaft.

Under these conditions, the equivalent power of the engine on the shaft of the propeller is

$$P_0\,(a,\, n) = \eta_G P_{N0} \left(a,\, N = \frac{n}{\lambda} \right) \quad \ldots\ldots [72]$$

Let us consider a plane equipped with a certain propeller 1. Let H_1 be its ceiling, corresponding to the range γ_{H1} of the propeller and to the number of revolutions N_{H1} of the crankshaft. From [21] and [72],

$$\frac{1}{\nu_{H1} \sqrt{\delta_{H1}}} = \frac{75 \eta_G P_{N_0}\,(a_m, N_{H1})}{4 W_{H1} \sqrt{W_{H1}/S} \left[\frac{\zeta}{\eta} \right]_{H1}} \quad \ldots\ldots [73]$$

Assume now that the propeller 1 of diameter D_1 is replaced by a propeller 2 of different diameter D_2. With this new propeller, the range γ_{H2}, identical with that of the ceiling with the first propeller, corresponds in horizontal flight to a certain value ζ_2 of the coefficient $\zeta = \frac{C_x}{C_z^{3/2}}$ differing from the value of ζ_{H1} which corresponds to the first ceiling.

At this range γ_{H1}, the plane equipped with the second propeller can reach a certain maximum altitude h_2 which is not its absolute but its relative ceiling for the range γ_{H1} considered for the propeller.

Suppose that the gear ratio λ of the engine is changed at the same time as the propeller and that it has a value λ_2 such that, at the altitude h_2 and for the range γ_{H1} of the second propeller, the engine turns at the same number of revolutions N_{H1} of the crankshaft as at the ceiling of the first propeller.

Finally, let us assume that the mechanical efficiency of the reducing gear and the total weight of the plane are not changed. The altitude h_2 or relative ceiling for the range γ_{H1} of propeller 2 is given by

$$\frac{1}{\nu_{h2} \sqrt{\delta_{h2}}} = \frac{75\, \eta_G P N_0 (a_m,\, N_{H1})}{4\, W_{H1} \sqrt{\frac{W_{H1}}{S}}\, [\zeta/\eta]_{h2}} \quad \ldots\ldots [74]$$

In comparing the altitudes h_2 and H_1, it is sufficient to notice that the ratio h_2/H_1 varies in the same sense as the ratio $\nu_{H2} \sqrt{\delta_{H1}}/\nu_{h2} \sqrt{\delta_{h2}}$, that is to say, by dividing [74] by [73], in the same sense as the ratio:

$$f = \zeta_{H2}/\zeta_{h2}$$

If ζ_{H1} is not a minimum, that is to say, if the case of ceiling with the first propeller does not correspond to the minimum of ζ, it is evident that ζ_{h2} can be made less than ζ_{H1} and consequently h_2 larger than H_1, that is, the ceiling of the plane can be raised by properly choosing the diameter of the second propeller. It is easy to see that:

If ξ_{H2} corresponds to a lift c_z less than that of the minimum

of ζ, it is necessary to increase the diameter D_1 of the propeller 1 to increase the ceiling;[4]

If ζ_{H2} corresponds, on the contrary, to a life e_z larger than that of the minimum of ζ, it is necessary to reduce the diameter of the propeller in order to raise the ceiling.

The conclusion is arrived at that the best propeller, for a given number of revolutions of the crankshaft at the fixed ceiling, and especially the maximum permissible for this number of turns, is that for which the ceiling corresponds to the minimum of ζ and for which, in consequence, the maximum efficiency corresponds at the same time to the said minimum. The best ratio λ of gear reduction for the engine results then from the value of the ceiling thus obtained.

Thus we find again the conditions set forth above relating to the maximum ceiling.

Ceiling of a Plane With Supercharged Engine

The preceding work is based on the assumption of a normal or undercharged engine.

It is interesting to show the influence which supercharging of the engine can exert on the ceiling.

Supercharging permits the prevention of the reduction of torque with altitude, that is to say, the modification of the law $\nu(h)$ of variation of the torque of an engine not supercharged. In particular, it allows the reestablishment of the torque Q_0 at a certain altitude.

The ceiling formulas given above are applied to the condition of giving to the law $\nu(h)$ the form corresponding to a supercharged engine, a form depending on the system of supercharging: namely, a mechanical compressor driven by the engine, or a compressor driven by an exhaust-gas turbine.

The study of the ceiling is then a case of the kind in which account must be taken of the increase of weight necessitated by the supercharging arrangements.

At any rate, supercharging permits a noticeable increase in the ceiling of an airplane. It necessitates, generally, changing the propeller in order to obtain the best utilization of the supercharged engine.

Limit of Ceiling

It is interesting to consider the limit of the increase of ceiling that can be obtained by supercharging a normal engine.

Let H_1 be the ceiling of the plane without supercharging. This ceiling, from [73], is determined by:

$$\frac{1}{\nu_{H1}\sqrt{\delta_{H1}}} = \frac{75\,\eta_{G1} Q_0\,(a_m, N_{H1})}{4\,W_{H1}\sqrt{W_{H1/s}}\,(\zeta/\eta)_{H1}} \quad \ldots\ldots [74]$$

If we assume that the engine and the propeller conform to these conditions, it is necessary that, at the ceiling H_1, the crankshaft shall turn at a speed N_{H1} equal to the maximum possible.

The supercharging limit that can be realized is that which enables the engine to give at the ceiling H_2 of the plane with a supercharged engine, the maximum power at the ground, viz., $Q_0(a_m, n_{H1})$, and consequently, to make the crankshaft turn at the same number of revolutions N_{H1} as formerly.

Let us distinguish by the index 2 the terms which characterize the flight at the ceiling H_2 of the plane with a supercharged engine.

The equation of power then gives for flight at the ceiling:

$$75\,\eta_{G2} Q_0\,(a_m, N_{H1}) = 4\,W_{H2}\sqrt{\frac{W_{H2}}{S}}\,\frac{1}{\delta_{H2}}\,(\zeta/\eta)_{H2} \ldots [75]$$

Assume that the total weight of the plane is the same at ceiling H_2 as at ceiling H_1. (Actually, however, the addition of the supercharging results in a certain increase of weight.) Dividing [74] by [75] we obtain:

$$\frac{\nu_{H1}\sqrt{\delta_{H1}}}{\sqrt{\delta_{H2}}} = (\zeta/\eta)_{H1}(\eta/\zeta)_{H2}\,\frac{\eta_{G2}}{\eta_{G1}}$$

If the propeller is chosen and adapted in the first case in a manner to give the best ceiling H_1, the case $(\zeta/\eta)_{H1}$ corresponds to the minimum of (ζ/η). Consequently, in order that H_2 shall be a maximum, it is necessary that $(\zeta/\eta)_{H2}$ shall also be equal to this minimum.

Finally let it be assumed that the reducers eventually used will have, in both cases, the same mechanical efficiency, that is, $\eta_{G1} = \eta_{G2}$. Under these conditions, the relation preceding is simplified and becomes

$$\nu_{H1}\sqrt{\delta_{H1}} = \sqrt{\delta_{H2}} \quad \ldots\ldots [76]$$

Such is the simple relation which determines the ceiling limit H_2 that the best supercharging can give a plane which has a ceiling H_1 when using a normal engine (of the same nominal power).

To show directly H_1 and H_2, assume up to $H_1 = 10{,}000$ m. the approximate law [3] of variation of ν for the normal engine, knowing $\nu = \mu$, and employ the formulas of the standard atmosphere.

The relation sought differs according to whether H_2 is less or greater than 11,000 meters, since, for this altitude the law of variation of δ and of μ in the standard atmosphere changes form. Calculation shows that $H_2 = 11{,}000$ corresponds to $H_1 = 3495$, or practically 3500 m. (12,250 ft.).

We thus obtain the following relations:

$$H_2 = 44{,}250\,[(1 - 0.0000226\,H_1)^{2.47}] \quad \ldots\ldots [77]$$

$$\text{within the limits } \begin{cases} H_1 < 3500 \text{ m.} \\ H_2 < 11{,}000 \text{ m.} \end{cases}$$

$$H_2 = 3300 + 215{,}560 \log \frac{1}{1 - 0.0000226\,H_1} \quad \ldots\ldots [78]$$

$$\text{within the limits } \begin{cases} H_1 = 3495 \text{ to } 11{,}000 \text{ m.} \\ H_2 > 11{,}000 \text{ m.} \end{cases}$$

Finally, when H_1 is small and less than 1000 m. (3280 ft.). Equation [77] can be replaced by the approximate formula:

$$H_2 = 3.47\,H_1 \quad \ldots\ldots [77a]$$

The results of applying these relations are summarized in the table at the top of the following page and shown in Fig. 3.

For an ordinary plane with normal engine, the maximum ceiling possible cannot practically exceed 10,000 m. It is seen, therefore, that the best supercharging obtainable will not make it possible to exceed altitudes of the order of 27,000 to 28,000 meters.

It must be remarked also, that the problem of the compressor establishing the ground pressure at an altitude of the order of 27,000 meters, at which the relative pressure μ of the standard atmosphere is equal to 0.018, does not appear susceptible of satisfactory solution in the present state of compressor technique.

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[4] It is possible to show very exactly that by an increase of diameter of a propeller placed ahead of the fuselage, as on the greater number of single-engined machines, that a slight increase of the effective efficiency of the propeller can be counted on for the same range $\nu = \dfrac{V}{\dfrac{2\pi}{60}\,nD}$ of operation because of the lessening of the interference between the propeller and the plane.

From another viewpoint, it is interesting to consider the problem of adapting the propeller to the higher ceiling.

H_1	H_2	$H_2 - H_1$	H_2/H_1
0	0	0	3.47
1000	3375	2375	3.38
2000	6560	4560	3.28
3495	11000	7505	3.15
4000	12170	8170	3.04
6000	16945	10945	2.82
8000	21970	13970	2.75
10000	27280	17280	2.73

Two principal cases can be considered:

1—Changing the Propeller Without Changing the Reduction-Gear Ratio. In this case the propellers of planes 1 and 2 turn at the same speed at the respective ceilings of the planes. The incidence of flight being supposedly the same, the speeds V_1 and V_2 are related by the equation:

$$V_1/V_2 = \sqrt{\delta_{H2}/\delta_{H1}}$$

The corresponding ranges γ_1 and γ_2 of these propellers are consequently in the ratios:

$$\gamma_1/\gamma_2 = V_1/V_2 \frac{n D_1}{n D_2} = \frac{D_2}{D_1}\sqrt{\frac{\delta_{H2}}{\delta_{H1}}} \quad \ldots\ldots\ldots [79]$$

At these ranges, the propellers must be considered by hypothesis as having the same efficiency:

$$\eta_1 = \eta_2 \quad \ldots\ldots\ldots [80]$$

Moreover, their thrust coefficients α_1 and α_2 ought, for the values γ_1 and γ_2 of the cases considered, to satisfy the power equation and this condition evolves, the c_x of the plane being the same:

$$\frac{\alpha_1}{\gamma_1^2} D_1^2 = \frac{\alpha_2}{\gamma_2^2} D_2^2 \quad \ldots\ldots\ldots [81]$$

It is necessary then to choose the type, the relative pitch, and diameter of the propeller 2 in such a manner as to verify in the range γ_2 defined by [79], the conditions [80] and [81].

It is not at all certain that this may be possible, particularly in the case where H_2 differs much from H_1.

Moreover, this propeller 2 is not longer adapted to the case of take-off and may be inefficient in this connection.

In any case one is led to consider here a propeller whose pitch or diameter will be variable during flight, or even one with pitch and diameter variable simultaneously during flight.

2—Changing the Ratio of the Reduction Gear Without Changing the Propeller. In this case, the conditions are as follows: At the ceilings H_1 and H_2 the speeds of the propellers are in the ratio $\frac{n_{H1}}{n_{H2}} = \sqrt{\frac{\delta_{H2}}{\delta_{H1}}}$ with the crankshaft turning at the same speed N_{H1} it is necessary that the ratios λ_1 and λ_2 of the reduction gears mounted on the normal and on the supercharged engine be such that $\lambda_2 = \lambda_1\sqrt{\delta_{H1}/\delta_{H2}}$. The customary gearing down of the normal engine can then be transformed into a gearing up for the propeller with supercharged engine.

But if λ_2 is fixed, the propeller 2 is no longer adapted to take off under the same conditions as propeller 1 and may become ineffective.

Thus one is brought to the idea of introducing gearing between propeller and crankshaft with a gear ratio that may be varied in flight, a problem which, in certain respects, is more attractive and perhaps easier to solve than that of the propeller with variable pitch and diameter discussed in the preceding case.

Undercharging and Supercharging

Supercharging (at an altitude) of an aviation engine, like undercharging (at the ground), increases the ceiling of the plane, and each method has advantages and disadvantages which limit the practical field of application.

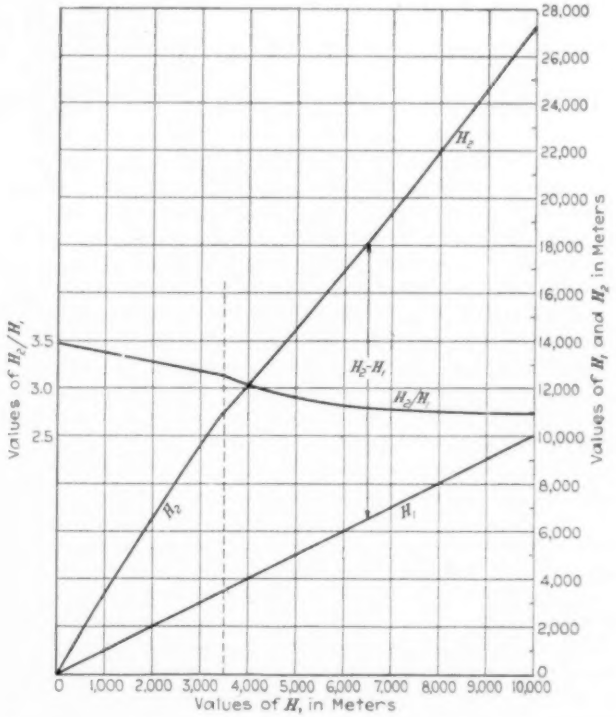


Fig. 3 Limit of Ceiling H_2 That the Best Supercharging Permits for a Normal-Engined Plane With a Ceiling H_1

If we undertake to solve the problem of superelevation of ceiling under the most advantageous conditions of lightness of weight and engine efficiency, we can consider a combination of these two procedures and this conception is capable of offering a real interest. In any case it permits us to deal with the very highest ceilings, with otherwise present unsolved accessory problems such as those of the variable-gear ratio, of the variable-pitch propeller, and of the cabin provided with an artificial atmosphere in order to maintain ground conditions.

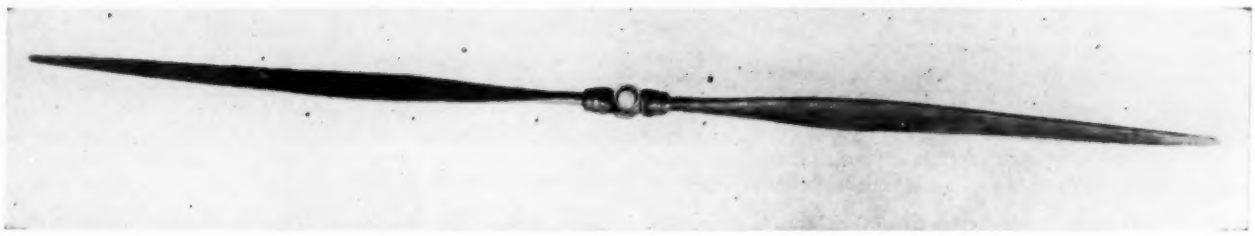


FIG. 1 FIRST DESIGN OF DETACHABLE-BLADE SOLID-STEEL PROPELLER (1918)

Development of the Drop-Forged Metal Propeller

BY FRANK W. CALDWELL,[1] PITTSBURGH, PA.

In this paper the author tells of various types of metal propellers for airplanes, and their advantages and disadvantages. These include a tubular type, a type built up of various structural members riveted or welded together, the plate type, and the drop-forged type which is the latest development. Hollow blades of aluminum alloy suitable for drop forging are used. The weight of these is slightly more than similar wooden propellers, and the limit of life has not yet been established.

MOST efforts toward the development of metal propellers previous to 1918 were concentrated along three lines: the tubular type in which efforts had been made to form a tube into a satisfactory form; the built-up type in which various structural and forming members were riveted, welded, or joined together; and the plate type, in which flat sheets were formed to provide blades for the propeller.

The tubular and built-up types had proved extremely expensive and had not been very successful in avoiding localization of stress leading to failure. These types did not appear to have great possibilities for economical production on account of the large amount of hand fitting necessary in their fabrication.

The plate type was not considered very promising on account of the low torsional rigidity of sheet and plate forms and the tendency of this form to flutter as the result of torsional oscillations.

During 1918 the author set out to develop a type of propeller suitable for drop forging. Considerations which led to the selection of this type were the relative economy of producing the desired forms by this method, the excellent properties usually obtained from sound forgings, and the freedom of selection of a form suited to the purpose.

In analyzing the design problem it appeared impossible to approach very nearly to existing forms of wooden propellers as the weight of the metal propeller would then be so great as to be out of the question. An obvious method of departure would be to thin down the propellers of the usual type with due regard to the various bending stresses and centrifugal restoring moments until the weight became comparable with existing wooden propellers. This would lead to a very thin propeller in which the bending moments due to the air pressures would have to be neutralized by the restoring moments of centrifugal force. A study of the torsional rigidity of such forms made it evident that a propeller of this kind would be very weak in torsion and that the resultant torsional instability would lead to violent fluttering.

The remaining method of approach was to decrease both width and thickness until the weight was as nearly comparable as possible with existing designs and to provide the blade area necessary to absorb the power of the engine efficiently by increase in diameter. Since the power absorbed by the propeller is proportional to the fifth power of the diameter, a relatively small increase in diameter would compensate for any deficiency in blade area.

This analysis led to the selection of a forged-steel blade of very high aspect ratio and great diameter with low camber ratios at the tip, but with comparatively thick sections in the inner part.

From the performance standpoint, it was anticipated that the following advantages would be gained in addition to the advantages of durability and dependability which were already appreciated. Obviously the larger diameter would act on a greater mass of air, and would consequently reduce the slipstream velocity and reduce the energy losses. This was already well known to the propeller designer from the momentum theory which is analogous to the conception of induced drag now made use of by the airplane designer. At the same time it was known in 1918 that the use of the thinner sections in the outer portions would contribute somewhat to the efficiency of the propeller due to the lower drag of these sections. It was not until 1920, however, that the importance of the thin sections was fully appreciated in making possible the use of higher speeds without burbling.

An analysis of the problem of a solid-steel propeller for the Liberty engine indicated that a four-blade propeller, 11 ft. 6 in. in diameter, could be built to a weight of about 110 lb., with an improvement in efficiency of about 4 per cent.

As this was a very novel departure in propeller design, it was decided to build two blades of this type for destructive whirling test. These blades designed for 200 hp. were run up to 800 hp. before failure, the failure finally occurring in the hub and not in the blades. The test was sufficiently successful to indicate that the general type of design should be developed further.

These blades showed a violent flutter at a speed of 800 r.p.m

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Presented at the Pittsburgh Aeronautical Meeting, Pittsburgh, Pa., March 12, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

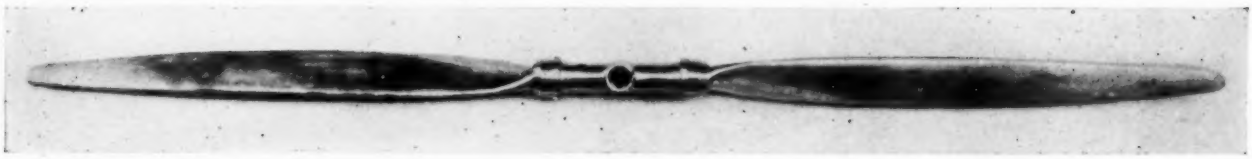


FIG. 2 FOURTH DESIGN OF SOLID-STEEL PROPELLER (1921)

to about 1400 r.p.m. Above this speed they were fairly smooth. The vibration at low speed was too bad, however, for actual use.

A second four-blade propeller of the same general design was next made up and run through a series of tests. This propeller failed in several places after a comparatively short test. The failure was traced to defective material. As the failure occurred in the outer part of the blades, they were gradually cut off until the blades were sufficiently rigid to run without vibration at all speeds.

It now appeared that the critical difficulty to be overcome in this type of propeller was the tendency toward torsional vibrations or flutter.

In order to design away from this tendency, a design was sought of reasonable weight in which the stiffness of the wooden

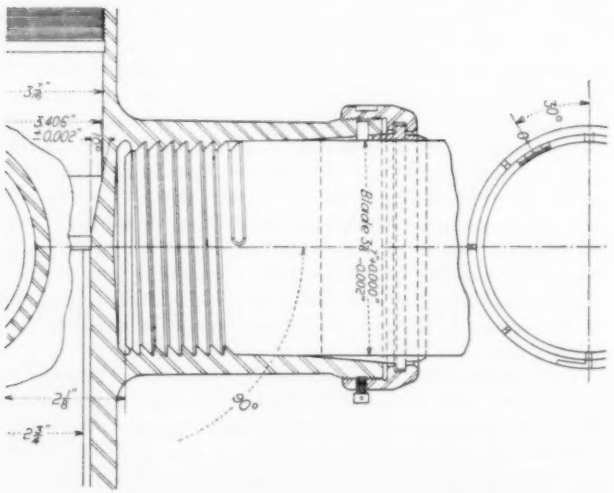


FIG. 3 THREADED TYPE OF HUB ATTACHMENT FOR SOLID-ALUMINUM-ALLOY BLADES

propellers would be approached as nearly as possible. A number of wooden-propeller designs were analyzed from the standpoint of stiffness, and a method of calculating defections under load was developed.

By applying this method to the solid-steel propeller designs, it was possible to approach sufficiently near to the current practice to be hopeful of eliminating the fluttering tendency. At the same time, an effort was made to localize the flexure near the tips, and to make the portion near the hub as rigid as possible.

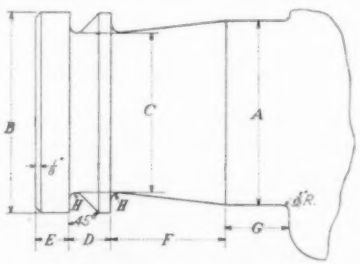
A new solid-steel propeller, the third step in the development was now built, and this propeller ran very smoothly at all speeds and passed a very satisfactory whirling test on the electric testing stand. As a further check before permitting it to be flown, it was placed on a very rough eight-cylinder engine and run until failure. Failure occurred at the point of attachment to the hub. The blades were attached to the hub by a threaded connection with a clamp adjacent to the threads. Failure occurred at the base of the threads.

The fourth step in the development of the solid-steel blade consisted in modifying the hub design to double the cross-sectional area of the blade at the threaded section, leaving all other parts of the blade the same. This blade again passed a satisfactory whirling test on the electric motor, but again failed at the root of the threads when run on the engine.

The last of these tests on solid-steel blades was run about 1923. During 1922, however, some tests had been run on hollow blades of an aluminum alloy suitable for drop forging, and it appeared that this alloy had very good mechanical properties. On account of the lower density of the aluminum alloy, its use made it possible to make blades of greater blade area and of greater stiffness than was possible with steel. It was thus possible to approach more nearly to the conditions existing in the best wooden-propeller practice.

In designing the first aluminum-alloy propellers, the deflections which would occur under load were again studied and were compared with the wooden-propeller practice and with the most successful of the solid-steel propellers.

By designing for a flexibility intermediate between that of the smooth-running steel propeller and that of the best wooden propeller it was felt that a serviceable design was fairly sure to



STANDARD BLADE END NO.	A	B	C	D	E	F	G	H
No.00-0 to 50 Hp.	2.250 +0.000 −0.001	2.695 +0.000 −0.003	2.000 ±0.010	0.500 +0.002 −0.000	1.0435 ±0.010	1.375 +0.005 −0.010	0.875 ±0.010	3/32
No.0-50 to 200 Hp.	3.000 +0.000 −0.003	3.245 +0.000 −0.003	2.625 ±0.010	0.6875 +0.003 −0.000	0.562 ±0.010	1.6875 +0.005 −0.000	1.250 ±0.010	1/8
No.1-200 to 475 Hp.	3.875 +0.000 −0.003	4.245 +0.000 −0.003	3.375 ±0.010	0.875 +0.005 −0.000	0.750 ±0.010	2.375 +0.005 −0.000	1.375 ±0.010	5/32
No.1½-475 to 800 Hp.	4.875 +0.000 −0.003	4.620 +0.000 −0.003	3.6875 ±0.010	1.0625 +0.003 −0.000	0.875 ±0.010	2.6875 +0.005 −0.000	1.500 ±0.010	5/32
No.2-800 to 1200 Hp.	4.500 +0.000 −0.003	4.995 +0.000 −0.003	3.875 ±0.010	1.250 +0.003 −0.000	1.000 ±0.010	3.125 +0.005 −0.000	1.500 ±0.010	5/32
No.3-1000 to 1500 Hp.	5.000 +0.000 −0.003	5.620 +0.000 −0.003	4.375 ±0.010	1.375 +0.005 −0.000	1.125 ±0.010	3.438 +0.005 −0.000	1.688 ±0.010	3/16

FIG. 4 STANDARD SERIES OF BLADE ENDS SHOWING DOUBLE SHOULDERS

be produced. At the same time, it was possible to hold the weight down to a figure only slightly above that of the existing wooden propeller.

It was also possible to use a diameter of the hub shank very much greater than had been possible in the steel propellers, so that the danger of breaking off at the threaded connection was greatly reduced. However, considerable study was given to eliminating any bending from the threaded portion, and finally a device was adopted as worked out by T. A. Dicks, who was then connected with the Standard Steel Propeller Company. This device consisted of a cylindrical wedge inserted between the blades and the hub at a point several inches out from the end of the threaded connection. It was hoped to absorb the drive through this wedge and eliminate the bending on the threads.

The first solid aluminum-alloy propeller designed and built along these lines was built by the Standard Steel Propeller Company for the Army Air Corps and from the author's design. It was completed early in 1923.

Whirling tests were started as soon as possible. The propeller was designed for the Liberty engine of 400 hp., and the whirling test was first run for 10 hr. at 600 hp., in accordance with the usual procedure for wooden propellers. At the end of this test, it was decided to continue the test to destruction if possible, and successive 10-hr. tests were run at 800 hp., 1000 hp., 1200 hp., and 1400 hp. As 1400 hp. was the limit of the power available for testing, the test had to be discontinued without doing any damage to the propeller.

Meanwhile an engine test was run with the propeller on the Liberty engine. This test was continued for 100 hr. without damage to the propeller, and it was considered satisfactory and that a serviceable metal propeller had been produced which could be built at a moderate cost.

Meanwhile, numerous flight tests were run on the propeller, and they showed a uniform gain in efficiency of about 7 per cent over the best wooden propellers in existence. These tests were run throughout 1923 and were supplemented by service tests in the Army and Navy and in numerous commercial planes. While these tests were quite satisfactory from the standpoint of service, a good deal of trouble was met in maintaining interchangeability of balance of blades and in replacement of blades in the hub, due to the threaded connection. It was considered desirable to maintain a neat fit of the thread due to consideration of strength. On account of the neat fit, small nicks in the threads or the presence of grit made it very difficult to remove the blades from the hubs.

In order to facilitate changing pitch and to improve interchangeability of blades, a split type of hub in which the centrifugal force of the blades was taken up by retaining shoulders was developed. At the same time a clamping ring was developed, serving the double purpose of clamping the two halves of the split hub together and compensating for small errors in vertical balance of the blades.

The first split hub developed made use of a single shoulder to take up the centrifugal force. While this type was tested at

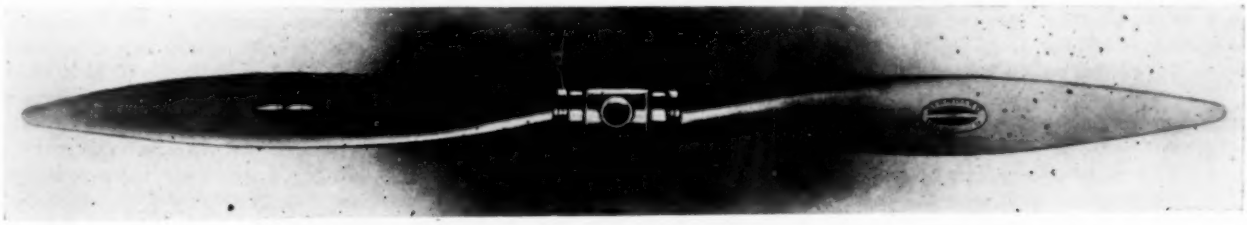


FIG. 5 STANDARD TWO-BLADE PROPELLER WITH ALUMINUM-ALLOY BLADES

overspeed with very satisfactory results, the stress analysis indicated that the shearing strength of the blade shoulder was not equal to the tensile strength of the blade root, so that the design was not well balanced from this standpoint. Consequently, a second design was developed in which the shear, crushing, and tensile strengths were equalized.

In order to check the strength of the three designs, full-size test pieces of each were made up and pulled in a testing machine with the following results: The threaded type failed in combined shear and tension at a load of 250,000 lb.; the single-shoulder type failed in shear at a load of 220,000 lb.; the double-shoulder type failed in combined shear and tension at a load of 320,000 lb.

As the double-shoulder type was obviously stronger in addition to its other advantages, it was adopted as standard, and a series of blade ends to take care of the requirements of various sizes of blades was developed from it. The efficiency of this type of connection is well illustrated by the fact that about 20,000 blades have been put in service during the last seven years, and in spite of severe abuse, none has ever pulled out at the hub.

Following the solution of the technical difficulties in producing an efficient design of moderate weight and satisfactory factor of safety, it was necessary to work out a series of designs in order to take care of the whole range of aircraft requirements from small propellers 6 ft. in diameter up to the large geared propellers ranging up to 16 ft. and for a range of from 60 to 800 hp.

In order to make this possible the range of propeller designs



FIG. 6 STANDARD THREE-BLADE PROPELLER WITH ALUMINUM-ALLOY BLADES

was divided into series which could be made from each die by modifications of blade width and diameter, and by setting the pitch. Fortunately it was found from model tests in the wind tunnel that the efficiency of the propeller was not appreciably reduced when the pitch was altered from the designed value by rotating the blades in the hub sockets. Consequently it was possible to produce a wide range of designs without an excessive number of dies so that the cost of individual designs has been held within the range of practical requirements.

Methods of inspection have been improved continually. At the present time all finished blades are etched in caustic soda and cleaned with nitric acid to permit inspection for cracks or forging defects. The quality of the forgings has been maintained at a very high standard and a comparatively small percentage of rejections is necessary due to forging defects. As in all metal parts subjected to dynamic stresses, fatigue has to be guarded against, and it is now recommended to all operators that propellers be etched and inspected after each 100 hr. of flying.

The limit of the life of blades has not yet been established. No doubt limits will have to be set on the number of hours which a propeller may be used. Probably this limit will have to be varied according to the design and the type of service.

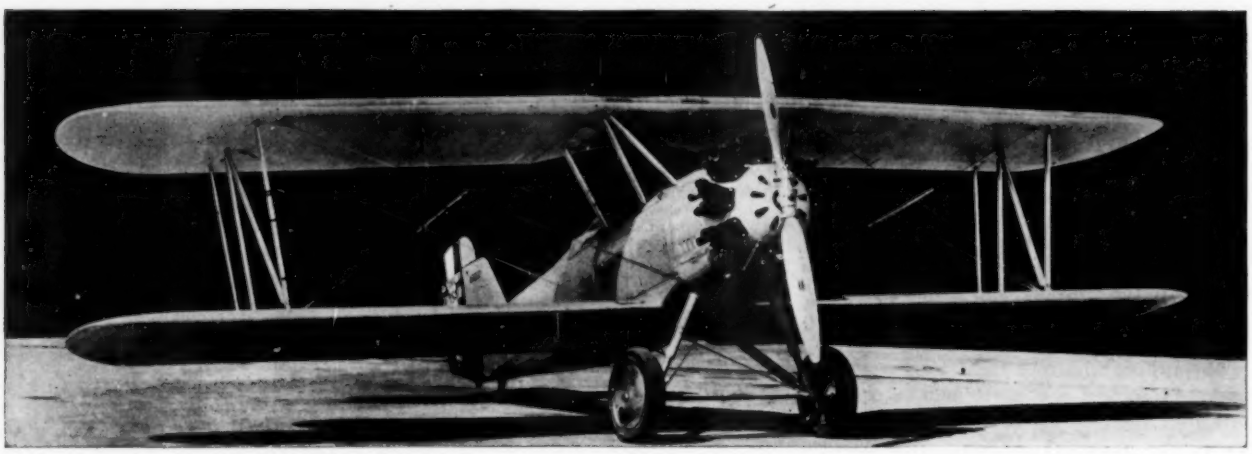


FIG. 7 STANDARD PROPELLER INSTALLED ON PLANE

A very troublesome problem has been the question of vibration. Vibration may be caused by unbalance of the propeller, by irregular form of the sections, by unequal angular setting of the blades, or by dynamic unbalance caused by errors in the track or alignment of the blades. These factors can be and are carefully controlled by inspection.

One of the most troublesome causes of vibration, however, is a periodic vibration caused by synchronization of the natural periods of the engine mount. Frequently this period can be changed by the selection of a propeller of a different size, but usually it is a matter that has to be taken care of by the airplane designer.

AER-52-10

Engineering Problems in the Operation of Air Transports

By FRANK R. CANNEY,[1] SEATTLE, WASH.

The author gives a masterly summing up of the hundred or more points that must be observed in the planning of a successful air-transport system, and reveals that there has been necessary only a slight modification of the original design of the Boeing transport plane as the result of the experience gained through its operation.

IT HAS BEEN found that, in an airplane, there are several major considerations, much the same as in an automobile or motor coach. In the first place, "power to go" is essential. This feature is quite heavily emphasized by makers of modern automobiles and is required to an even greater degree in the modern airplane.

Consider the "efficiency" and "reliability" of the power plant. It is necessary that the engine weigh as little as possible with respect to the power output, for each bit of weight added to this unit, beyond that required for reliability, displaces just that much pay load. The fuel economy is also an important factor, and in modern aviation engines should run very close to $^1/_2$ lb. per hp-hr. In considering the subject of fuel consumption, it becomes necessary to determine just how much shall be carried. This, of course, depends upon the route to be flown and distance between landings, and there should be provided a good reserve for getting around bad weather. The average time between regular stops on the transcontinental mail route is well under $2^1/_2$ hr., but against a head wind or in bad-weather conditions, it has been found desirable to have a fuel supply for better than 5 hr. Normally this will give a cruising radius of about 550 miles, but on one occasion at least it has been found necessary to refuel twice on a flight taking normally but $4^1/_2$ hr. This was on a trip from Cheyenne, Wyo., to Salt Lake City. The author found as much as 1 hr. and 55 min. difference in flying time required to make the trip from Oakland, Calif., to Salt Lake City, Utah, on two different nights of the same week. This, of course, was due to weather, the longer time being required when it became necessary to climb 16,000 ft. to avoid clouds.

After determining the amount of fuel required, or fuel per engine in the case of multimotored ships, other features of the power plant are considered. The installations of water-cooled engines, of course, run to more weight than for the air-cooled type, and in all of the Boeing transports the latter type has been adopted. This step also eliminates one complete set of plumbing from the power plant, although it may possibly complicate the oil system somewhat.

Without some auxiliary form of heat control, the air-cooled engine will not operate satisfactorily under the great variety of weather conditions encountered along the transcontinental route. For example, take the western end of the route from Oakland to Elko, Nev. The temperature at Sacramento, Calif., may be as high as 80 to 100 deg. fahr., while in Elko, only 3 or 4 hr. away, temperatures well below zero may and do exist simultaneously at the altitudes traveled there. It is for this reason that oil tanks are lagged with asbestos to retain heat, and at the same time a controllable shutter is provided in the nose of the engine cowling to cool the crankcase or not, according to what is desirable at the moment. Other items made necessary by sudden temperature changes are oil coolers with automatic bypass control and controllable air-intake heaters to prevent the formation of frost and ice within the carburetor. This last cause alone has been responsible for numerous forced landings prior to the installation of the heaters. The source of heat in this case is a small steel air scoop which is attached along the trailing edge of the round exhaust collector ring. It employs an air scoop at the upper end of either half, and serves as a streamline to the collector ring, which, due to its round section, offers considerable head resistance. A valve box in the outlet leads causes the warm air to flow either into the carburetor or to spill out as desired.

Aside from power-plant considerations, the weather affects the design in several other ways. Frost forms quickly on the windows under certain conditions, making it necessary to have them installed in the control cabin so as to be readily opened. In the first three-engined Boeing ships, these windows were designed to slide in felt-covered rubber channels, but they were frozen tight in the closed position so quickly on several occasions that it became necessary to break them out. To avoid the recurrence of this situation, they are now installed as is done in some automobile windshields—that is, with horizontal hinges and worm-operating mechanism for the windshield section, and gear-operated levers for raising and lowering the side windows. The side windows may be left open under most conditions without discomfort to the crew, as the V-shaped windshield deflects the air slightly away from the sides and causes the air to flow outward through these windows.

The angles at which the plates of glass are placed need careful attention for night flying, as lights on one side of the course may be reflected in such a way that they apparently are on the opposite side, or in most any direction. In flying among broken clouds, when only a few of the beacons may be "picked pp," this is rather disconcerting.

The pitot-static head of the air-speed indicator will freeze up in damp air at certain temperatures, but electrical heating elements are now provided to clear these instruments and they operate very nicely. Trouble has been experienced with freezing up of the venturi, which is used to operate turn indicators, but this difficulty was overcome by mounting the venturi directly on the exhaust manifold.

Drinking- and washing-water supplies are carried in tanks to the rear of the cabin just inside the fabric cover in a rather exposed position, and during cold weather may have to be drained and the water be carried in thermos containers. All water in the tanks and pipes was found to be frozen one night after a climb to 16,000 ft. where temperatures in the neighborhood of 35 deg. below zero were encountered in avoiding frost and snow clouds at lower levels. The tanks and lines are all made of annealed aluminum, and no harm was done, but neither was any water available.

The elimination of structural failures in the air has been quite complete where radically new types of construction are not

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Presented at the Seattle Aeronautic Meeting, Aeronautic School, University of Washington, Seattle, Wash., May 28 and 29, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

employed and where only materials of known strength values are used. Of course the utmost attention to details and a careful inspection of parts and assemblies as to quality of materials and workmanship are required. Careful consideration must be given to flexible members in which destructive vibration may occur, and often the direction at which strut and wire hinge pins are set is determined by the deflection of the members in question. Where pins and bolts are subject to wear they must be made well oversize, or provisions made for ready and frequent replacement. The general trend now is toward the employment of ball or roller bearings, even at the sacrifice of considerable weight.

The necessity of having easy access to structural or other parts which are subject to wearing out or failure due to fatigue and other causes was responsible for leaving such parts exposed as much as possible in the original transports, but as the speed requirements are increased, it becomes necessary to "cowl in" more of these parts. It is then necessary to provide access doors for the frequent inspection of such parts, and sometimes they are quite easily overlooked. For example, on the present job no inspection door in the landing struts was provided until it was found that the springs used in the oleo shock-absorbing units failed quite regularly. Such a failure is difficult to detect without a visual inspection, but has proved expensive on a few occasions when the rather costly oleo units suffered damage.

Other design conditions are rather difficult to foresee. In the horizontal tail surfaces of the original transport, the air flow around the tip of the stabilizer was such that it spilled on to the balancing surface at the tip of the elevator so as to cause considerable vibration of that part with subsequent excessive wear on adjacent hinges. This was corrected by changing the plan of the surfaces somewhat. Much trouble also was experienced with the tail-wheel installation. Under ordinary conditions it seemed to function satisfactorily, but on muddy fields when one of the main wheels dropped into a chuck hole, the result was to pick the tail wheel up off the ground and impart a rather sharp turning effect to the whole airplane. The resulting side-swipe caused when the tail wheel dropped back into the mud was too much for the original wheel-supporting structure. It became necessary to increase the length of the caster arm in the wheel fork and install low friction bearings to insure quick and easy following action of the wheel.

The percentage of structural failures is becoming very small as time goes on, and forced landings on this account are practically nil on the organized air-transport lines of this country.

As is the case with automobiles, there are many times when it is desirable to stop "right now." Of course an airplane cannot stop as short as vehicles on the ground do at the present time, and may never be able to. Much depends on other considerations than the brakes and tires. A very fast ship, for instance, has a higher landing speed than a slow one. There seems to be a limiting ratio between the high and low speeds of any given ship, beyond which it is very difficult to go. Boeing transports are designed to land around 60 m.p.h. at sea level, however; so the main worry after that factor has been established is to get effective brake action.

The equipment procurable to date has not been entirely satisfactory for a number of reasons, mainly due, it is suspected, to the fact that airplane brakes are a rather new development, and the efforts to save weight have had a strong influence on their design.

The light steel liners, shrunk and screwed into the drums, are not very true and cause considerable efficiency loss due to spotty bearing. The woven-fabric linings used up to now have proved too soft, wearing out quickly, and are considerably affected by moisture.

The controls become rather long and complicated in large airplanes, and have so much drag that it has been found necessary to add return springs in the linkage for releasing the brakes. Rods give a more positive control than cables, or a hydraulic control is probably most desirable, but up to the present the hydraulic-brake manufacturers have not been interested in designing a system for installation in the transports. It is felt that for the company to undertake the design and construction of such a system for its own ships might be rather a costly proposition in time, money, and weight.

While the brakes now in use are of the so-called self-energizing type, the method used to apply the braking effort in small ships which weigh somewhere from 1 to 3 tons is not at all satisfactory for the large ships which weigh nearly 9 tons when loaded. Considering the fact that the brakes are just 20 in. in diameter and 2 in. wide, it seems surprising that they are at all effective.

In small ships the brakes are generally applied by simply tilting forward, with the toes, small pedals attached to the rudder stirrups. The effective lever arm of this pedal is only 5 to 6 in., with a movement at the tip a trifle over 3 in. Sometimes the pedals are reversed and the heel is used to apply the brake. The former type is used on the new Boeing transports, however, and it developed that longer pedals with more motion were required. Both the length of pedals and the amount of movement at the tips have been just about doubled. This makes them rather awkward to use, but it appears to be a necessary evil.

Two other more satisfactory types may be employed, but they have the disadvantage of being hand-operated. First, the "joy-stick" type, which is a hand lever similar to an automobile parking brake, except that it tips to the sides as well as moving straight back. Second, a straight-pull handle with ratchet for setting it to any desired tension on the brakes.

Steering of the airplane when on the ground was done with the air controls until the adoption of brakes; but now, by using a free-swiveling tail wheel and differential braking between the two main wheels, control very comparable to that effected in an automobile is attained. The means for securing the differential brake action through the devices described previously are quite simple. With the pedals, there is simply applied to one pedal a heavier load than to the other. As each pedal controls but a single brake, the resultant differential effect is obvious. With the "joy-stick" type of lever control, the connections are made in such a way that a pull straight back applies both brakes equally, while tipping it to one side applies a greater load to the brake on that side of the ship. In the last-mentioned type, where a ratchet handle is employed, any desired equal tension may be applied to both brakes through the handle. In this case, the steering is effected through the rudder controls, which are connected to the brake linkage in such a way as to give a differential effect, although they are not actually used to apply the loads.

The "joy-stick" type of control was used in the first transports and was quite effective, except that it required the use of hands which could be employed to considerable advantage at other tasks. The types used on the newer ships, although probably somewhat of an improvement, still leave something to be desired.

Good vertical control, or "steering" in the air, under normal flying conditions is quite easily effected by means of an adjustable stabilizer. This may be set for level flight, climb, or descent, and to offset changes in the center of gravity location due to different loading conditions or the movement of passengers back and forth in the cabin. The main requirements for the stabilizer-adjusting mechanism are that its operating crank be in a handy location and that it may be easily and quickly used. This is especially important where the first attempt to land in a short field may be missed. The stabilizer which is placed at the full-down adjustment just prior to landing will cause the tail to drag low without excessive use of the elevators, unless it can be rolled back up,

immediately the decision is made to circle the field for another try at landing. It is necessary to employ low-friction bearings in this system to gain the operating ease desired. The elevator, which is used to supplement the effect of the adjustable stabilizer, is aerodynamically balanced to keep the loads reasonably light.

The effectiveness of the elevators was proved one day in a flight test, when, through the leaving out of a nut and cotter in the stabilizer-adjusting mechanism, a bolt fell out causing the stabilizer to drop about 6 deg. below its normal extreme downward position. The tail was kept up and a safe landing effected, although this happened very shortly after passing the end of the field on a take-off. Needless to say, the design has been modified to prevent a recurrence of such a failure.

The vertical fin and rudder, principally the latter, have given the most difficulty of any of the control surfaces on the new transports. Pilots invariably insist on a light rudder and one that "feels" right. To give them just what they want in these respects, and still retain effectiveness sufficient to offset the turning moment when one of the wing motors goes dead, is no easy task. First, a fin with an adjustable nose section was tried, but that method did not give the desired results. Next, a smaller fin was employed and the balance and area of the rudder were increased. This was an improvement over the first attempt, but with a balanced section large enough to retain ease of control with one wing-motor dead, directional instability resulted; that is, the rudder was slightly overbalanced and caused the ship to "hunt around" slightly. At the present time a rudder large enough to give ample control is being used, but with a balance just small enough to overcome the hunting tendencies. Rather heavy control loads result when a wing motor is throttled, so to offset this a spring device which may be quickly attached to absorb the unbalanced part of the load is employed. This installation is considered only temporary, however, and a "servo" rudder is being developed. In this, the rudder will be operated by an auxiliary surface, and it will be necessary for the pilot to operate only this much smaller surface. One can never be certain of how well new devices are going to work on an airplane, but it is felt that this last scheme will prove satisfactory.

Ailerons used are of the leading-edge balanced type, and function nicely without causing too much reverse yaw, even when it has become necessary to pick up a dead engine. Experienced pilots do not care to have a dead wing engine get too low, as that spells trouble. The original Boeing tri-motors employed a set of cams to give differential aileron action and thus eliminate yaw, but this device was found to be unnecessary on the later models.

Riding comfort must be considered for both the pilot and the passengers in a transport, the same as in a car, but as the driver of a car does not have quite as many controls to operate as a pilot, the problem of the pilot becomes more involved.

Comfortably upholstered seats are provided for both the pilot and co-pilot. The left hand, or pilot's, seat is also adjustable vertically through 4 in., as about this amount is required to give best vision to pilots of different heights. The rudder control stirrups are also adjustable toward or away from the seat by about the same amount. Even then some of the shorter pilots require an extra cushion at their back, while the taller men find their knees getting up into the way of the control wheel. As mentioned elsewhere, the windows or windshield may be opened for ventilation.

In grouping the instruments it is quite difficult to satisfy all pilots who will operate the ships. After sending out questionnaires to the operating personnel, almost as many suggested arrangements were received as there were pilots. The power-plant instruments, however, are all grouped on the center panel, with right-hand engine instruments to the right of the panel, and so on. The navigation and flight instruments are placed on a panel directly in front of the pilot with the exception of the compass, which cannot be accurately compensated when in this position. It is placed to the rear of the pilot, being read by means of a fixed mirror. All electrical instruments and the instrument and navigating light switches are placed on a panel in front of the co-pilot. This panel also carries a switch for turning off the cabin lights, the electrical water-heating element, and air-speed pitot heater. Provision has also been made here for the installation of radio controls, when that equipment is available.

The pilot and co-pilot may, and usually do, wear heavy fur-lined flying clothes when cold weather is to be encountered. Some heating is provided in the control cabin, but as it becomes necessary to open the windows for visibility, due to frost forming on them at temperatures around 30 deg. below zero and less, the heating system loses its effectiveness.

In these transports, designed to carry mail, express, passengers, and luggage, considerable thought was given toward getting what was felt to be the most practical and safest general arrangement. It was decided best to carry the heavy mail, express, and luggage loads ahead of the passenger cabin, where they would serve as a bumper in case of a crash. This location also prevents the possibility of such material coming forward on the passengers in case of trouble. Access to the luggage and express compartment is through a small door forward on the left side of the airplane. This is on the same side as the main exit door, and affords an opportunity for passengers to identify their luggage readily in the event it has not been checked. All mail is handled through the second or emergency exit door on the opposite side of the ship. As mail forms the bulk of the loads carried, this arrangement is proving quite practical, and keeps the trucks used in handling this cargo at the various fields in the clear and out of the passengers' way. The lower wings are covered with metal sidewalks out beyond the wing motors, and these walks are very useful in handling the mail, luggage, and express from truck to ship and vice versa, as the wings are at the same approximate height above the ground as the body floor of an ordinary truck. A locked compartment for the mail is located just inside the door, and an aisle past this compartment allows communication between the control and passenger cabins.

Overhead racks are provided in the main cabin for light parcels, wraps, hats, and the like, although a commodious coatroom with hangers is provided for heavier coats, etc., at the rear of the cabin.

A total pay load of approximately 3500 lb. may be carried by the new transports. With the mail loads running around 1600 lb., this leaves a balance of about 1900 lb. which may be used for passengers, luggage, and express.

The passengers have very good control over the heating and ventilation of the cabin. Windows are fixed and cannot be opened, but each is provided with an adjustable ventilator which provides a pleasant draft of fresh air at the will of the passenger. In addition to these, a large fresh-air intake is provided near the top of either side wall at the rear of the cabin. These have proved to be very effective as what flow of air exists in the cabin is forward. The heating system in this ship serves a dual purpose, being used as an air exhaust when the heat is turned off. This is accomplished by making the air-heater stove around the exhaust stack in the form of a venturi with a butterfly valve to the rear of the throat. The hot-air outlet from the stove is placed at the throat of the venturi, and when the butterfly valve is opened the resultant reduction in pressure causes a reversal of flow in the hot-air ducts. These ducts are placed along the full length of the cabin just above the floor at either side, exhausting at the rear.

To make the heating system effective, as well as to deaden the engine and propeller noise, some sort of cabin insulation must be provided. A wood-fiber wool insulator, of about 1 in. thick-

ness, was first tried and found to be very effective for decreasing the noise, but had the disadvantages of being too good a water absorbent and difficult to install in such a way as to prevent air leakage at the joints. On the latest transports a special plywood is used, the core of which is balsa wood of approximately $^1/_4$ in. thickness. This is proving to be nearly as good for deadening the sound, and far superior to the wood fiber as a heat insulator, as tight joints can be made readily. Further, it lends itself readily to giving a finished appearance inside the cabin. The inside-face ply is of highly finished mahogany, which in combination with the buffed aluminum and nickel fittings makes a very presentable decorative scheme.

Passengers' seats are comfortable and light, weighing only $13^1/_2$ lb. each. This weight does not allow much for upholstery, but does include an adjustable back which may be readily reclined at will, and a light safety belt. The seat framework is all of laminated walnut to match the interior finish, except the back frame, which is hickory for strength. All the seat fittings and back hinges are of aluminum alloy, and the seats are attached to the floor in such a way that they may be removed from the cabin in just a few minutes. Compared to the steel-frame seats in the original transports, they are more rugged, yet the weight has been cut to just one-half.

The seats are arranged in rows of three, two at the left of the aisle and one to the right. A window, 18 in. by 20 in., is provided in both side walls at each row of seats. The arrangement of the side body trusses was arrived at from this window arrangement, as it was desired to leave the window openings clear of structural members. Auto-type roller shades help to cut down the glaring light, which is rather objectionable to persons not used to it, and adjustable shaded lamps at each window provide illumination for night flying. Lights of 15 candlepower were found to be satisfactory for reading purposes, as the field of illumination is small.

For the further comfort of passengers, drinking water is provided at the rear of the cabin, as well as a coat room and a wash room. The latter is provided with running hot and cold water, with a soap dispenser in the wash basin. The basin is small but sufficient. The wash room is not cramped, being large enough to allow a good sized man to stand erect or move about at will. A dry chemical toilet is provided, and is so well vented that the reduced air pressure in the wash room is apparent upon opening the door. A clock, an altimeter, and an airspeed indicator, all with 6-in. dials, are placed on the wall at the front of the passenger cabin within easy view of all, so that those interested may keep a more or less accurate log of the trip.

In the cabins of the new transports a more rugged structure is provided than in the first model, and a considerable saving of weight results for the actual structure and lining of the body. While being theoretically sufficiently strong, the long slender steel tubular members of the original body design were much more flexible than the present larger aluminum-alloy tubes. In the newer structure there is no apparent weaving or vibrating of members, and there is no doubt that passengers will feel more at ease when there is no flexing in and out of the side walls, or weaving up and down of the ceiling. This feature, together with the rigid plywood lining, which replaces a fabric lining backed up with duralumin chicken wire, gives the cabin a much more substantial appearance.

In addition to refinements in flying equipment, there was at the beginning of last year, and still remains, much room for improvement in ground equipment. In making a trip over the line, from Oakland to Salt Lake City, in August, 1928, the ground equipment did not seem so important. A blowout experienced on the rocky field at Reno did not annoy much as the weather was fine, and an overnight delay, due to the airway not being lighted, was no cause for worry. Reno is a pleasant town in which to spend the night and all abroad enjoyed the layover. We were not getting any place, however, and that is the prime purpose of air transports, to go some place and to get there quickly.

Matters were considerably improved in this respect by February of this year. In making two round trips from Oakland to Salt Lake, at that time, the only delay experienced was at Elko, Nev., one morning when it became necessary to wait from 2 a.m. to 9.15 a.m. for weather reports. An attempt was made to start up at six o'clock for the purpose of seeing what the weather was like up above, but before the motors were sufficiently warmed up for taking off, a snow storm closed in, which lasted about two hours. The pilot then decided that we should take time off for breakfast and give the weather man a chance. Difficulty in securing weather reports is being eliminated, and at the present time the radio and teletype machines should be grinding them out with clocklike precision. It was no fun that morning last February, however, on an unsheltered field covered with snow, and the thermometer reading 4 deg. when we landed.

At Oakland, Reno, and Salt Lake comfortable waiting rooms are provided with sheltered concrete loading aprons, and soon similar facilities will exist all along the line. The arrangement at Oakland is especially attractive. There you enter the main lobby which is flanked at either side by the coastwise and transcontinental operating offices. A stairway leads up through two nicely appointed waiting rooms to a passageway across the main hangar. This passage is enclosed with glass, and the whole of the hangar interior may be seen without danger to the observer or interference with the work of the ground crew. A few steps down, at the far side of the hangar, and one is in the observation lounge. This is a very comfortably furnished waiting room where passengers and their friends may chat in comfort. It is on the second floor of the field post-office building, and through the large windows on three sides an unobstructed view of the entire airport may be had. Leading from this room are stairs to the field level, and a concrete walkway with telescoping cover gives safe access to the waiting ship, at the same time affording protection from the air blast of the propellers. Compared to other arrangements observed, this layout is a very good model, although an arrangement with the administration offices, waiting rooms, etc., in an entirely separate building, has also been nicely worked out.

In conclusion, it may be said that the fundamental conception of the Boeing transport airplane, as originally laid out, has been modified only slightly as a result of the experience gained through its operation. On the other hand, hundreds of minor alterations and additions to the original design were found necessary, and only through a long process of trial and error was it possible to develop a transport which could give real satisfaction in service. To emphasize this statement, it might be pointed out that since the first flights of this ship, there has been expended approximately 2000 "man-hours" of engineering work on the project.

Assume that the design is completed before testing begins. The indications are that, regardless of the previous experience and care exercised in laying out the details of the design, it will be practically impossible to arrive at the ultimate design before the model is actually placed in service. This is due, of course, largely to the requirements of that service which cannot be accurately predicted.

AER-52-11

Aircraft-Float Design and Construction

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Designs of aircraft floats have included the canoe type, inflated bag or skid, box type, multi-rib type, canoe system with cloth over a thin smooth skin, girder type, and all-metal construction. The author follows the development and discusses necessary characteristics. Late construction is principally of aluminum alloy for both floats and boats. Bulkheads are used to carry concentrated loads, and the bottoms are sometimes reinforced by exterior longitudinals. Extruded shapes, angles, channels, and tees are becoming popular as framing members.

AIRCRAFT-FLOAT design is a special form of naval architecture meeting many of the problems of surface craft and many problems in addition not encountered by such craft. These additional conditions, in particular, affect the details of design due to the variation of load with wing lift, and impose certain new requirements at what is known as the "hump" speed, and at the "get away" speed.

Aircraft-float construction has passed from the canoe and the inflated bag on a skid, through the box-type float, the multi-rib type, the canoe system of cloth over a thin smooth skin, to the girder type, with the longitudinal system of framing, and with veneer or two-ply bottom; from there to the metal floats of aluminum-alloy or nickel and monel-metal construction. Today both design and construction follow closely the practice in high-speed boat construction, but with greater stress being laid on a light structure of adequate strength.

Seaworthiness and airworthiness requirements must be satisfied to the highest degree practicable, but as these requirements conflict with each other compromises become necessary, and these compromises are also affected by the purpose for which the plane is to be used and the disposition of the crew. The nature of the compromise depends largely on the purpose for which the design is intended and the locale of operations, and the results depend also to a degree on the system of flotation adopted.

To follow through the developments from the first successful design of a sled-type float used by Curtiss in 1910–11 would in itself require a small volume. It may suffice to state that today there are certain arrangements and features which are well recognized and generally used.

As to arrangements, these group themselves into the float and boat types. In the float types there is a system of floats which in many types are substituted for wheels, so that the floats are associated with the usual fuselage type of plane. In the boat types the hull itself forms a fuselage, or is intimately associated with it.

Another classification might be based on the float arrangement, and divides into the principal groups of central-float and twin-float systems. The central-float system is one in which the central float is the main supporting member when afloat. For lateral stability it requires some additional means. This may be in the form of the usual wing-tip float, normally just clear of the water surface, or of larger side floats contributing to lateral stability and to buoyancy at rest through partial immersion, or it may be in the form of side sponsons to the central floats which are partially immersed at rest. These lateral stabilizers are normally well clear of the surface after, or even before, full planing is attained.

Still another classification applies to a system very generally used, known as the twin-float system. In this system the main supporting members comprise the pair of symmetrically placed floats which provide both lateral and longitudinal stability; though in some forms the floats are short and a tail float is used to prevent turning over backward. This latter form is considered obsolete, and in general the main floats of any system are depended upon for longitudinal stability.

Certain float features are found in almost all of the latest hull and float designs. It will be interesting to discuss these and to point out their general effect and the reasons for their incorporation.

In naval architecture it is customary to define the lines of a vessel by means of a body plan giving sections at various points along the length of the hull, a half-breadth plan giving the contour of the waterline at different distances above the keel, and a profile plan giving the contours of the deck and keel and showing what are known as buttocks.

The first successful type had the profile of a sled. The deck was parallel with the keel, and the keel rose at the bow in an easy curve to meet the deck line, while the deck line at the stern dropped in an easy curve to meet the keel. This type was of the simplest construction. It planed nicely, but after planing had been attained and the hull was gliding on the surface of the water it was difficult to depress the tail of the float, which necessarily was well behind the center of gravity. This type of float had, however, several deficiencies, the first one being that in order to make it possible to take the air without having too much difficulty it was necessary to keep the stern fairly close to the center of gravity and this permitted a very small margin against tilting over backward. Also the curvatures at the bow and at the stern, while providing a fairly efficient streamlined form for the float, also gave the float a nosing tendency when in the air. For smooth-water work this type of float was quite good, but if the water was rough the flat bottom caused considerable punishment to be sustained by the plane. The next variation of this type consisted of the introduction of steps in the bottom of the hull. Various attempts were made to determine how many steps would be useful, and model-basin experiments indicated that little or no advantage was gained by providing more than

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

two hydroplaning surfaces by the introduction of a step. It was found in these early days, where there was very little reserve of power, that there was some advantage of providing ventilation for the steps, and there still appears to be some possible advantage in providing such ventilation where the float is flat bottom or where the chine of the float is relatively deeply immersed initially. As the evolution proceeded, the profile of the keel gradually became changed, and the present forms practically all embody what may be termed the "rocker form of keel," embodying what might be still recognized as the sled type of profile from the bow to the first step. This step is usually located slightly aft of the center of gravity of the plane; abaft the first step, the keel rises at an angle of between 5 and 10 deg. to the keel forward to the step, and it is usual to have this part of the keel a straight line to avoid suction which would be present if curved lines were used. In the simple float construction, this defines the profile of the modern float.

In boat construction, however, it is necessary to provide a long tail to support the tail surfaces of the airplane, and in this case it is usual to find a second step at some distance behind the first step, and this is particularly necessary if the keel is curved from the first step to the stern; abaft the second step the keel usually rises sharply and very often the tail of the hull is lifted considerably in order to keep the tail surfaces well clear of the water.

The problem of "suction" is a very serious one, presented where too much of a "rocker keel" is involved or where the keel aft of the first step is curved downwardly, as the flow of water about such a curve tends to produce an important force acting radially outward from the center of curvature of the surface. Since water is over 800 times as dense as air, this force on a very small surface in the water may become important at airplane operating speeds. However, there are advantages to the "rocker" form of keel, provided steps are properly located to break the suction involved.

Floats with a deep rocker profile for the keel generally run hard at the "hump" speed; that is, their resistance at from 20 per cent to 30 per cent of the get-away speed is higher than profiles which are not so pronounced. If, however, there is ample reserve of power so that there is no question of getting by this "hump" speed, the deep profile floats have an advantage in that at the get-away speed they run with much less resistance than the more normal type of hull, and, in addition, after planing is attained and the plane is up on the step, they may be rocked back to get greater wing lift without again bringing the tail of the float into contact.

On the other hand, just before planing is attained the waves that are thrown off from the side of the hull converge under the stern of the float, and where these two waves come together a "roach" is formed, which may in some cases again make contact with the float, and may seriously endanger the tail surfaces of the plane unless they are located at a good distance above the water.

In plan form, as shown on the half-breadth plan, practically all floats are given a substantially streamlined form, with the major ordinates somewhat forward of mid-length. This is practically the only feature of a hull design which does not have to be a compromise in the attempt to arrive at the best aerodynamic form.

The major or midship section of a float usually occurs in the vicinity of the step, and it is at this point that the greatest variation is found in up-to-date construction. The flat-bottom section is practically obsolete. The V-bottom has come into very general use for the reason that it forms a simple and efficient shock absorber, and as far as planing properties are concerned it is substantially the equal of the flat bottom or the hollow bottom, or the inverted V-bottom. The V's themselves may have straight sides, but more generally have a certain amount of hollowness introduced between the keel and the chine. Experiments on the variation of this feature, some with flat bottom, up to V's of 25 deg., show that, so far as the horsepower required for a float is concerned, the difference is almost imperceptible in the results obtained from model tests. The flatter the bottom the greater the punishment, and as a rule the dirtier is the run up to planing speed. A 20- to $22^1/_2$-deg. bottom is quite satisfactory. In recent years there has been a strong tendency to make the bottoms quite hollow at the sides, and this has the advantage of holding down the spray, but toward the get-away speed the V's at the keel are so steep that they afford almost no planing capacity, while at the same time they cause considerable drag. This drag is not alone at the keel itself, but is at the tail of the plane, due to the narrow groove, plowed in the water, closing and forming a roach, which may again make contact with the tail in the neighborhood of the get-away speed. The hollow bottom and the inverted V-bottom plane well and run clean, but do not appear to have any advantage over the straight V-bottom, and do have the disadvantage that in rough water they pound rather hard, subjecting the plane to considerable punishment.

Since 1911 in this country and abroad, and notably in England, the development of aircraft floats has been due in a large measure to the use of the model basin, and recently in Germany the same thing has taken place in the development of the hulls of large seaplanes, and recourse to model-basin tests is recommended for any one starting out on a new plan of development. Naturally the conditions of a model test are to a degree artificial, particularly in that only straight-away runs can be made, and also that the wing lift, while provided for in the model test, cannot easily be made to vary as it does on the plane due to the changing attitude of the plane in the course of the run to take off. But, admitting these artificialities, it has been found that the results obtained from model tests are an excellent indication as to the possibility of flight with a particular design, and also as to the behavior which is to be expected of the full-size structure.

A comparison of a large amount of data from model-basin tests on the three principal types of construction—the flying boat, the central float, and the twin float—shows that at hump speed the efficiencies are substantially in the order named; that is, the resistance at the hump speed is less for the boat than it is for the central float, and less for the central float than it is for the twin float. The reason for this appears to be that generally in boat construction the ratio of the waterline length to the beam is lower than in the case of the central float, etc. In the case of the boat, great length is relatively easily obtained because of the size of the boat, and also because of the necessity of a tail for the float to support the tail structure, and this provides ample longitudinal stability, whereas in the central float relatively greater length is required for this purpose, and this is also in effect true in comparing the central float with the twin-float type.

The question of the determination of the beam for a float for a long time was a puzzling one to the author, and it is believed to others, for an examination of the data available did not appear to show any particular systematic method of variation. A number of methods which have been proposed were examined, but were not found to bear close analysis, although in many cases for the particular design under consideration they gave fairly satisfactory results, but, for instance, when it came to attempting to apply this method to the three cases just referred to—that is, the boat, the central float, and the twin float—it was found that the discrepancies were wide and that in many cases ridiculous proportions would result.

Finally, after considering a great many designs from many

sources, both American and foreign, it was found that the beam required is a function of the length-beam ratio, and that this became reasonably consistent if the length and beam of the submerged portion of the hull were taken into consideration rather than the overall length, and once this was discovered, it was found that good design conformed very closely to this criterion. It is not surprising that many designs which have been flown do not conform exactly to this because up to this time there has been no satisfactory criterion for the determination of the beam of a float.

It now appears that the beam of a float can be determined by the following equation:

$$B = \frac{KW^{1/3}}{(L/B)}$$

In this formula the beam is determined in inches, and the constant may vary from 16 to 18, and the L/B ratio may vary from 3 to 7. First hand, this may appear to be no determination whatever; but it is found in practice that the L/B ratio for floats is about $4^1/_2$, for single floats about $5^1/_2$, and for twin floats about 7, if good results are to be obtained.

The latest construction is principally in aluminum alloy for both floats and boats. In the boats the hull usually incorporates two deep girders in the sides or well out to the sides, and a central-keel girder and intermediate keelsons supported by floors from chine to chine and attached to the keel.

Watertight and partial bulkheads are used to carry concentrated loads. In some cases longitudinals appear only at the chine and keel, the bottom plating being generously supported by frames, floors, or bulkheads.

In floats the decks are usually of approximately circular form, supported by frames and reinforced by longitudinals either inside or outside. The latter are favored as they avoid cutting of frames or longitudinals at their intersections, and the plating forms natural gussets.

Bottoms are sometimes reinforced by exterior longitudinals, preferably solid, for hollow sections are subject to damage which may cause them to leak and convey water to remote portions. They are also thus subjected to severe corrosion in inaccessible places.

Castings are favored for stem and stern pieces and in some cases for steps, and are also favored as fittings where longitudinal members pierce floors or bulkheads, particularly if watertightness is required. They greatly simplify connections which would otherwise be intricate. Extruded shapes, angles, channels, and trees, with or without bulbs, are becoming generally popular as framing members.

In floats subject to salt-water operations difficulty is still found with rivet heads as the forming operations of the rivets appear to alter their galvanic properties, making them especially vulnerable to attack unless they are thoroughly protected. As rivets in bottom plating are particularly subject to abrasion, it is found that machine screws are preferable and bottom plating is now commonly so secured.

Corrosion is an important factor, good results being obtained by anodic treatment of the material or by the use of alclad, but in either case priming coats of iron oxide or zinc chromate are desirable for exteriors or interiors, to be followed by lacquer or enamel coatings of good quality.

Interiors of inaccessible spaces are best protected by bitumastic coating. In tropical waters anti-fouling over bitumastic has been found a desirable finish for the bottoms of floats which are moored out. As the size of floats increase, the principles of ship construction are being more and more incorporated in the design of the details of the structure.

AER-52-12

Installation of Air-Cooled Radial Engines

By P. B. TAYLOR,[1] PATERSON, N. J.

This paper tells of some of the installation problems of air-cooled radial engines, and the author points out some improvements that are desirable. Engine exhaust and cowling systems are described, and the purposes of carburetor air heaters and cleaners are given. Variation of the viscosity of oil with temperature, together with its low heat conductivity, causes many difficulties in the lubricating system. Oil foaming is another source of concern to designers. An effective method of preventing this is to discharge the oil on a plate sloping into the oil volume and allow it to run down the sides into the oil tank.

There are many types of fuel systems, and the author gives some general considerations that should be followed in installation. To provide for inverted flight requires special arrangements to give satisfactory results.

THE reliability of an air-cooled engine depends directly upon the amount of engineering put into its installation and the degree of intelligence (if any) with which it is serviced. At the present time, the airplane manufacturer is far more interested in aerodynamics than installations, and the engine manufacturer whose engine is installed in 100 different ships, all of which are rushed to completion, does not have the necessary control of the installation details. The result does not promote reliability, and is one great hindrance to selling aviation to the public. Department of Commerce statistics show at least 50 per cent of engine failures are directly chargeable to gas-, water-, and oil-line breakage, which can be charged to stupidity and carelessness only. If this industry cannot install a few pieces of copper pipe so that they will not vibrate and fail, there is little hope that aviation will become as safe as it is now considered.

Engine Mounts

Radial engines are almost universally mounted on a ring having from eight to twelve points of attachment to the engine crankcase. This ring is attached to the ends of the longerons by eight tubes meeting on the vertical and horizontal centerline of the structure (Fig. 1). The engine crankcase which is accurately machined on the mounting faces is usually called upon to straighten the engine mount which has been distorted in welding. The author fails to see the utility of the attaching points which are not adjacent to the struts leading to the longerons, as no stretch of the imagination will subject these points to more than engine-torque load which is 200 lb. per bolt in the case of a 600-hp. engine with a nine-bolt mounting which can be easily absorbed at four attaching points.

The logical engine mount for a radial engine is a structure consisting of eight tubes from the ends of the longerons which meet at four opposite points on the engine crankcase, at which points a substantial mounting lug is incorporated in the engine crankcase which, with the tubes, forms a complete structure (Fig. 2). To hold the ends of the mount in position when not attached to the engine crankcase, they may be tied together by light tubing. The crankcase is a far more rigid structure than a tubular ring and can be made quite adequate to withstand mounting strains.

No fear need be had for the overhang of an engine beyond the mounting point if properly designed, as the air-cooled V-1460 mounted on the stand indicates (Fig. 3). This engine has been run for several hundred hours, as illustrated.

The automobile industry has been forced to rubber or spring mounts for their engines, and since the vibration of the aviation engine is far worse than the automobile, the public will certainly demand shock-absorbing mounts as soon as it knows why a plane is so uncomfortable. The airplane has the advantage of a very elastic structure throughout which can be damped easily to prevent vibration from reaching the passengers. A flight behind an engine on rubber mounts will convince the skeptic of their effectiveness.

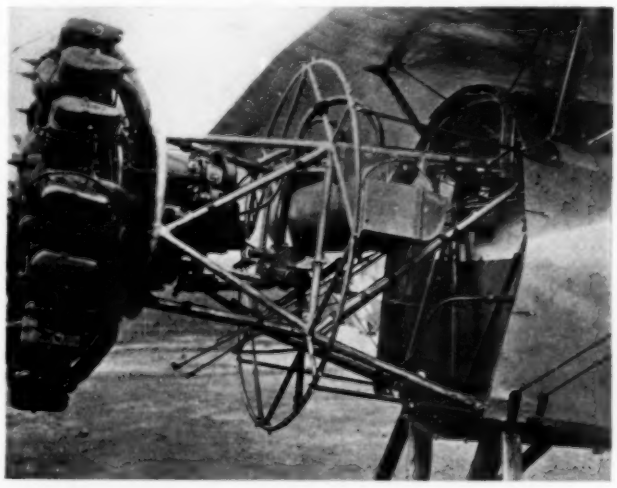


Fig. 1 Typical Radial-Engine Mounting With Four Triangular Structures Tied Into a Mounting Ring

Engine Exhaust Systems and Cowls

The simplest exhaust and cowling system is undoubtedly open stacks from rear exhaust-port cylinders. This in no way interferes with a cowl from the engine nose to the ship's lines (Fig. 4). For the past two years an effort has been made to improve this construction with both front and rear exhaust-collector ring, as open stacks often are disagreeable to the pilot and impair his vision at night. The front exhaust collector forms a neat installation, and due to its small diameter is lighter than the rear collector. Fig. 5 shows a typical installation. How-

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

ever, the rear collector can be made into just as neat an installation as the front collector. Figs. 6 and 7 illustrate such a manifold and cowl. The front collector allows the engine manufacturer to finish his part of the cowl in front of the cylinders,

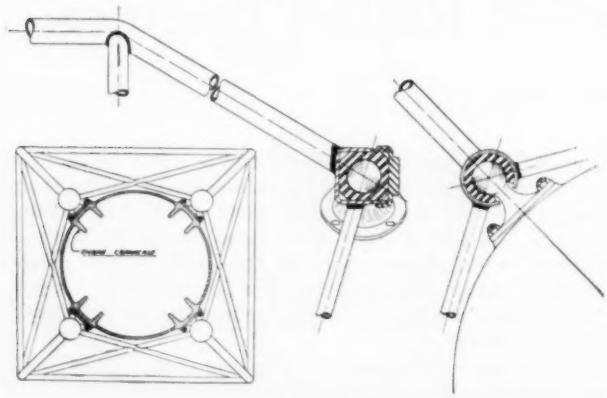


FIG. 2 FOUR-POINT ENGINE MOUNTING WITH RUBBER INSULATORS AT THE POINTS OF ATTACHMENT TO THE ENGINE CRANKCASE

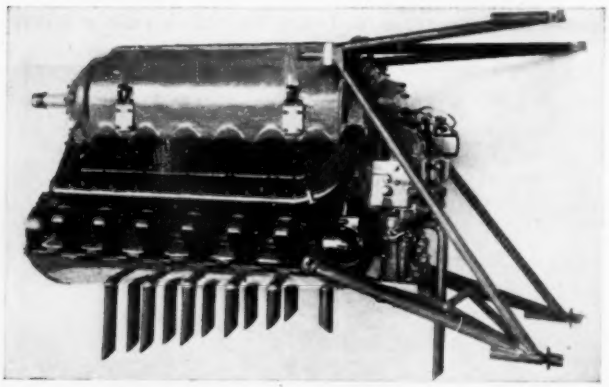


FIG. 3 WRIGHT V-1460 ENGINE MOUNTED AT THE REAR. THIS ENGINE HAS HAD SEVERAL HUNDRED HOURS OF TORQUE-STAND OPERATION MOUNTED AS SHOWN

that the engine manufacturer should complete the engine cowl and establish a ring behind the engine to which the ship's cowl will attach. If this dimension is once established, new ships can be made to conform without difficulty. In the case of the rear collector, cowl to the rear point of the exhaust system is essential in a well-appearing installation. From an aerodynamic standpoint, rear exhaust rings directly behind the cylinder have very little effect whether streamlined or not because the air flow over a radial cylinder is so turbulent that a small thing like an exhaust ring cannot make it worse. Fig. 8 shows the Pratt and Whitney Wasp Junior with such a ring.

Shutters in a nose cowl are in vogue at present, most of them adjustable by the pilot. Considering the cylinder barrels, these adjustable shutters are unnecessary since the barrel temperature of most engines will be satisfactory from 100 to 300 deg. fahr., which is a wide enough range to cover all operating conditions. The pilot makes his nose-cowl adjustment, if at all, from his oil temperature, which is a very approximate indication of barrel temperature. This subject will be discussed further under oil systems, and a different method of controlling oil temperature suggested. Nose shutters adjustable on the ground for seasonal temperature variations will simplify the duties of busy pilots without affecting the engine performance.

The principal function of standard cowl is to control the engine temperature since it has little or no effect on drag. This has been lost sight of by many plane manufacturers who accept the louvers in the nose cowl without providing a corresponding area of louvers behind the engine, and if cooling of the oil tank is desired, behind the oil tank. Air will not enter a cowl unless an exit is provided, and it will not pass over the oil tank unless the exit is behind this equipment. These principles are not at all profound but are still ignored in many installations.

Spinners were discarded several years ago because they were found to have little effect on drag and because they would not stay on. They cause a distinct improvement in cylinder cooling, however, as they mask the round section of the propeller which throws the air in a radial direction away from the louvers in the front cowl. It is hoped that if spinners are again supplied they will be provided with a continuous ring at their trailing edge with strength enough to hold them in one piece.

The N.A.C.A. cowl is most easily developed around front



FIG. 4 INSTALLATION SHOWING OPEN EXHAUST STACKS ON WHIRLWIND J-5 ENGINE

leaving more space to the plane builder in which to conform the cowl ring into the lines of the fuselage or nacelle without abrupt changes of section. It is the author's opinion, however, exhaust cylinders where the exhaust manifold forms the leading edge of the cowl (Fig. 9). By the proper use of intercylinder baffles, it is possible to obtain excellent cylinder cooling either

with or without an inner cowl. Without baffles an inner cowl is desirable to decrease the drag by reducing the quantity of air passing through the outer ring and out of the rear slots. It is also possible to work out a satisfactory N.A.C.A. cowl with rear exhaust-port cylinders, but attention must be paid to exhaust manifold cooling. Townend rings, which give promise of distinct advantages over N.A.C.A. cowls in all respects except appearance, can be fitted over most standard cowls regardless of exhaust systems. Fig. 10 shows a ship equipped by the N.A.C.A. The cooling characteristic of this cowl is in most cases better than the conventional cowl with no ring.



FIG. 5 FRONT EXHAUST COLLECTOR FITTED TO A WHIRLWIND 225 ENGINE

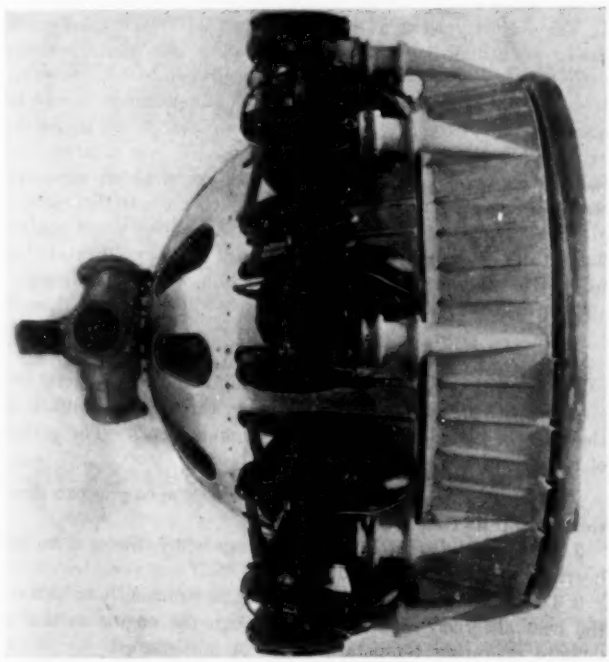


FIG. 6 WRIGHT CYCLONE ENGINE EQUIPPED WITH REAR EXHAUST COLLECTOR STREAMLINED INTO THE COWL

Recent fires from minor crashes have taken a toll of lives and done much to prevent public confidence in aviation. The cause of these fires is at present unknown, but it is suspected that uncooled exhaust manifolds play a large part. The British establish a maximum temperature for exhaust manifolds which is well below the flash point of gasoline. This reduction in temperature can be accomplished by exposing the exhaust system to the direct air blast and providing it with cooling fins. Cast-aluminum exhaust elbows provided with fins will also assist in reducing exhaust manifold temperatures. It is hoped that the N.A.C.A. will see fit to conduct experiments on crash fires to establish their exact cause. If this theory proves to be correct, a Department of Commerce regulation and an Army and Navy order on exhaust-manifold temperature would be in order.

Accessibility of engine accessories and oil connections is important for all types of exhaust systems and cowl. With conventional cowl developed around front exhaust rings or open stacks, doors are usually provided through which a mechanic can at least reach the part to be adjusted. Referring again to Fig. 6, in this installation the exhaust manifold and cowl are permanently secured together and both are supported by the

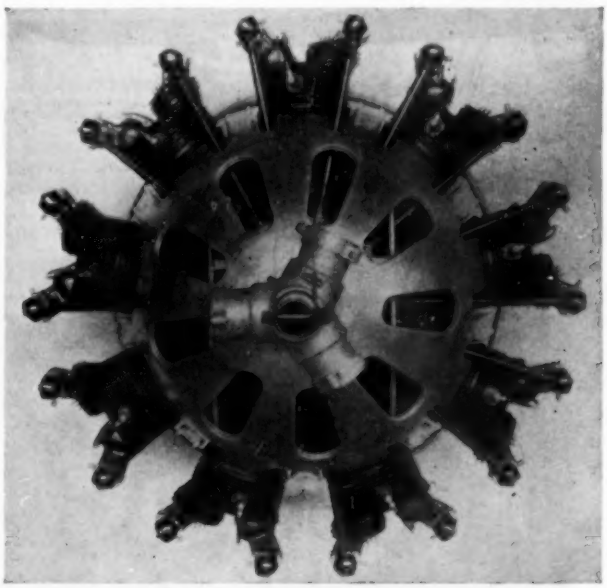


FIG. 7 WRIGHT CYCLONE REAR-EXHAUST MANIFOLD VIEWED FROM THE FRONT. NOTE THE CLEAN APPEARANCE OF THIS INSTALLATION

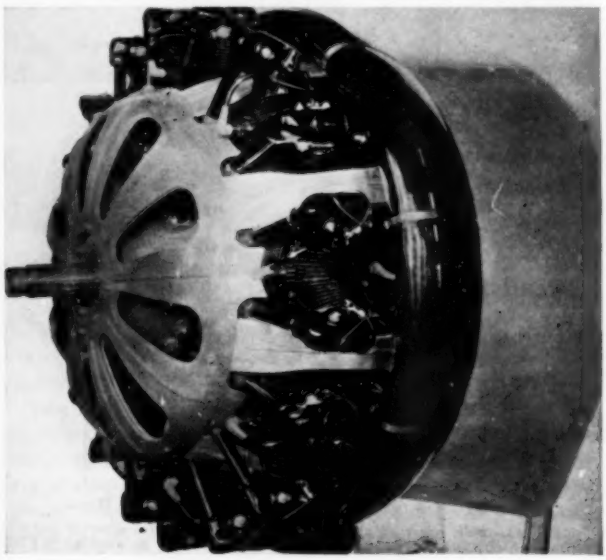


FIG. 8 PRATT AND WHITNEY WASP JUNIOR ENGINE WITH EXHAUST COLLECTOR AND NOSE COWL

exhaust flanges on the cylinders only. The exhaust ring and cowl are split on the centerline so that by removing the exhaust-flange screws the complete assembly can be removed in halves. The nose piece can be taken off over the crankshaft after removal of the propeller. Some such arrangement is advisable for all rear-exhaust systems.



FIG. 9 N.A.C.A. COWL SHOWING AN EXHAUST MANIFOLD WHICH FORMS THE LEADING EDGE OF THE COWL

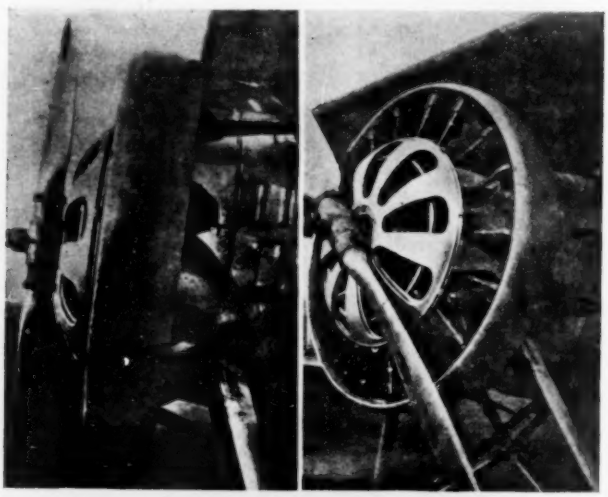


FIG. 10 ENGINE EQUIPPED WITH A TOWNEND RING
(Upper left, side view of cowling; upper right, three-quarter view of narrow-ring cowling; at bottom, front view of cowling as used on XF7C-1 airplane over service-type cowling.)

N.A.C.A. and Townend rings usually present a difficult accessibility problem. A segmental Townend ring attached to the cylinder heads under the rocker-box cover is suggested. In this case the rocker-box covers should be made angular to conform with the circumference on the ring. This permits the removal of rocker-box covers for servicing without disturbing the cowl. N.A.C.A. cowl should be made removable in halves secured by quick detachable fasteners.

CARBURETOR AIR HEATERS AND CLEANERS

Many types of carburetor air heaters have been fitted to radial engines, most of which increase the installation difficulties and spoil the appearance, often without accomplishing all of the functions for which they were designed. Air heaters and cleaners are installed for the following purposes:

1 To heat the fuel-air mixture, thus assisting the vaporization of the fuel and improving the distribution.

2 To prevent the formation of ice at the carburetor jets and throttle valve.

3 To prevent spray from entering the carburetor on seaplanes or in rain.

4 To prevent at all times foreign matter, especially sand and dirt, from entering the carburetor and causing damage or wear to moving parts.

5 To eliminate the possibility of fire caused by flame from an engine backfire.

6 To provide a control of hot and cold carburetor air for various atmospheric conditions.

In addition to the above, the unit must not interfere with installation, must not be of excessive weight or bulk, must not form an excessive restriction to volumetric efficiency in either the hot or cold position, and must not add to the fire hazard by requiring uncooled exhaust pipes under the cowl.

The unit which in the author's opinion most nearly meets all conditions is shown in Figs. 11 and 12. This heater obtains sufficient heat from under a front or rear exhaust manifold, or from open stacks, to prevent ice formation at the carburetor. It was formerly thought by some that a hot spot above the carburetor would be required to accomplish this, but experience with this and other heaters proves this belief to be erroneous. If more heat is desired for unusually cold conditions, it can be obtained on this banjo type by fitting a larger shield under the exhaust manifold, thus forcing the intake air over a larger surface of hot exhaust pipe. The heater contains an air-maze-screen cleaner impregnated with oil which collects dirt, and is also accepted by the fire underwriters as positive proof against flame from engine backfires. The use of hot or cold intake air to control oil temperature, in conjunction with the carburetor air heater, is an added advantage which will be discussed later.

LUBRICATION SYSTEM

The variation of the viscosity of oil with its temperature, coupled with its low heat conductivity, throw many pitfalls in the way of the designer of lubrication systems. The object of a lubrication system is briefly:

1 To store a quantity of oil from which the engine can draw to make up its consumption.

2 To supply oil to the pressure pump which forces it to the bearings at constant pressure.

3 To cool the oil coming from the engine sufficiently to balance the heat absorbed when passing through the engine so that a definite operating temperature can be maintained.

4 To prevent the entrance of dirt into the system which would cause wear of the working parts of the engine.

The suction line for normal operation must be of sufficient size to offer no great restriction to oil flow. It should be arranged so that the level of oil in the tank is slightly above the pressure pump to prime the latter under all conditions, since a gear pump which is universally used in oil systems, will not produce a suction unless primed. A reservoir should be left below the tank suction line to collect dirt and foreign matter which would otherwise be sucked into the engine. The line must be absolutely tight as air will be drawn through a smallest leak, and lower

the pressure delivered by the pressure pump. In cold weather this line must be protected from extreme cold as the increase in viscosity of cold oil will prevent circulation. This is usually accomplished by lagging the line with tape or asbestos cord and protecting it inside the cowl. The suction line is the main artery of the engine and should therefore receive scrupulous attention. It should be as short as possible and secured at frequent intervals to prevent vibration. Hose connections and fittings should be checked and rechecked for tightness. The service routine should always cover inspection of this pipe.

From the suction line the oil passes through a finger strainer to the pressure pump, which consists almost generally of a pair of spur gears running in a housing with appropriate inlet and outlet passages. From the gear pump the oil is usually forced through a fine-mesh screen, and from here it is delivered to the engine bearings. Oil leaking from the bearings in the main crankcase is thrown on the cylinder walls and pistons which are operating at high temperature. This oil runs down

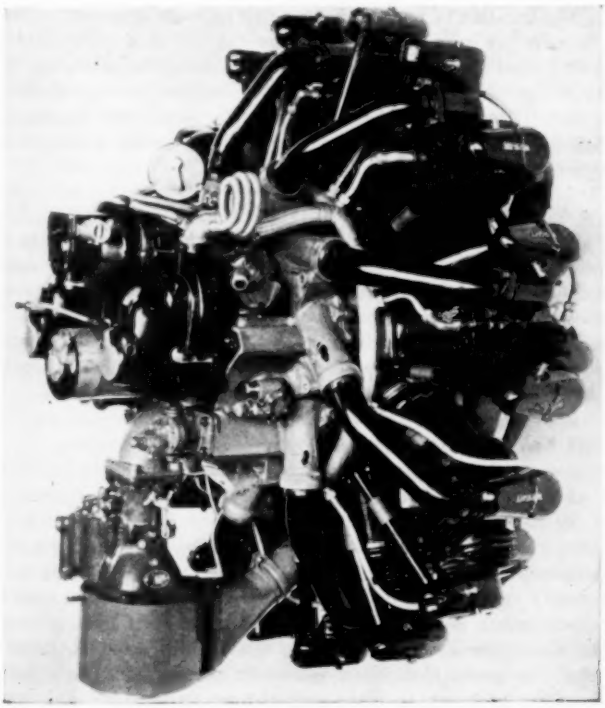


FIG. 11 REAR VIEW OF WHIRLWIND 300 ENGINE SHOWING BANJO-TYPE AIR HEATER AND CLEANER

the walls of the crankcase to the sump where it is drawn out by one or more scavenge pumps. The oil from the sump is frequently at 250 to 300 deg. fahr. during normal operation. This will astonish the pilot who has been brought up to look with horror on oil temperatures of 190 deg. fahr. The reason for this apparent discrepancy can be found in the mixture of the cold oil coming through the pressure relief valve, which mixes with the hot oil from the engine and indicates to the pilot what is considered a normal oil temperature of, say, 140 deg. fahr. Many pilots if suddenly confronted with the exact temperature of the oil in their engine crankcase would make an immediate landing and call loudly for service. The company with which the author is connected at one time placed an oil thermometer in the engine crankcase, but was immediately forced to move it to another position because no one would fly the engine.

From the oil sump, which may be little more than an embryo, the oil is sucked through one or more scavenge pumps, and returned by an outlet line to the oil tank. This line should be well braced to prevent failure, but is of nowhere near the importance of the inlet line because it is under pressure which will force a flow of oil under nearly all conditions. Should an oil leak develop, it means a dirty ship but not a forced landing unless the leak is so large that all the circulating oil is pumped out. The outlet line may pass through an oil cooler or oil strainer or both, on its way to the tank, which it should enter above the oil level. Frequently the effect of oil coolers and cooling means for oil tanks is very disappointing. This is due to the change in viscosity between hot and cold oil. The conventional oil cooler is constructed with a number of passages in parallel. The oil cannot be expected to flow equally through all passages.



FIG. 12 TYPICAL INSTALLATION OF A BANJO-TYPE AIR HEATER INSTALLED ON A WHIRLWIND 225 ENGINE

The passage having the greatest oil flow will increase in temperature and lower the viscosity. This action causes a more rapid flow in the working tube, and a complete or partial stoppage of the tubes which started with limited flow. In the oil tank the oil is cooled on the surface to which it adheres, forming a blanket to heat emission. The oil lines themselves will form a layer of slow-moving viscous oil on the outside with a fast-flowing core of hot oil in the center. The best method of efficiently cooling hot oil in a plane installation is to pass it through small flat passages in series. The pressure will then force all the oil through all the tubes which should be narrow enough to prevent the formation of an insulating layer of cold oil.

Oil coolers have long been the bugbear of plane manufacturers because of their unstable characteristics and the difficulty of installation. To require the plane manufacturer to hang an oil cooler in the slipstream of an airplane, which he has spent much thought and hard work to clean up, seems like adding insult to injury. Many small engines can obtain sufficient cool-

ing from the oil tank and lines, but most large engines and nearly all geared engines need oil coolers.

A new method of installing coolers has recently been developed which transfers the burden from the tired shoulders of the plane manufacturer to the engine manufacturer who originated this system. He has heated the oil, let him cool it. This new installation puts a series radiator in the air heater below the carburetor (Fig. 13). All the air entering the engine passes over the oil cooler and furnishes adequate circulation for the most extreme conditions. More than this, by this method the oil can be quickly warmed at starting when the cold-air doors on the heater



FIG. 13 INSTALLATION OF AN OIL COOLER IN A CARBURETOR AIR-HEATER ASSEMBLY FOR WRIGHT WHIRLWIND ENGINES

are closed and warm air from the manifold passes over the cooler. As the oil temperature rises, the pilot cracks the cold-air doors to obtain any degree of control of his oil temperature. This same control is satisfactory to furnish hot and cold air to the engine carburetor, as when the oil is cold the engine carburetor requires hot air, and when the oil is hot the engine carburetor can be operated on colder air. Heat from the oil also furnishes some of the heat for carburetion.

OIL FOAMING

How many times have pilots been forced down by low oil pressure! When on the ground they remove the relief valve, find nothing, "cuss" the engine manufacturers for building a poor engine, replace the valve, take off with full oil pressure, and continue until the same thing happens again. If the pilot had left the engine alone and smoked a cigarette instead, the trouble would have been repaired just as quickly. His trouble was not the relief valve but foam in the oil tank.

The scavenge pump or pumps of an engine must necessarily have a greater capacity than the pressure pump to prevent an accumulation of oil in the sump. Since all the oil supplied to the scavenge system comes through the pressure pump, the difference in capacity must be made up by air passing through the scavenge system back into the tank. This air becomes intimately mixed with the oil in the form of minute bubbles. An additional quantity of bubbles may be induced into the system by the return oil splashing into the volume of oil in the tank. This action can be observed when running a swift stream of water into the bath tub. Under favorable conditions, a large proportion of the water in the tub will contain small bubbles. The effect of these small bubbles in the oil is to reduce the oil pressure. Oil foaming will occur more readily under the following conditions: (1) hot oil, (2) small quantity in tank, (3) short oil return line, (4) engine having numerous scavenge pumps and a small sump, and (5) high narrow oil tank.

The question of how to prevent this pressure drop next arises. An effective method is to discharge the oil on a plate sloping into the oil volume, and allow it to run down the sides into the oil in the tank. Fig. 14 shows a suitable oil baffle for foam elimination. It is hoped that plane manufacturers and mechanics will recognize this phenomenon and stop blaming an innocent relief valve for their troubles.

FUEL SYSTEMS

There are at least as many fuel systems as airplanes, most of which resemble the engine control of a battleship in complexity, especially in multi-engine planes. A few general considerations may be in order.

1 Gasoline cannot be lifted with a suction head without causing vapor lock under certain conditions which completely breaks down the system.

2 Vertical bends with high points in a gravity-feed system will become vapor bound and stop the flow of gasoline.

3 Gear fuel pumps must be primed before they will function.

4 Fuel pumps will not operate dry for extended periods.

From these considerations it is obvious that the fuel tank must be slightly above the engine-driven pump, unless the engine pump is used to transfer gasoline from a low tank to a gravity tank from which the carburetor is fed. If the engine-driven pump is on the engine, this presents a serious limitation to the plane designer. Present conventions seem to dictate that the pump shall be mounted on the engine. It is quite possible, however, to place a diaphragm in the fuel tank with suitable valves which is actuated by oil pressure from a unit on the engine. This gives excellent satisfaction in supplying steady gasoline feed to the carburetor from a low tank without danger of vapor lock or losing prime. Such a system has been operated experimentally but never put on the market. Possibly some plane manufacturers would be interested in further developments along these lines.

Engine fuel pumps are often equipped with relief valves to maintain a constant pressure. If a hand pump is used, the carburetor floats will be subjected to high pressure from this pump unless the relief-valve line discharges behind the hand pump. Fig. 15 illustrates this point diagrammatically. When a number of tanks are employed, each working through one fuel pump, the system becomes complicated as the relief line must return to the tank from which the fuel is drawn to prevent overflow of an unused tank. In this case a short bypass to the engine fuel pump can be used, carrying the fuel discharged by the relief valve to the intake side of the pump. In this case an additional relief valve should be inserted to bypass the hand pump

to prevent damage to the carburetor float from the excess pressure of the hand pump.

As fuel-line failures not only cause forced landings but are a serious fire hazard, it is impossible to be too careful with their installation. As an illustration of carelessness in this respect, one transatlantic plane made the ocean trip with a fuel line on which two hose clamps were applied to one pipe, and the other end of the joint was stuck together with nothing. Fortunately for them no one intimately connected with this flight ever learned of this condition. Fuel lines should, of course, be braced at frequent intervals to prevent vibration and failure. Each installation is a special study, requiring experienced engineering, and is not a job for the mechanic alone. Too often the mechanic is given the entire responsibility for this most vital part of an expensive ship. His judgment and methods may be good or bad, but he will probably take the path of least resistance and neglect important details. Landing gears are designed before they are built; why should fuel systems receive less attention?

Pusher Installations

Nearly all radial air-cooled engines are designed as tractors. When they are used as pushers, a number of alterations should be made which are ordinarily overlooked. The carburetor is designed so that when the ship is accelerated, causing the gasoline to surge to the rear of the float chamber, the fuel discharge opening will not be uncovered. Reversing this unit will maintain this design characteristic, which if omitted, will result in cutting out the engine during acceleration. Modification of the air heater may also be required to accomplish this.

The intake pipes of a pusher are in front of the cylinder, and usually exposed directly to the slipstream which may cool the pipes to a point where the mixture condenses on the walls and causes rough running. Lagging of intake pipes is the obvious remedy for this condition.

Air-cooled cylinders are designed to cool the hot points of the cylinder best with the air blast of a tractor. The cooling as a pusher will therefore be less efficient resulting in poorer cylinder performance. They may require a better grade of fuel, richer mixture, or a reduction of compression ratio to insure the same degree of durability. Special cylinders would eliminate this limitation, but to date have been considered unnecessary.

The pusher installation also suffers because of the reduced slipstream velocity on the ground and in takeoff and climb. Since a propeller takes air from all directions and delivers it to the rear, the air velocity over the cylinder of a pusher is very low until flying speed is obtained. This also makes cylinder cooling more difficult. By the use of a Townend ring and special baffles the cooling of a pusher installation can be greatly improved. It will always be somewhat less effective than the equivalent tractor installation, however.

The conditions of a pusher engine apply to the rear engine of a tandem installation except for two aerodynamic effects. One is the front propeller slipstream which helps to cool the rear cylinders, and the other is the turbulence created by the forward engine which usually hinders cooling on the rear engine. Here again, the use of ring cowl and special baffles properly designed will greatly increase the cooling of both the front and rear engine. This study has not yet been completed but will undoubtedly improve tandem engine performance when the results are available.

Geared Engines

The problem of cooling the cylinders of a geared engine is necessarily more difficult than a direct-drive engine. The large-diameter slow-speed propeller delivers very little air at the cylinder barrels when running on the ground. The condition is greatly improved in the air, but is still far less satisfactory than a straight drive engine. Since one of the principal advantages of gearing is to permit operation at higher engine speed and thus higher power, the cooling requirements are even more drastic than the direct-drive engine. All the statements concerning cowl and spinners for maximum cooling apply to geared engines, and should be even more carefully applied because of the more severe requirements.

Inverted Flight

High-performance planes are invariably called upon to fly for short periods in the inverted position. This requires special

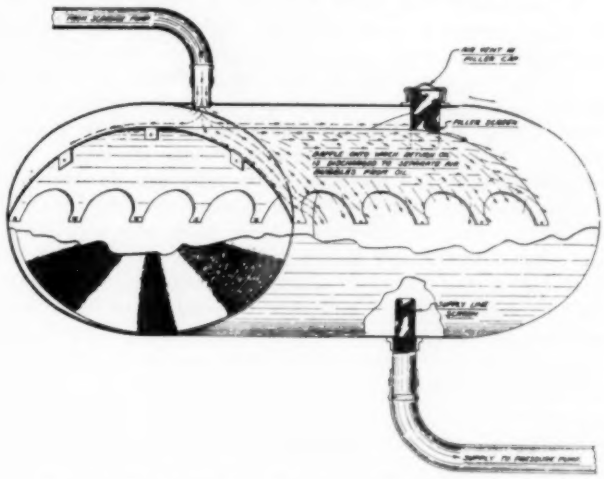


Fig. 14 Oil Tank With Baffle for Foam Elimination

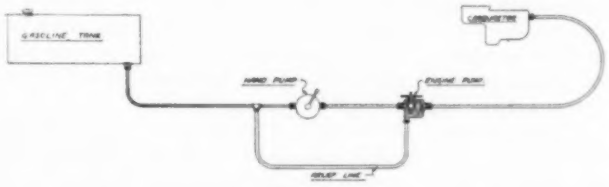


Fig. 15 Fuel System Relief Valve Line Bypassing Hand Pump

installation considerations. For indefinite inverted flight a complete carburetion and oiling system is required for the inverted position. This is very nearly a duplication of the system used in the upright position. For short periods it is possible to make small modifications which will give satisfactory results.

The fuel system must be arranged with a fuel pump having a suction line at all times below the gas level. A line to the center of a tank more than half full of gas, or a flexible line inside the tank weighted to submerge it below the surface of the fuel, will accomplish this. The carburetor must be fitted with a restriction in the gas line which will maintain an appropriate fuel-air ratio when the fuel pump is discharging through it at a predetermined pressure with the carburetor needle valve opened by the float in the inverted position. The vent in the gas tank must be brought by a pipe below the level of the bottom of the tank so that no gas will leak out when inverted. These modifications enable the fuel to get to the intake passage in combustible proportions when running at a narrow range of engine speeds. This will enable the engine fuel system to operate indefinitely.

The suction side of the oil-pressure pump should be arranged

in a similar manner to the fuel line with a weighted flexible pipe in the tank or an outlet in the middle of a tank more than half full of oil. Either system will assure engine oil pressure. No provision is made in the usual installation for scavenging the engine crankcase when inverted. The oil, therefore, accumulates in the case and is returned to the tank only when the plane is again in the normal position. The oil tank and engine breathers must be arranged to prevent spilling. This can be done either by check valves which close in the inverted position, or a pipe from the top opening of the tank leading below the lowest point of the oil system. The engine and oil tank can be connected through the breather and supplied with one vent. A check valve may be necessary to prevent filling the engine crankcase from the oil tank when inverted.

In concluding this paper the author wishes to bring out the shortcomings of both plane and engine builders. Neither have made a real study of engine installations with the idea of obtaining both aerodynamic and engine efficiency. It is the duty of each engine manufacturer to study and determine installation details, giving the best results for his particular engines, together with the best possible aerodynamic efficiency. This information should be passed on to the plane manufacturer, who should incorporate the installation details, furnished him, with the least possible modification. Many parts now supplied by the plane manufacturer should properly be supplied with the engine since they materially affect its performance. It is hoped that intelligent research and close cooperation between plane and engine manufacturers will eliminate present installation faults, and pave the way to further progress in the safety and reliability of aircraft.

Discussion

CAPTAIN T. E. TILLINGHAST.[2] This interesting and comprehensive paper shows that at least one engine manufacturer has been giving considerable thought and study to the problem of installation of radial engines. It is hoped that this paper will be made available to all airplane manufacturers and that they will read it carefully and profit by the sound advice and recommendations contained therein.

It is only natural, at this stage of the art of aviation, that the experience of two airplane manufacturers should not agree 100 per cent. It is quite remarkable that the paper should coincide so perfectly with the ideas of a competing company.

The author's remarks regarding the number of mounting points furnished on the average air-cooled radial engine are well taken. The typical engine-mount structure described and illustrated is practically in general use, and it is hardly conceivable that those points in between the four apexes of the tubular members can be doing very much work. Before one abandons this so-called mounting circle, however, he should consult those airplane manufacturers who are using a monococque type of construction, to see whether or not they desire to utilize the circle mounting. It is quite probable, of course, that the monocoque construction might end well after the engine compartment, and that the tubular structure be built in for the forward part of the fuselage to obtain complete accessibility to the engine from the outside.

As pointed out by the author, from the point of view of the airplane manufacturer, the front type of exhaust collector has a much more general application than any other type. It undoubtedly will be lighter than the rear manifold and should be at least as well cooled. There would seem to be, however, still some question as to whether this front manifold might necessitate some sacrifice in engine cooling. The writer has had no experience in this regard and would appreciate the author's opinion on this point. Another advantage of the front type manifold is visibility, where the engine is so installed in the plane that the pilot must of necessity depend upon peering between the cylinders for forward vision.

The suggestion of the establishment of a ring in back of the engine, forward of which the engine manufacturer would furnish all cowling, and aft of which the responsibility would be that of the airplane manufacturer, is a very good one. The Pratt & Whitney Aircraft Company has for some time been attempting to establish such a ring. It has met with considerable resistance, however, due to the many different shapes of fuselage already in use. It would seem as though some such system will of necessity have to be adopted, and the newer planes being designed and built can readily conform to it.

The exact location of the exhaust collector in the rear of the engine is a compromise between visibility, weight, and cooling of the manifold. Naturally, the smaller the diameter of the manifold the lighter it is going to be. It cannot be sunk into the cowling to any extent and still be cooled. If it is too great in diameter, it may interfere some with the forward vision. When radial engines first came into use, they were installed in fuselages of approximately the same shape and size as were used with the V-type engines. As a result, the visibility forward was in most cases very poor, and it was necessary for the pilot to look between the cylinders for forward vision. There are still airplanes in use with this unfortunate condition. Those manufacturers supplying airplanes equipped with radial-type engines for military purposes have now learned how to place the pilot so that he has good vision forward, which is not interfered with by the engine cylinders and is still good when the ring cowl or the completely cowled engine is used.

Tests which this company has made to date indicate that, as the author has pointed out, a manifold in the rear of the engine can be made as large in diameter as the engine itself with no increase in resistance. It would appear, then, that so far as performance of the airplane is concerned, the diameter of the collector matters but little. It is probable that, for installations with the ring cowl, the collector will be approximately the diameter of the engine and a part of the cowl, and that for installations without the ring the diameter will be about that of the cylinder head, allowing the valve ports and rocker housings to project above the collector ring. This latter arrangement permits a fair amount of vision forward between the cylinder tops.

The writer's experience in the matter of nose shutters is somewhat different from that of the author. While it is true that the temperature of the cylinder flanges varies from 100 to 300 deg. fahr., during which range the engine operates satisfactorily, the writer believes that it is an advantage to have the engine warmed up with the shutter closed. Uneven temperatures fore and aft of the cylinder barrel causes some distortion. Tests also indicate that when the throttle is first opened the piston gains in temperature rapidly and expands. The inner surface of the cylinder barrel may also gain in temperature rapidly, but the tips of the cooling fins remain comparatively cool and will not allow the cylinder to expand very much. This condition is apt to cause an unduly severe piston rubbing condition during the warming-up periods in cold weather and to necessitate the operation of the piston with excessive clearance to provide for this condition. For this reason it is recommended that the nose shutter be controlled from the cockpit and not be used with a seasonal adjustment.

The writer concurs heartily in the author's statement regarding air-exit louvers to the rear of the engine and oil tank. This company appears to have had identical experience with that of

[2] Pratt & Whitney Aircraft Corporation, Hartford, Conn.

the Wright Aeronautical Corporation. It has carefully provided cowling in front of the engine to permit adequate cooling, and the airplane manufacturer has blissfully made an airtight cowling, or practically so, to the rear of the engine.

While spinners may materially improve the air flow around cylinders when the airplane is on the ground, it is doubted if they add much to the cooling in the air where air is approaching the center of the engine at the speed of the airplane; also, as was pointed out in the paper, spinners are very difficult to manufacture so that they will be durable, and they do add nothing appreciably to performance. They have the added disadvantage of sacrificing the ability to lead cooling air to the crankcase nose, which it is believed contributes some to cooling of the oil.

One of the most important developments in connection with the cowling of a radial engine is the ring type of cowling. This detracts but little, if any, from the visibility forward, and in addition to the obtained gains in performance approximately equal to that of the completely cowled radial, they appear to add some to the cooling of the engine. They can be made in segments which are easily detachable, to allow accessibility to the rocker boxes. As a matter of fact, with a well-designed ring cowling, the accessibility to a radial engine is infinitely better than the accessibility to a V-type engine with its usual cowling. With all planes with radial engines designed from now on there appears to be no reason why they all should not provide for the ring type of cowling. It is one of the few examples of getting something for nothing. If there is any provision which the engine manufacturer can make on his engine for the installation of this cowl, he should certainly make every effort to provide it.

It is also possible to provide combination ring cowling with rear exhaust manifold, as has been so well demonstrated by the National Advisory Committee for Aeronautics. This combination does limit the visibility between cylinders, but, as already pointed out, airplane designers are now placing the pilot in a position where this type of visibility is unnecessary. This combination in no way interferes with the cooling of the engine and probably improves the cooling conditions as compared to an uncowled engine.

All engine manufacturers have had the disastrous experience of having their engines flown in the dusty fields of California and Texas. There appears to be two ways of largely eliminating the undue wear caused by the dust in the engine; namely, top air intakes, or at least intakes above the horizontal center line of the fuselage, and the use of air cleaners. The Army Air Corps has had considerable experience in these two methods of providing freedom from dirt. It is believed that both are equally efficacious. The cleaner, however, particularly the screen type, must frequently be cleaned and dipped in fresh oil. The top air intake is practically of no inconvenience. As soon as a downdraft carburetor becomes available for aircraft engines, it should be a simple matter to reverse the accessory section of an engine 180 deg. and easily provide for a top air intake.

It is unfortunately necessary to apply heat to the induction system of a carbureted engine to obtain good distribution. There are several ways in which this heat may be applied: the suction-type supercharger, the so-called hot spot, and a preheater for the air. This last method heats the air before it has gone through the carburetor and is used extensively by the Boeing Air Mail and Pacific Air Transport Line with very satisfactory results. With the hot spot and the supercharger some preheating of the air is absolutely necessary to insure against ice forming in the carburetor barrel, and if all of the heat can be applied efficiently in the preheater type without resorting to a hot spot or the supercharger it would naturally be more convenient. It has been the experience of the Pratt & Whitney engineering department, however, that the hot-spot type of heater, while not so simple in installation as the preheater type, does nevertheless give more satisfactory results, particularly in very cold weather and at altitudes where low degrees of supercharging, such as ordinary rotary induction, are employed.

In the first place, the hot spot accomplishes good distribution at a lower mixture temperature than that obtained with the use of preheated air. As a result, less horsepower is lost due to lowering of the volumetric efficiency.

In the second place, in cold weather the carburetor itself seems to be a very excellent radiator, and a great deal of heat from the preheated air is lost in passing through the carburetor before it ever reaches the fuel. To be able to operate satisfactorily in all temperatures, it has been the writer's experience that the heating arrangement must have ability to increase the temperature of the air intake by at least 200 deg. fahr. Such an increase in temperature very naturally requires a comparatively great amount of hot surface over which the intake air must pass. For these reasons the Pratt & Whitney Aircraft Company has resorted to the use of the hot spot to obtain distribution with a comparatively small amount of preheating to prevent ice formation.

The trend toward larger horsepowers for several years past has brought one face to face with the problem of some means of cooling the oil. It is probably safe to say that engines of greater than 400 hp. require some external means of oil cooling. The Army Air Corps has been able to control oil temperatures very successfully by subjecting the tanks to a flow of warm air either from the water radiator or from the cylinders in an air-cooled engine. With the adoption of larger horsepower, however, it is believed that some type of oil radiator will be necessary. Just what form and type of radiator this will be and where it will be placed, the writer is not prepared to say. Certainly the type of radiator, together with its position, as suggested by the author, is very ingenious and has much to recommend it. There seems to be little question also that the passages must be in series in so far as possible. The use of an oil radiator with but little radiation from the oil tank has a great deal to recommend it. With a bypass valve for cold weather, the oil in the tank will warm up far sooner than in a system depending upon heat radiation from the tank. When the oil gets to the temperature where it will require cooling, the bypass valve will close and the oil will be sent through the radiator. This system probably will necessitate some shutter arrangement in order that the radiator may be warmed up and filled with hot oil prior to exposure to cold air. The type and the position of the radiator described by the author seem to combine all the desired characteristics, and furthermore the radiator is not placed in the slipstream where it will cause drag.

Unquestionably commercial aviation has experienced quite some difficulty with its fuel systems. The Army Air Corps during the past ten years has evolved standard fuel systems to fit practically every need. It is believed that these fuel systems are considerably in advance of the majority of those used in commercial aviation, and it is strongly recommended that these Air Corps systems be adopted as commercial standards. They would eliminate many of the troubles now encountered commercially.

Author's Closure

Mr. Tillinghast's suggestion that the plane manufacturers be consulted before any radical change in engine-mounting structure be made is accepted. However, if this question is left entirely to the decision of the plane manufacturers, little progress will be

made, as they have little desire to change their equipment for the ring type of engine mount. The advantage of rubber installation and the four-point method of support is mainly to the engine manufacturer in protecting his crankcase and insuring against engine vibration. The manufacturer must therefore push this change down the throat of the plane manufacturer unless the operation should prove too drastic for other reasons.

The Wright Company has had a tremendous amount of experience with front-exhaust collector rings, and the cooling of Wright engines is quite satisfactory with this equipment. If the engine is to be supercharged to the point where cylinder cooling is the deciding factor in its operation, the removal of the exhaust from the front of the engine will undoubtedly reduce cylinder temperatures to some extent. It is felt that the cylinder cooling is sufficiently effective to permit the slight increase in air temperature caused by the front exhaust manifold on present engines. The installation advantages are undoubtedly sufficient to warrant this arrangement.

The author cannot subscribe to the theory that the tips of the cylinder-barrel fins will restrain the expansion of the cylinder barrel during the warming-up period to any great extent under any ordinary conditions, as he believes the heat flow in the steel cylinder barrel is sufficiently rapid to conduct the heat to the tips of the barrel fins almost as soon as it is generated. The author would like to raise the question of how many engines are warmed up with the nose-cowl shutters closed. He believes this is one control given to the pilot which he seldom takes the trouble to operate unless his oil temperature is definitely too hot or too cold.

The effectiveness of the top air intake in preventing dirt from reaching the engine was tested by the Army Air Corps and compared with an air-maze cleaner. The result, as Mr. Tillinghast states, shows little advantage for either type. The air-maze installation used for the comparison was definitely unsatisfactory to this unit, as the air maze was exposed to the direct propeller blast in the most unfavorable position from the standpoint of dirt. The new type of heater and cleaner assembly suggested in the paper completely protects the air maze from propeller blast and shields it from dirt to a large extent. In other words, some of the advantages of the top air intake are incorporated in the heater unit, to which an air-maze cleaner is also added. It is believed that it would be well worth while to repeat the comparative tests with an air-maze cleaner in a modern installation. The disadvantage of the top air intake is principally in the difficulty of installation, as wherever the intake passage is placed it will interfere with the structure or fire wall in some installation. The author recently has had occasion to examine several top air-intake installations and has found a considerable deposit of sand and dirt inside the pipe. He therefore believes that the air-maze unit is probably the more effective of the two. The use of downdraft carburetors would make possible the combination of the top air intake and air-maze cleaner, which could take the form of an inverted banjo heater with the air maze incorporated. This is an ideal installation from the point of view of elimination of dirt in the engine.

Mr. Tillinghast states that a 200-deg. temperature rise in the mixture from the air heater is desirable for carburetion. This figure, in the author's experience, is altogether too high and must necessarily greatly reduce the power of the engine. Computing the power of an engine with and without an air heater of this capacity by the means employed by the Army or Navy would entail a loss of 15 per cent in power due to the reduction of volumetric efficiency. The Wright Company has been able to produce very satisfactory air heaters in which the total temperature rise is 80 to 90 deg. fahr. Results of tests made by disinterested parties have indicated that the performance of an engine is identical whether the air is heated before or after the fuel is introduced. The elimination of exhaust piping under the cowl and a hot pipe in the mixture itself is thought to be highly desirable from the point of view of possible fire hazard. The abandoning of the hot-spot development by the Wright Company was based upon this consideration.

AER-52-13

All-Metal Airplane Construction

By WELLWOOD E. BEALL,[1] PASADENA, CAL.

The loads to which airplanes are subjected are pretty well known so the stresses in various members can be calculated. A light structure is necessary, but strength and durability must be increased due to the demand for greater speed and more hours in the air. Metal has thus been called into construction to help the designer.

Some of the advantages from use of metal in airplane construction include ease of determining the strength of members, decreased fire hazard, less danger to passengers in case of a crash, and decreases in cost of production, labor, handling, bulk, and waste.

Use of aluminum alloys is increasing, and monocoque construction is becoming more popular. Methods of analyzing the stresses in various members are given in the paper.

STUDY of various papers and articles written by those closely connected with the airplane industry indicates that the term "all-metal construction" is sometimes applied to an airplane whose covering is fabric but whose structure is metal. An all-metal airplane should mean one whose strength members are not only of metal but whose covering is also of metal, and this type will be discussed in this paper. Most metal covering is not supported by the conventional ribs such as fabric requires, hence, an "all-metal" airplane is in a class by itself.

The substitution of metal for wood has followed the advancement of aircraft design. With the increase of knowledge of the aerodynamic qualities of the airplane, more knowledge of the application of the loads on the structures has been gained. The first airplanes consisted of structures figured by the rule of thumb. A member of any size was placed at any point the pilot-builder might deem wise, but later, as certain loads were recognized, the structure was designed to fit that load. The stick and wire structure then sufficed. As time went on more exacting design conditions were set down and structures had to become more accurate.

At the present time considerable is known of the loads to which present-day airplanes are subjected. Not only that, but due to the efforts of the government, exacting conditions of strength must be met. Hence, to design to such exacting loads material must be used that is absolutely safe and whose exact mechanical properties are known. The pay load of an airplane is an all-important item. As the importance of high pay loads is realized, a parallel thought comes to the engineers' and producers' minds. The lighter the structure the more the pay load. To lighten the structure, a greater strength-weight ratio must be had. The structural weight must be reduced, but strength and durability must become greater because of the increasing demand for speed and more hours in the air. Hence, metal has to play its part in helping the designer.

Some new construction is being carried on with metal ships covered with fabric. Metal construction has many advantages and also a few disadvantages. Every one agrees that as much metal as is economically possible should be used, but for small job-lot production schedules, small wooden parts are cheaper than metal. Where real production is paramount, metal is by far the cheapest, especially in larger and in highly stressed parts.

Use of metal permits a much closer degree of accuracy and saves weight. Metal has a moral advantage over wood in that the public feels much safer on a steel bridge than on a wooden one, and justly so. Since wood can have such varying characteristics, it is necessary to make many tests and inspections of the material that goes into wooden parts. If the properties of the wood in any part are to be known, it becomes necessary to test a piece of material cut from the same piece of wood from which the part is made. It is oftimes rather difficult to obtain a specimen for test, and in many cases next to impossible, because of the cost of the numerous tests, the wasted material, the time lost, and the inconvenience. In metal, all these tests are not necessary as the physical properties are controlled throughout its manufacture and treatment. If treated identically the properties of two runs of metal will be the same, and all parts made from standardized metals and treatments can be counted on to carry the same load and individual tests are not required.

The fire risk is another all-important factor in aerial transportation. Metal has the advantage over wood in this respect, and if the ship is all metal the only fire risk is due to the fuel and inflamable upholstery.

Another advantage of metal is in the event of a crash the passengers and crew are not subjected to the effects of broken and splintered wooden members, whereas metal members will usually bend and crumple and not break and splinter. This has been effectively proved in the case of wooden and steel railway coaches. Metal has many other advantages, such as the effect of weather on the structure, cost of production, labor, handling, production practice, bulk, waste, and of course, a structure with a much improved strength-weight ratio.

Of course the problem of corrosion of metals has to be met, and it is believed that if the proper preventatives are made very little trouble will be experienced from corrosion. Two vital problems in all-metal construction are the choice of metals and the type of structure. The metal to use in any part is a function of the type of part, the service for which the part is intended, the strength required, standardization, the weight allowable, and the cost allowable. However, even after consideration of the above functions, it is sometimes a rather difficult question. The two metals that seem to have been adopted by the designer, and which have been almost universally used are alloyed steel and aluminum alloys.

In both the United States and England steel has been used to a great extent, however, in different forms. The English are very favorable to structural members built up of highly alloyed high tensile-strength steel. These members are usually made up of very thin steel strips fabricated into various sections by rivets. Thus, material that has been thoroughly heat treated may be used, and in almost all cases the heat treating is done before fabrication. Stainless steel is also very popular. In the United States the use of steel has been in the medium steels in the form of tubes and in built-up fittings, nearly all steel parts being fabricated by welding.

The use of aluminum alloys is becoming more and more prevalent every day. Several manufacturers both here and abroad are building ships constructed entirely of duralumin with sections made from bent sheet metal fabricated with rivets. All the

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

problems connected with duralumin have not been solved as yet, but this metal has wonderful possibilities.

The form of the structure is of utmost importance. The problem seems to be whether the structure should consist of a skeleton framework which carries the loads, and is covered with a skin that merely makes the form of the ship, or whether this skin should be made to do double duty and also carry the loads, eliminating the skeleton framework. There is an increasing demand in modern airplanes for cleaner and better interiors with more unobstructed space. The answer to this is to put the skin to work and use monocoque construction, which eliminates all these interior members. More total and more unobstructed room is possible with the same outside dimensions, expensive flairing is eliminated, and better lines and contours are possible. Since a skeleton structure is a series of stressed members joined with a series of bulky joints and connections, it will run into more weight than will a single unit structure where there is no multiplicity of joints, and where the stressed material is farther away from the neutral axis and consequently does more work. In other words, a monocoque structure will show up more favorably in a strength-weight comparison. The classic example of this type of structure is the egg, whose shell has phenomenal strength yet is very thin and light in weight.

To date there has been little real work on all-metal monocoque structures. Some companies have done some research work on the problem, and one or two have built some experimental ships and a very few are now under construction. Quite a number of ships with wooden monocoque fuselages have been manufactured. Most of the work that has been done has been on the fuselages, but the wing as well might be of the monocoque type.

There are two kinds of shell-type construction, the full monocoque and the semi-monocoque. The full monocoque consists of a shell in which transverse diaphragms or bulkheads are placed. These are placed along the longitudinal axis of the shell at certain intervals determined by design characteristics and loads. The direct longitudinal, torsional, and resulting diagonal stresses are carried by the shell alone while the bulkheads are subjected to only the shear load. These buklheads also hold the shell in shape and act as formers. The semi-monocoque structure is the same as that outlined except with the addition of longitudinal stiffeners that carry some of the direct stresses, leaving the shell or skin to take care of the diagonal and torsional loads. The diagonal loads in each case are a result of the shear load.

This type of construction may be used for wings as well as for fuselages. Wings of monocoque construction will run much lighter per square foot of surface than will an all-metal wing with skeleton structure. While an all-metal full-cantilever wing will weigh in the neighborhood of 2 lb. per sq. ft., a monocoque wing will weigh only about $1^1/_2$ lb. per sq. ft. However, even this figure may be reduced if the wing is continuous from tip to tip and there is no break in the monocoque structure. It has been the author's experience with monocoque wings that they will weigh much less and simplicity will be gained if the wing is continuous.

One of the greatest problems encountered in a wing built in sections is the picking up of the load in the stressed covering and concentrating it on fittings. By a wing built in sections is meant a wing whose outer portions are fastened to a center section. Not only is the analysis very difficult, but the connection itself becomes heavy. Nearly 20 per cent of the total wing weight can be charged to the load-pick-up members and the fittings. If the wings are continuous, they can be fastened to the fuselage or cabane by means of cradles and slings. For instance, with a high-wing monoplane, a cut out on top of the fuselage would form a cradle on which the wing could be placed. Then metal bands or straps could be placed over the top to hold the wing on. Small lugs could be fabricated on the covering and then could be attached to the fuselage and to the hold-down straps to keep the wing in exact position. All the direct stresses due to lift, etc. would be taken either directly on the fuselage or by the straps. On the other hand, if the entire wing is to have only its outer portion of the monocoque type and the center section is to consist of spars in order to hold engine nacelles or landing gear, some means of attaching this shell and outer portion to the skeleton structure must be devised. The conventional wing consists of two spars and usually, if the wing is cantilever, there will then be four wing fittings on which to attach one-half of the whole wing on one semi-span section. Hence, the load must be picked up from the stressed covering and concentrated at four points. Then the load must be divided between these four points. Not only is the problem of picking up the stresses difficult, but proportioning the load in the correct manner among the fittings is difficult.

The manner of analysis of any monocoque structure is always open to question. That of semi-monocoque construction is not especially difficult if fundamental mechanics are applied to longitudinal stringers and transversal bulkheads, but with full monocoque construction the fundamental relations do not always hold. The exact path of the forces and stresses is not always known, and due to the many interruptions in the shell due to holes for various purposes these stresses cannot be traced. Several universities are equipped with light projectors with tourmalene lens which project polarized light through models of these structures made of pyrolin. Small loads are then applied to the model. The image thrown upon the screen will then show dark "tracers" indicating the flow of stresses and the highly stressed portions of the shell due to the loads applied. A great deal more work could be done in this direction and perhaps a new set of fundamental relations could be set down for use in analyzing and designing stressed shell structures. The kind of skin used in this type of structure is open to debate. Several engineers stick steadfastly to corrugated covering while others are exponents of flat skin stiffened by stringers or thin drawn sections fabricated to the skin or individual channels or corrugations riveted together. Sometimes, the question of cost, production, utility, availability, and various other considerations are the determining factors. The logical solution seems to be in the corrugated covering with the corrugations running parallel to the longitudinal axis of the structure, and their being covered over by very thin flat sheet metal. This flat sheet on the outside not only produces a very pleasing appearance, but also holds the corrugated covering into form, and makes a very rigid structure indeed.

Perhaps it may be interesting to outline briefly the procedure followed in making an analysis of a pure monocoque wing. This is a full cantilever wing fastened to the center section by four pin joints. First the type of covering must be chosen. As mentioned before this covering may consist of corrugated sheet metal with whole sheets corrugated, of channels, or of individual corrugations riveted together, or of flat sheets which have been stiffened by the addition of various forms of small bent-up sheet-metal stringers. At any rate, several tests should be made to determine the mechanical properties of a standard section. For instance, the covering might consist of a number of bent-up channels riveted together by their flanges, these channels running along the span and inside of the wing leaving the external surface smooth.

In the analysis the load on the wing is not generally taken in two planes as in the conventional wing. The lift and drag loads for all conditions are combined into a resultant load. This resultant load in its direction is the load used for analysis. The resultant component of the dead weight of the wing and the net air load per pound per inch run is plotted against the span in

inches. This curve is integrated once to find the shear at any point along the span, again to find the bending moment, and twice more to find the deflection of the wing at any point along the span.

The wing, in most cases of cantilever construction, is tapered. Sometimes only in either plan or thickness, but often in both. The moment of inertia of the stressed skin around the wing section must be calculated. This is really not very difficult. First, the centroid and neutral axes of the wing section must be located. The skin is then divided up into small individual sections, such as one channel, in the case of a wing built up of channels. The characteristics of each individual section is usually known. The axes are drawn through the centroid of the wing section, one parallel to the chord line. One is then known as the horizontal axis and the other the vertical axis of the wing section. The moment of inertia of the skin of the wing section is then calculated about each axis. The sum of the inertias of the individual channels and the areas, times the distances to the axis squared will give the moment about any axis.

Using the formulas: $I_x' = I_y \sin^2 a\ I_x \cos^2 a$

$$I_y' = I_y \cos^2 a\ I_x \sin^2 a$$

The moment of inertia for the wing section is found with the axes in any direction where "a" is the angle through which the axes are rotated. Then using the equation for bending, Mc/I, the maximum fiber stress may be found that is imposed on the stressed skin at the point of the wing where the wing section is assumed.

The ideal conditions would be, of course, to have the fiber stress on the wing remain constant throughout the span of the wing. This can be done by varying the amount of thickness or plan taper, or both. If one is fixed, the other may be varied. However, equations may be set up with both as variables, and the shape of wing determined giving the most area with the least amount of metal. The maximum allowable P/A and fiber-stress value is determined by individual tests and these values, along with the load carried, and the area, are constant. So in reality one solves only for the plan and thickness-taper ratio.

The moment of inertia will change with the wing section, and will vary all along the span. This value must then be calculated at numerous points along the span and plotted in $^1/_4$ inches against the span in inches. The value of "c," the maximum distance the metal is from the neutral axis, is plotted in inches against the span in inches. This curve will, in nearly every case, be a straight line.

The fiber stress can now be calculated at any section along the span by the basic formula. The values of I, c, and M are taken from the charts at any given point along the span and f_s is calculated. If the fiber stress is calculated at a number of points along the span, and plotted against inches along the span the efficiency of the structure is seen. The ideal condition would be to have the maximum fiber-stress curve a straight line with no slope. In other words, the maximum fiber-stress should be as close to the maximum allowable, and constant throughout the span.

If these conditions are not met and it is not feasible to change the plan form or the thickness of the wing further, it is possible to correct by varying other things. For instance, the gage of the skin can be changed at various parts, thus changing the area of the individual sections and in turn changing the total value of I of that particular wing section. If channels or built-up sections have been used, the flanges could be made longer.

The maximum allowable fiber stress has been mentioned. Sometimes this value is rather difficult to determine. It is a function, in the wing, of the spacing of the transversal diaphrams or bulkheads, and the material and shape of the individual section itself. It was mentioned that tests should be made to determine the strengths of individual sections that make up the skin. These tests should be similar to those made by the Materials Division and published in Air Corps Information Circular No. 598. These tests consist of sections tested as columns. These columns are of various lengths and of different gages of sheet material bent up into shape, giving P/A and f_s values of the sections for various L/ρ ratios. In making these tests several considerations must be made of the installation of these sections in the wing. For instance, if the channels are to be riveted together, individual tests, as shown in the circular, are reasonable, but if the stressed covering on the wing is to consist of corrugations or of sections bent from one sheet, testing individual corrugations or sections will introduce losses due to the free edge of the test sepcimen. Whereas, in the actual wing, the corrugations or sections would have no free edges and consequently be stiffer and stronger... Hence, in tests of corrugations or a number of individual sections bent up from a single sheet, more accurate results can be made by having the test specimen include several individual sections.

Another problem is the type of fixity to be used during test. The Materials Division used knife edges for the ends in the tests in circular No. 598, but where these sections are to be continuous throughout the span, squared ends will probably suffice. This type of end will give a column fixity of between $1^1/_2$ to 2 according to the individual sections. All these results should be plotted. A very good method is to plot the P/A value of any given section against L/ρ ratios varying from lengths over radius of gyrations of 10 to 150.

Transversal bulkheads are heavy in any case as they must be designed to carry the vertical shear of the structure at their location. Hence, the fewest number that it is possible to use will help produce a light-weight structure. Since these bulkheads are used to fix the lengths of the columns of the skin the determining factor of their spacing is that consideration, but the longer the column the lower the allowable fiber stress, and the heavier the skin must be. So here again an optimum value of this spacing must be found to produce the lightest weight structure.

It is often noticed on the curves of the test specimen that the allowance stress on the specimen drops off rather slowly as the L/ρ ratio is increased up to a certain point. After this point the stress drops off very swiftly. This point will indicate a certain length for any given individual section and very often this length will have its effect on the choice of bulkhead spacing.

Now, having all this data plotted and compiled and having applied it to the design, there should result an ideal wing designed accurately to fill exacting requirements. However, the exact behavior of such a structure is not always known and perhaps the design data outlined above will not foretell the results. All strength calculations are based on the characteristics of the individual sections whether they are channels or corrugations, regardless of their form. When these are all fabricated and formed into a wing or fuselage their characteristics are very apt to change somewhat. To what extent depends on several things. The ratio of skin thickness to the cross section, and the ratio of the size of each individual section to the whole section are among the most important considerations. The effect of these items and the effect of the shape of the cross section of the whole structure will introduce a factor called a form factor. This form factor can only be determined by test. In most cases the structure will be stronger as a whole due to the form factor and especially so if there are several radii in the section. A section with circular form or an eliptical fuselage increases with calculated strength considerably, much more than most sections, with the exception

of those having reverse curves. Sections that have reverse curves usually are exceptionally rigid in bending.

These remarks have been directed toward structures subjected to pure bending. In the case of a fuselage where the rudder is mounted on top, a torsional load is applied as well as a shear load. The form again is an important item. In a perfectly round section or in the eliptical fuselage section, the structure will have great torsional rigidity but in a flatter section or one in which there are reverse curves the rigidity is not so great. It has been the authors experience in testing a number of sample sections of various forms and cross sections that a torsional load in addition to a shear load which carries bending, will not materially reduce the fiber stress and consequently will not greatly reduce the total load carried.

This monocoque type of structure has proved to be very interesting and the results of the effects in its direction have been very gratifying. With the presentation and confirmation of a few new basic formulas and a way of predicting accurately a form factor for this type of structure, no reason can be seen why, in the future, the monocoque type will not become extremely popular, and there will be better airplanes carrying bigger pay loads with a faster service.

Discussion

JOSEPH S. NEWELL.[2] The writer hesitates to discuss this paper because his own ideas as to the merits of all-metal airplanes are in such a turmoil that he finds it difficult to present them to his own satisfaction. Within the past three months he has discussed the relative merits of various materials and various types of construction with designers in the East while his colleague, Professor Niles of Stanford University, made similar inquiries in the West. The results obtained were compared during the recent aircraft show at St. Louis.

The following opinions, all of which were obtained from engineers and designers of considerable repute, will indicate the reason for the writer's hesitation and they will, he feels sure, indicate the almost total lack of dependable facts on which to base all-metal airplane designs.

Regarding durability of materials it was found that aluminum alloy properly heat treated plus anodic treatment plus an oxide primer plus enamel or varnish was as good as alclad, if not better. From another source it was found that aluminum alloys could not be protected against corrosion in service by any means so far discovered, and the opinion was expressed that stainless iron would be the solution for the durability problem. On the other hand it was stated that salt-spray tests on the thin-gage sheets of stainless iron, which were required to give a structure of practicable weight, indicated that its rate of corrosion was very nearly the same as for the aluminum alloys. The most definite opinion came from one of the most experienced designers in the industry. His idea was that for durability and economy of maintenance Sitka spruce was most suitable for aircraft.

Regarding the efficiencies of different types of construction some engineers advocated the use of stiffened flat sheets, some offered corrugated sheets. Opinions as to the behavior of flat sheets with stiffeners also varied. Some engineers maintained that the skin carried most of the load, others said that it was extremely inefficient and that the stiffeners were the structural members that actually carried the loads. The writer has test data from which one may draw either conclusion, and until the preliminary tests now being conducted by some of his students at Massachusetts Institute of Technology are completed he reserves his opinion.

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He does, however, wish to differ with the author on the subject of metal permitting a closer degree of accuracy in design than wood. He admits that wood has varying properties, but he would like to emphasize the fact that the variations are such that the strengths average above rather than below the specified values for wood that passes Army and Navy specifications. It is true that a great deal of understrength spruce has been purchased by various commercial manufacturers during the past three years, but that is less the fault of the material than the inspection. The admirable work of the U. S. Forest Products Laboratory which has given form factors, allowable stresses, and specifications for spruce and other woods, all of which contribute to dependability and lightness in conventional design, can be nullified neither by statements regarding the difficulty of obtaining test specimens in wood nor by expatiating upon the dependability of metal due to the fact that the physical properties are carefully controlled throughout its manufacture and treatment. At least it hardly seems possible that such should be the case so long as mild-steel tubes will show up satisfactorily in chemical tests yet have a modulus of elasticity as low as 18,000,000, nor while the properties of stainless steels vary over so wide a range as they do at present. The modulus of elasticity of one set of specimens with which the writer has become acquainted varied from 26,000,000 to 34,000,000 lb. per sq. in., and the modulus of elasticity, it is hardly necessary to remark, is usually regarded as the most constant of the properties of metals.

As for the virtues of full-monocoque versus semi-monocoque construction it is again difficult to state which is the better. A recent set of tests indicates that flat sheets stiffened along the edges, but having no intermediate stiffeners will carry practically the same load in compression regardless of their width. This is contrary to elastic theory. These tests were fairly comprehensive as they were run on four different materials, various gages were used, and widths ranged from 4 in. to 24 in. in 4-in. increments. All specimens were 24 in. in length along the axis of loading. Very meager data on stiffened sheets, on the other hand, indicate that with a 4-in. spacing of stiffeners, a flat sheet 12 in. wide will carry the same load that would be taken by three 4-in. sheets stiffened along the edges plus the load which would be carried by the stiffeners acting as columns. These data are extremely meager and further tests may necessitate revising the above conclusions, but if it is true one is forced to conclude that full-monocoque structures involving flat sheet in compression are impracticable. Here again nothing can be established definitely due to lack of test data.

Such data are sorely needed and until aeronautical engineers possess allowable stress data comparable to that now available for the design of spruce members or of steel tubular members it is doubtful that efficient all-metal structures can be proportioned without involving expensive tests on minor parts and on major assemblies. Civil engineering practice from which most methods are derived, has nothing to assist with the thin-gage, highly stressed materials. The author states that tests must be made to determine the strength of individual sections that make up the skin. They most certainly must. The Army Air Corps has recently completed a series of tests on corrugated sheet carrying compression, and has done some work to determine relative torsional stiffnesses of box-shaped structures simulating the spars, ribs, and skin of a metal wing from which the leading and trailing edges have been removed. It is expected that these data will be published shortly and they will be of great utility since they will furnish some allowable stress data. Parallel studies of stress distribution in similar box structures subjected to torsion have been made on Zylonite models in the photo-elastic laboratories at the Massachusetts Institute of Technology. The results are now being prepared for publication.

Service for Airplane Engines

By C. H. BIDDLECOMBE,[1] NEW YORK, N. Y.

Properly equipped service stations with competent mechanics are being established at some of the principal airports to take care of airplane maintenance. These have been found necessary by airplane-engine manufacturers in order to protect both their own interest and those of the customer. The work varies from a simple engine overhaul to complete rebuilding of large aircraft, and includes such items as packing and loading large airplanes for export shipment; designing and building gliders; dismantling, trucking, and reassembling airplanes for aviation shows; demonstrating airplanes to customers, and many other details, but not including the making of new engines and airplanes.

It seems probable that the major overhaul stations will eventually function on lines similar to those of present wholesale jobbers in the automotive industry with dealers doing all repair work, and buying necessary spare parts from jobbers in the territory in which they operate. Engine manufacturers look forward to having service representation at every active airport. The author tells of various methods and equipment used in overhaul work and the care necessary to ensure satisfactory operation.



THERE is no surer or quicker method of preventing sales than failure to provide good service facilities. Until 1929, there was practically no widespread organized system of customer service for any airplane engine on the market, due of course to the fact that the comparatively few airplane engines in use were either closely grouped in the hands of a few large customers or owned by individuals who were capable of doing most of their own service work. The manufacturers' share in keeping these engines running was thus based on a spare-parts supply department at the factory and the activities of a number of traveling service men who went from customer to customer as required. This method was in use by the Wright and Pratt & Whitney companies, for example, both these firms having service men on the factory payroll, but traveling over the country and remaining with various customers long enough to deal with sundry engine troubles and overhauls as they occurred.

The Wright company's service men earned well-deserved fame by their efficiency and far-flung travels during the years prior to 1929, being sent out to customers at the North Pole, Australia, Europe, South America, etc., and one such man accompanied Admiral Byrd on the South Pole expedition. In fact, to a very large number of customers, such men as Ken Boedecker, Eddy Mulligan, Doc Maidment, and Doc Kinkade *were* the Wright Company. This system of service representation by individual factory employees was satisfactory when only a few engines were in existence outside the army and navy flying services and the large fleet operators of aircraft. This centralization of engines in the hands of a few large customers resulted in comparatively little service work being required by factory men in assisting the mechanical staffs of such customers. The practice of training mechanics employed by these customers for a brief period at the engine factory also simplified the service question. With the great increase in private ownership of engines, however, it became at once necessary to adopt a different method of giving customers service, as the factory could no longer be expected to carry the very large number of men required to cover the whole United States and a large portion of other continents. Some idea of the increase in the demand for service from engine makers may be gathered from the figures for engine production during the years 1926, 1927, 1928, and 1929; these being 834, 1397, 3263, and 7378 engines, respectively.

The 1929 production increased 126 per cent over 1928, and was nearly nine times that of 1926. Of even greater significance to the service problem is the fact that the military engines produced in 1929 exceeded those for 1928 by only 448 engines, or 32 per cent, whereas the commercial engine production went from 1850 engines in 1928 to 5517 in 1929, an increase of 3667 engines, or 193 per cent.

The horsepower of the engines produced is also an important factor, as the smaller engines are obviously more likely to be owned by individuals or small operators needing widespread and efficient service. This need looms large when it is seen that of the 5517 commercial engines built in 1929, no less than 4113 were of 300 horsepower or less, with 3082 of them between 76 and 225 horsepower, exactly the class for which service is of great importance.

With these figures to consider, it becomes clear that the establishment of authorized and properly equipped service stations, manned by competent personnel, was urgently needed as a support to the limited number of factory employees that could be sent into the field. The first steps to this end were taken by the Wright Aeronautical Corporation as early as the end of 1927, when negotiations were started in Los Angeles to locate a complete service station there to take care of customers on the Pacific Coast. It had long been evident that it was impracticable to provide satisfactory service to an ever-increasing number of private owners and smaller fixed-base operators situated nearly 3000 miles from the plant at Paterson, New Jersey.

Preliminary arrangements were made with the Pacific Airmotive Corporation of Los Angeles to act as service representatives early in 1928. This firm thus was the first fully equipped

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[1] Consulting Engineer. Mr. Biddlecombe is British-born and took out his first papers for American citizenship on July 8, 1927. He served in the British Navy, 1906 to 1915; was transferred to Royal Flying Corps May 6, 1915; served as pilot and on administrative and executive duties 1915 to 1918 in England, France, Egypt, Italy, and Palestine. He was detailed to staff of Controller-General of Civil Aviation (Sir F. Sykes) in December, 1918, with acting rank of Major, Royal Air Force. From December, 1918, to January, 1926, Mr. Biddlecombe was engaged in the development of commercial aviation throughout Europe and the British Empire. He came to the United States in February, 1926, on leave of absence, and joined the Colonial Air Transport, then being formed, as Operations Director in March, 1926. He organized and managed operations of Contract Air Mail Route No. 1 from April, 1926, to August, 1927. He made a tour of the United States and Eastern Canada for the Wright Aeronautical Corporation to survey the whole aviation industry to determine the engine market for 1929 and 1930, remaining with the Wright company in charge of sales promotion and advertising until November, 1929. He has since continued his own business as aeronautical consultant engineer.

Presented at a meeting of the Metropolitan Section of the A.S.M.E., New York, N. Y., May 8, 1930, sponsored by the Aeronautic Division of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

independent organization to act in the capacity of airplane engine manufacturers' repair depot, as far as the author is aware, who from March to October, 1928, had an interesting and pleasant assignment to travel over the greater part of the United States for the Wright company in order to recommend locations and corporations to act in a similar capacity at other places.

During the course of these journeys, the pressing necessity for local service depots by any engine manufacturer hoping to remain in business became obvious to a marked degree. Trivial engine troubles that were curable in a few hours by an expert factory-trained mechanic assumed alarming proportions and a greatly exaggerated seriousness when the nearest point of contact with skilled service was a factory some hundreds of miles distant. The receipt of frantic long-distance telephone calls, hundred-



FIG. 1 AERIAL VIEW OF BUILDINGS AT ROOSEVELT FIELD, NEW YORK, OF AIR ASSOCIATES, INC.

word night letters, and finally acrimonious correspondence from the customer caused much worry and considerable expense to both engine maker and customer. In an appreciable number of cases of engine trouble, it was found that the customer was solely to blame for his troubles. One airplane owner complained long and bitterly to the factory that his engine was completely useless, that something was broken internally, that it vibrated terribly, and that he was entitled to a new engine free of charge at once. A factory service expert found that the propeller on the engine was very loose, the customer's mechanics having forgotten to tighten the locking nuts. Within ten minutes the engine was running perfectly.

Loose spark plugs, some even falling out in the air, loose oil, and gasoline pipes and joints, empty oil and gas tanks, unsuitable low-grade oil and gasoline were some causes that produced scores of hurry calls to the factory, and, of course, some unique alibis from careless or insufficiently educated mechanics. Another particular incident of instructive type occurred when a famous transatlantic pilot and another almost equally well-known set off from New York to Washington. When nearing Baltimore, the oil temperature rose alarmingly and the oil pressure decreased with equal rapidity. Good fortune and good flying landed the airplane on the Baltimore airport. Inspection of the oil tank by pilot No. 1 brought the exclamation, "We are out of oil, didn't you put any in this morning?" The reply was "No, I thought you did."

Instances of this kind were frequent, but it is probable that the chief troubles experienced by the majority of engine makers were caused by faulty installation of engines into airplanes. Again and again, it was found that in the desire to avoid work and save time, airplane factories installed engines in their planes in such a way that constant engine troubles were inevitable. The engine cowling made by the airplane manufacturer was frequently designed so that it was impossible for a cooling airstream to reach the cylinders; oil tanks were placed so that the oil was always overheated on warm days; parts of the engine were removed, particularly carburetors, and replaced so carelessly that they either did not function or fell off after a few hours' running.

Another prolific source of trouble to the engine maker was the use of unsuitable oil and gasoline, usually due to the engine owner having mistaken ideas on the matter of real economy in the purchase of these materials. In one case of this kind, the cylinders of an engine were worn 0.025 in. oversize after 700 hours' use. This rapid wear was traced directly to the use of a very poor oil and failure to clean the oil system and provide fresh oil at reasonable intervals. As a contrast to this, there are a large number of engines which have had over 1000 hours' service with no appreciable cylinder wear, but good oil and proper care has been used in their operation.

One of the chief troubles experienced with poor oil is the formation of an excessive quantity of sludge and carbon, leading to the blocking of an oil hole or the complete clogging of an oil screen. The author's experience while operating an air-mail line included an example of this nature; the result was major



FIG. 2 ASSEMBLING A THREE-MOTOR FORD AIRPLANE IN THE REPAIR SHOP OF AIR ASSOCIATES' PLANT AT ROOSEVELT FIELD, NEW YORK

damage to an airplane and the wrecking of an engine due to the seizing of a cam-ring bearing, which followed the stoppage of oil passing to the bearing through small holes which had become blocked by sludge and carbon particles. A particular make of oil which met the recognized standard specifications was being tested under actual service conditions, and failed to stand up. It is probable that the asphalt-base oils are more liable to this excessive carbonization and sludge deposit than those refined from a paraffin base; at least, such has been the author's experience. Gasolines unsuited for aircraft engines, particularly air-cooled engines, have also produced many service difficulties, especially a few years ago when high-grade aviation gasoline was not widely distributed. Conditions are much better today and excellent gasoline can be obtained at almost every airport. There is, however, still a danger of burning out piston heads and wrecking an engine if poor fuel is used in an emergency and this still happens occasionally.

The work of the engine service stations includes advice and

instruction to private owners on the very important question of using only proper fuel and oil and avoiding the risk of injury to life and property in attempting to save a few cents per gallon on these items.

A further minor source of trouble to the engine maker was the use of unsuitable accessories on engines and the maladjustment of proper accessories. Examples are the use of spark plugs of a type that are not approved by the engine maker and which fail constantly in service, also the use of poorly made high-tension wiring when this is renewed after an overhaul. In both cases the ignition system fails to function properly, and unless the mechanic in charge of the engine is a thoroughly experienced man, the engine maker is usually blamed for the trouble.

Many engine makers recommend and fit to their engines as standard certain makes of starters, generators, magnetos, and carburetors. In some cases the approved makes have been mal-adjusted and thus failed to operate correctly. In such cases, the engine owner invariably turned first to the engine manufacturer as the source of his troubles, and the responsibility of getting the engine running properly was generally accepted by the manufacturer, but this acceptance of the results of lack of knowledge on the customer's part could not, of course, be continued indefinitely owing to the great increase in the number of engines sold.

These general considerations emphasized most strongly the immediate need for a wholesale expansion of engine service facilities. Engine manufacturers were obliged to hasten the completion of their plans to provide universal service and widespread education in the proper use of their products in order to protect both their own interests and those of the ultimate customer.

The basic question to be answered when contemplating the



Fig. 3 Main Building of Air Associates, Inc., at Roosevelt Field, New York

setting up of a service organization is obviously the extent and location of the market for the product to be serviced. A map with pins at points where aeronautical activity is most prevalent will show many cases where towns are close together. It is equally important to know the number of airplanes in use at each particular airport. The map is divided into areas of varying size, each to be serviced by the town where flying activity is greatest in that area, and which may or may not be approximately in the center of the area. In the case of Canada, the Canadian Wright Company, manufacturing engines under license from the Wright Aero Corporation, provides service for that country, with headquarters at Montreal. A station at Mexico City is entrusted with service matters in Mexico and adjoining territory, while South American customers rely on service and spare parts from traveling factory employees pending an increased use of the airplane in that part of the world.

Throughout the United States 10 major service stations are located at Chicago, Kansas City, Boston, Philadelphia, Los Angeles, Atlanta, St. Louis, Fort Worth, Cleveland, and Memphis. In addition, there is a very completely equipped service division at the main plant of the company at Paterson, N. J. Each of these depots is equipped to give a complete engine overhaul, including cylinder grinding to oversize, etc., and a complete stock of spare parts is carried which is replenished from the factory when necessary. As a general rule, one or more of these 10 major stations is within three hours' flying time of any airport on which Wright engines are normally based. In some few cases, the base of operations is further removed than this distance, but under these circumstances, the operator himself is capable of effecting all normal repairs, and can also give service in emergency to other individuals who may be in trouble in the area near his base.

In the selection of representatives, it is necessary to use the greatest care as to their financial stability, experience in busi-



Fig. 4 Chicago Service Plant of Air Associates, Inc.



Fig. 5 Front of Service Building at Chicago Municipal Airport

ness of a nature somewhat similar to that under discussion (such as the operation of a high-grade automobile repair agency), knowledge of aeronautical matters generally, and last but most important, integrity of character. This latter point is capable of exercising a wide influence on the continued success of a manufacturer's sales in any business where a service system is used, but is particularly important to the airplane-engine maker. The results of careless work, unfair prices, and use of "pirate" spare parts not up to specification in order to increase profits temporarily, are likely to be disastrous both to the customer thus cheated and to the engine manufacturer.

It is trite to say that the oceans of the air, like Father Neptune, are unforgiving opponents to those who make mistakes; there is no back door and few alibis when an airplane engine fails at a critical moment, so that human lives and the welfare of great engine factories are to a great extent dependent on conscientious work by the service stations. It is of interest to note in passing that the airplane accidents caused by power plants in the last six months of 1928 and first six months of 1929 were about 16 per cent and 18 per cent of total accidents, respectively. These figures leave room for improvement, but indicate that the average man entrusted with engine maintenance and overhaul is doing a good job of work.

Having settled exactly where the service stations are to be located, the next step is to agree on the terms of a contract with the personnel to whom a service and spare-parts franchise is to be given. This stage of affairs should not be hurried nor

regarded lightly by the management of an engine factory who wants to retain customers. In addition to the considerations previously outlined, there are numerous questions of policy to be settled in arriving at a contract mutually satisfactory to the service agency, the factory management, and the customer.

Of these, the scale of discounts to be applied in the distribution of spare parts is perhaps of the greatest importance and likely to cause most difficulty in the absence of a clear understanding from the start of business relations. It will be appreciated that the 10 major overhaul stations given cannot provide a sufficiently wide distribution of spare parts to satisfy the needs of aviation as a whole; it is necessary also to place stocks of spares at numerous other airports throughout the country, together with facilities for minor overhauls. In the case of the Wright company, about 100 such smaller depots are in operation, and for convenience, the 10 major stations may be referred to as distributors and the minor stations as dealers.

Furthermore, there are to be considered the large transport companies operating a fleet of aircraft with a great number of one particular maker's engines; in addition, the small fixed-base operator with a smaller but still appreciable quantity of engines of one make also needs thought in drawing up a scale of trade



FIG. 6 ENGINE OVERHAUL DEPARTMENT OF AIR ASSOCIATES SERVICE PLANT AT CHICAGO MUNICIPAL AIRPORT

discounts. At the end of the line comes the private owner and the operators of a very few engines for whom the manufacturer's list price on engines and spare parts is established and maintained. The primary object in reaching and maintaining a scale of discounts is, of course, to protect the distributor and dealer against unfair competition, and to allow a sufficient price differential to permit these agents to remain in business while giving efficient service to the factory customers. It is evident that the fairly impressive investment and superior type of executive and mechanical personnel required in the conduct of a service station business cannot be attracted unless a reasonable margin of profit is available. In this connection, it should be noted that the caliber of men and standard of work required in the manufacture and overhaul of airplane engines is much higher than that normally found in other industries. As stated previously, the result of cutting costs too far by using low-grade labor or material is more likely to be a dead customer than one who is merely dissatisfied. Per contra, list prices must be competitive and large customers permitted to exercise their quantity purchasing powers if sales are to remain satisfactory.

There is ample precedent, both good and bad, to guide in drawing up contracts of this kind between airplane-engine builders and their distributors, so that one can pass over the existing practices in this field by remarking that the discount policies in force by the Wright, Pratt and Whitney, and Curtiss companies, for example, are roughly comparable each to the other and are based on the experience of automobile makers.

A second important factor in the relationship between the factory and its representatives is the extent and financial detail of free service and replacements to customers. This portion of a distributor's business is likely to be of fair magnitude at times, particularly immediately subsequent to the introduction of a new model engine. A clear agreement as to the division of expenses incurred in this type of work is of value when discussing matters prior to the granting of a distributor's franchise. The remainder of the items involved in reaching agreement on the franchise are of a more or less routine business nature, such as terms of payment and credit, reservations regarding governmental purchases of engines and parts, re-acceptance of distributors' stocks on termination of contract, etc.

Arriving at the actual operation of a distributor's or dealer's franchise, it is well to note that up to the present, no service station of either major or minor type has succeeded in obtaining sufficient volume of business from engine overhaul and repair solely. From the brief experience gained it appears that the authorized service stations which have been in operation during the past two years have obtained from 5 to 25 per cent of their gross business from the overhaul, etc., of any one make of engine; one of the most active of these dealers has received only 5 per cent of his gross business from engine overhaul work. This situation will improve when the engine factories further develop the sale of complete engines by distributors and dealers as replacements. Up to now engine sales have been made almost exclusively direct from the factories.

As a result, the engine factory representatives appointed have in many cases been engaged in the sale of airplanes, equipment, flying clothing, maps, airplane repair work, etc.; obtaining as many agencies as possible for the sale of the multifarious materials and supplies needed on and around an airport. Other overhaul franchises have been granted to airplane operators, air mail contractors, etc., as an addition to their existing activities. Almost the only type of airport business that engine service stations appear not to have yet entered is the sale of the ubiquitous "hot-dog."

An an example of the major depots now established, one may take that of Air Associates, Inc., at Chicago, Ill. This organization has invested approximately $120,000 in buildings and equipment at the Chicago municipal airport, of which about $60,000 represents building costs. Further expansion is provided for when increased business warrants the erection of more buildings. At present this station is composed of a building 80 ft. by 160 ft. of two stories, with a hangar entrance facing on to the flying field. The airplane accommodation for service and storage purposes covers about 5280 sq. ft. and provides room for about 10 average-size aircraft; the remainder of the building is divided up into workshops, offices, etc., as follows:

	Sq. ft.
Machine shop	1890
Engine test-room	320
Airplane repair and wood-working machinery	1890
Welding and sheet-metal work	880
Doping and painting shop	800
Oil and paint store	200
Stores and stock room	1300
Instrument repair room	620
Office accommodation	1600

In addition to the repair facilities noted, rest rooms, locker, and shower-bath rooms are provided for the comfort of customers, and all equipment needed for flying purposes is on sale in the

showroom. This includes flying clothing, maps, helmets, goggles, all flying instruments, in fact, everything required by pilots and passengers. The basic policy of this service station is the provision of every possible requirement in the operation of aircraft, including the supply of airplanes of various makes for which Air Associates, Inc., holds the sales agency. This somewhat comprehensive scope of the corporation's business is necessary to produce the annual turnover which cannot be obtained from the operation of an engine repair station only.

During the busy season in the summer months, the employees number about 30 people, including branch manager, sales manager, service superintendent, aircraft repair foreman, storekeeper, mechanics, and office staff.

The machine-shop equipment includes the usual drilling, grinding, and turning machines, with presses, washing tanks, engine stands, air-pressure cleaning system, and a complete range, of the special tools designed to simplify the overhaul of Wright engines.

It is estimated that the total investment required in installing a duplicate set of machine and other tools would approximate $14,750 of which $1750 would represent special tools for Wright engines. As an average for the 10 major overhaul stations listed, it can be accepted that an investment of from $35,000 to $50,000 is needed to establish an engine repair depot capable of handling every kind of overhaul. This does not of course include the erection of hangars or buildings for more extensive operations such as those conducted by Air Associates, Inc.

This corporation also operates at Roosevelt Field, Long Island, similar service that includes everything from airplane storage, and the sale of flying requisites, to the complete overhaul of of airplanes and engines, and the rebuilding of any airplane damaged by accident.

A brief description of the operation of this station is of value in the study of service stations generally. The storage hangar is situated on the edge of the flying field, in close proximity to the field operations office, weather bureau, telegraph, press offices, and the Department of Commerce office. It provides transient storage for visitors to the field and permanent storage to sportsman-pilots and to business houses who use airplanes for business purposes solely, utilizing Roosevelt Field as their home base. At the present time, about one-third of the airplanes in the hangar are used for strictly business purposes.

Lockers are provided for the convenience of the pilots, and the hangar rental charges include such services as inspection of the airplane and engine before and after each flight, inspection of gas and oil, and re-fueling at the customer's request, movement of the airplane from the hangar to the line, starting the motor and having the airplane ready, inspected, re-fueled, motor running, and warmed up, at any time specified. When the flight is finished, the owner leaves the ship on the line; it is taken in charge by the line crew and returned to the hangar, where it is inspected and wiped down. The pilot is thus relieved of all the routine work of preparation for flight; he simply telephones, requesting that his ship be on the line with the motor running and warmed up at a certain time.

All airplanes in this hangar are subjected to regular close inspection. The flight superintendent reports immediately to the field manager every detail, no matter how small, that would in any way affect the safety or operation of the airplane. If the detail is small, and can be corrected easily, it is corrected without notifying the pilot, and at no charge to him. If the defect is of some relative importance, the pilot is notified immediately and the defect is remedied as soon as his approval is received. Under no circumstances is any ship allowed to fly unless every detail is correct and in accordance with Department of Commerce regulations.

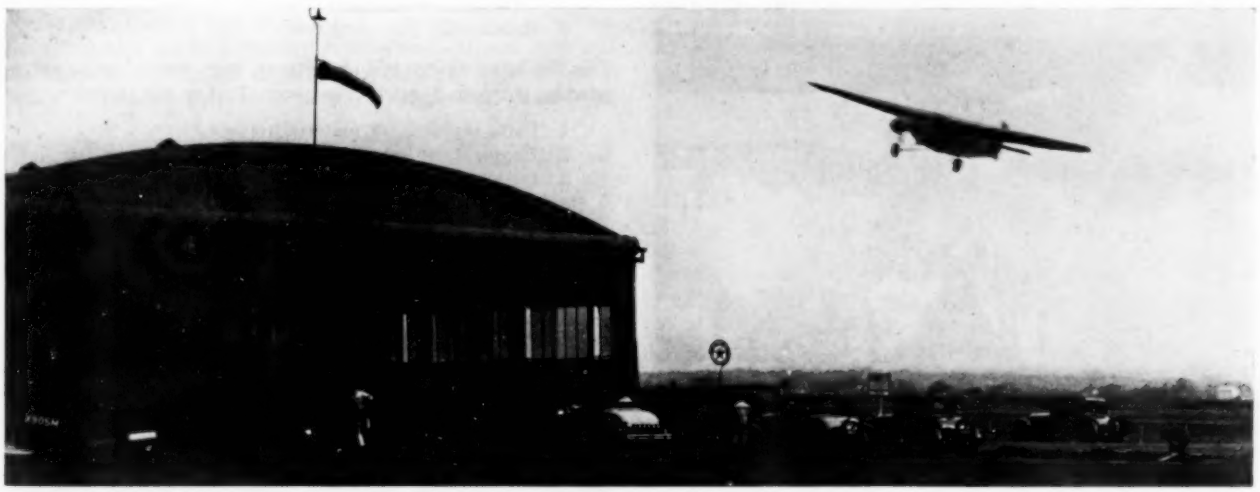


FIG. 7 WRIGHT SERVICE HANGAR AT TETERBORO, NEW JERSEY

No service work is done in this hangar with the exception of such work as washing, waxing, greasing, etc., all mechanical work, either on the airplane or the engine, being carried out in the service hangar. This hangar is situated away from the line, but in proximity to the line hangar, and can be divided roughly into four units, (1) airplane repair department, (2) welding and sheet-metal department, (3) engine-overhaul department, and (4) machine shop. One stockroom serves both the service department and the merchandise department, material being requisitioned by the service department from this stockroom as it is used on the various jobs. This stockroom also has in it the service department tool-crib, which contains all the standard small tools used by the shop, and all the special tools for motor, carburetor, and magneto overhaul. Special tools are mounted on tool boards, each board containing the special tools for one model of one make of engine. One board, for instance, has mounted on it all special tools for the Wright J-5, another board all special tools for the Wright J-6. Special tools for Pratt & Whitney "Wasp" and "Hornet" engines are kept in a large case, each tool having a separate compartment or receptacle. Com-

plete special tools for the overhaul of accessories, such as Scintilla magnetos, are kept in locked cases that can be requisitioned out as a whole and taken directly to the magneto test bench, carburetor overhaul bench, etc. All tools are taken from the tool-crib on checks; provision is made beside the position of each tool on the tool board for deposit of the check until such time as the tool is returned. Silhouettes of the tools on the tool board are colored differently for the different types of engines; thus the silhouette for a Wright J-5 valve-guide puller is black, and the puller has a black marking on it; the silhouette for a J-6 valve-guide puller is red, and the puller has a red marking on it, etc. Beside the silhouette of each tool, also, is a tag giving the name and manufacturer's part number, and it is only by means of a careful system of this kind that lost tools and needless expense can be avoided.

The airplane repair department has a two-ton hoist and

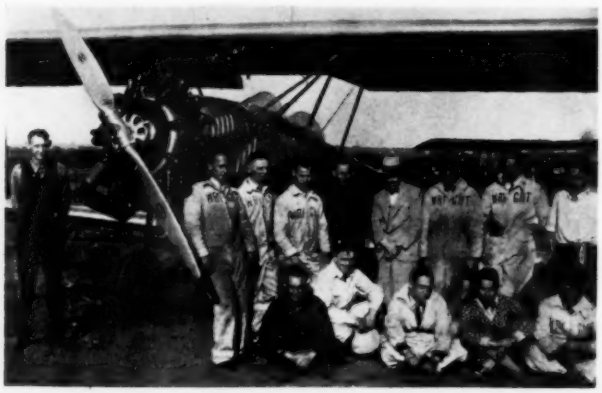


FIG. 8 WRIGHT SERVICE CREW

trolleyway running its whole length with all machinery arranged at the side, so that the maximum floor space is available. It has complete wood-working machinery, including bandsaw, universal circular saw, jointer, sander, etc. Glue mixers, sewing machine, eyelet set, and all necessary equipment for this type of work is provided; this department being equipped to do everything from putting a small patch on a wing to rebuilding the whole wing itself. Complete doping and painting equipment is installed, including cup-type guns for touch-up work and pressure-tank equipment for spray doping.

The engine overhaul department, exclusive of machine shop, occupies a space about 25 by 75 ft.; benches with maple tops run along both sides of the space and the propeller balancing equipment is located at one end. The machine shop occupies an equal space, with a large door opening from the overhaul department into this shop.

The engine overhaul department is essentially a tear-down, inspection, and build-up department. It contains no fixed tools with the exception of special master-rod reaming tools, lapping plates and fixtures, microscopes and other inspection tools, etc. It contains motor stands, on which the engines are torn down and rebuilt, and motor parts stands, on which the parts of each engine are placed and kept together.

The machine shop is equipped with all necessary machine-tool equipment, including a tool-room lathe, large and small drill presses, grinding wheels, wire-wheel and buffing stands, air-compressor, arbor presses, sand blast, spray-cleaning booth, magneto test bench, Scintilla magnet recharger, and the carburetor test bench.

A better idea of the way the equipment is used will be obtained by considering the various steps in the overhaul of an individual engine. One of the first things of interest to the customer is the price. Air Associates, Inc., was one of the first organizations to establish flat labor charges on all operations possible, and it now quotes flat labor charges on checkovers, top or complete overhaul, magneto overhaul, or any other standardized job on any of the popular models of engines. Work of complete overhaul is done on a flat labor charge; this charge will include:[6]

1 Teardown, cleaning, and inspection of all engine parts
2 All labor to recondition engine parts, with the exception of such operations as re-sleeving cylinders, re-grinding cylinders, installing valve seats, replacing rocker-boss studs, reconditioning cam and hub assemblies, installing new valve guides, etc.
3 Re-enameling all engine parts
4 Inspection and overhaul of magnetos, including re-charging the magnets
5 Reassembly and block test.

This flat labor charge includes grease, rags, emery cloth, valve-grinding compound, safety wire, etc. It does not include:

1 Parts replaced, or material replaced
2 Removal of the engine from, or installation in the ship
3 Carburetor or starter repairs
4 Gasoline or oil used for testing.

When the customer authorizes the work, he signs a numbered work order. One manila copy of this work order remains in the office of the service manager; the job number on this work order is the number on which men working on this job charge their time and against which they requisition necessary material from the stockroom. Two copies of this work order go to the office, where labor other than that done on flat charges is entered on both, and where material requisitioned on the job is also priced and entered. One of these copies is for the customer, and is given to him with the invoice when the job is finished. The other copy is a file copy, and on this copy all labor is entered, whether the job is done on a flat price or not. This gives a check as to the actual cost to Air Associates, Inc., of the work that they are selling at a flat price.

When the engine comes into the overhaul room it is torn down completely, everything of interest as to condition being noted during the process. A complete record sheet for each engine is kept up by the engine overhaul foreman from the time the engine leaves the plane until it is re-installed. This sheet contains spaces on which are entered the actual size, or the amount of wear on each part of the engine, with space for notations as to the parts replaced. A copy of this record is given to the customer when the job is finished, or is used to show him what parts need replacement during the progress of the work.

As the parts are removed from the engine they are placed in a special engine-parts rack that was developed to meet the requirements of aircraft-engine work. Each of these parts racks has space for the parts of one engine; the engine serial number is marked on a card at one end of the rack, the parts thus being kept together during the time the engine is in the shop. These racks have several shelves, and have dust-proof curtains that cover all sides; these curtains are rolled up during the time that the parts are being worked on.

The parts racks are mounted on rubber-tired rollers, and after the engine is completely dismantled the rack is run to the sand-blast and cleaning booths. A special sand-blast removes all carbon from the cylinder head and inlet and exhaust passages; a sleeve that fits the cylinder bore tightly, and plugs that fit into the valve guides, protect these surfaces during this opera-

tion. This special tool greatly reduces the time necessary for this cleaning operation, and does such a thorough job that the interior of the cylinder head is put in the same condition as when new. Another special sand-blast strips paint from the cylinders.

The parts are then thoroughly cleaned, either with Oakite or in a special ventilated booth with air pressure and gasoline. Practically all this cleaning equipment was developed to meet aircraft-engine requirements. After cleaning, the parts that can be buffed are buffed on a $1^1/_2$-hp. extended-spindle buffer; after this buffing operation various parts, such as wristpins, connecting rods, crankshaft, etc., are given a final cleaning with metal polish, so that they are in perfect shape for inspection.

When the parts have been cleaned and polished, they are brought in on the parts rack to the inspection bench, where every part is measured for wear and where all vital parts are subjected to microscopic inspection with a binocular microscope. Fillets are etched in some instances to make incipient cracks more evident, and every part is considered not only from the standpoint of being within the allowable limits of wear but also from the standpoint of being in perfect physical condition.

Many special gaging tools have been developed for this in-



FIG. 9 ENGINES DISMANTLED READY FOR INSPECTION

spection work. "Go" and "no-go" plug gages for valve guides, for instance, are made with flat sides, so that out-of-roundness can be gaged. In every instance inspection and rejection is based wholly on the permissible limits of wear set up by the engine manufacturer. In aircraft-engine work there is no place for the interjection of engineering ideas from the field except when such ideas have been thoroughly worked out by the engine manufacturer. In no case will Air Associates, Inc., deviate from the limits and instructions set up by the engine manufacturer, even at the request of a customer. Every engine turned out, therefore, is exactly as if turned out by the factory.

After the necessary replacements are determined by inspection, the rebuilding process starts. Valve seats are re-cut with special self-centering cutters, the valves are re-faced on an electric valve lapper that has been specially adapted to fit airplane-engine valves. In order to bring down the cost of an overhaul, so that reasonable flat rates can be quoted, it is necessary to utilize all possible labor-saving equipment. Where no such equipment has been available for aircraft-engine work, Air Associates, Inc., has developed it, or has adapted standard equipment so that it is suitable.

When new parts are used, each is fitted carefully so that the working clearance will be toward the lower side of the permissible limits set by the engine manufacturer. Plug gages, either purchased from the engine manufacturer, or made up

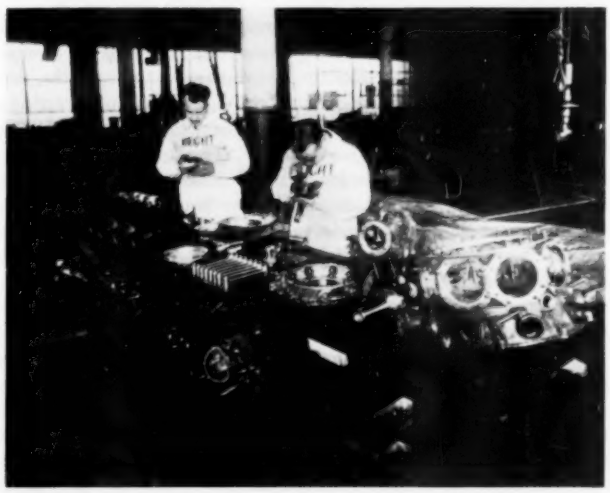


FIG. 10 INSPECTING ENGINE PARTS DURING OVERHAUL

specially, are used wherever possible. Standard tool-room micrometric equipment, cylinder gages, etc., are used freely.

The cylinders, rocker-boxes, crankcase, etc., are re-enameled, rings are fitted to the ring-grooves by carefully lapping to the desired clearance, timing is checked carefully, and when the engine is finally set up it is essentially in the same conditions as when new.

While the engine is down, the magnetos are completely dismantled and inspected, magnet readings are taken on a magnetometer and the magnets are recharged if necessary. After the magnetos are re-assembled, they are given a standard series of tests which cover a range of speed from their coming-in speed, which must not be above a certain fixed maximum, to 4000 r.p.m. Since the requirements for testing an aircraft magneto cover a much greater speed range than the requirements for an ordinary magneto, it was necessary to develop special testing apparatus for this work.

The carburetor is also completely inspected, any unduly worn parts replaced, and the float level checked before replacement on the engine. After the engine is finally re-assembled, valve timing set, tappet clearances checked, etc., it is taken to the engine test stand and given a run-in for at least 8 hr. During this period the engine is run in very carefully, and is under the supervision of an engine mechanic at all times. The instrument board from the test stand is located in a small house having windows on all sides, so that the man in charge of the engine can comfortably observe its operation during the run-in period.

During, and after, this run-in period adjustments are checked and the engine is ready for installation in the airplane after a final thorough check and inspection by the service superintendent.

The service hangar also contains an instrument repair room complete with all necessary testing equipment, where flying and engine instruments are inspected, tested, and repaired as required. In addition, a parachute packing-room in charge of an expert undertakes the inspection, repair, and periodic repacking of parachutes and the proper fitting of parachute harness.

The airplane repair division includes a welding and sheet-metal department equipped to handle any work from a minor repair to a steel axle or the making of a small section of engine cowling, up to the rebuilding of a large all-metal transport airplane. Complete stocks of duralumin sheet and tubing, Alclad flat and corrugated sheet, aluminum sheet, and tubing in various tempers, duralumin and aluminum rivets, etc., are on hand at all times, together with supplies of S.A.E. 1025 carbon steel and 4130X specification chrome-molybdenum sheet and tubing.

All repair and overhaul work is carried out in strict accordance with the regulations of the Department of Commerce, and is always subject to inspection by the representatives of the department. An engineering division provides data and drawings when it is necessary to submit any particular repair job for Department of Commerce approval. The merchandising division of the station includes a retail store and a wholesale department that supplies dealers in the trade territory of the station, shipments being made to all parts of the United States and to many foreign countries. Every conceivable practical airplane necessity is handled by this merchandising division, from cotter-pins to complete airplanes, including spares and parts for engines, starters, generators, magnetos, carburetors, propellers, etc. A card-index system is used for a perpetual inventory record and control. One of the major problems in the operation of such a widely diversified business is the gathering of personnel capable of handling the varied jobs that come to hand. These jobs range from a simple engine overhaul to the complete rebuilding of large aircraft; and include such items as packing and loading large airplanes for export shipment; designing and building gliders; dismantling, trucking, and reassembling airplanes for aviation shows; demonstrating airplanes of all types to customers; and, in general, doing everything that has ever been done in aviation with the exception of making new engines and airplanes.

From this review of the activities of a typical airplane-engine service station, it will be seen that a successful business of this kind must go far afield from the work of engine overhaul only, in order to maintain a satisfactory total volume of business, and it is evident that the further increase in airplane sales to private individuals will produce changes in the present service and distribution system for engines.

It seems probable that the major overhaul stations will eventually function on lines more nearly comparable to the wholesale jobbers in the automotive industry, with the present dealers carrying out all repair and overhaul work and buying the necessary spare parts from the jobber in whose territory they operate. To a certain extent, this is the situation at the moment, with the exception that of the 100 or so dealers holding the Wright franchise, for example, very few are equipped to undertake a major overhaul involving the need of expensive machine-tool equipment. These smaller service stations are competent to carry out minor overhauls and to keep engines in first-class running condition for the private owners lacking the time or inclination to maintain their own engines by a regular periodical inspection and adjustment. The work of the small stations is in general limited to a "top overhaul" comprising removal of all cylinders, de-carbonizing, valve grinding, replacing piston rings and small worn parts, etc., but not including such items as cylinder grinding, renewing main bearings, etc. The equipment needed by these minor stations is obviously less elaborate and expensive than that employed for major overhauls, and all the necessary facilities can usually be installed at a cost of from $10,000 to $15,000, including a stock of spare parts, but excluding any structural building costs.

The ultimate hope of most engine manufacturers is to have service representation on every active airport in the country, in the same way that popular makes of cars can be properly cared for by expert personnel in almost any town in the United States. This expansion cannot be realized, however, while it remains essential for even minor service stations to invest, say, $15,000 in order to obtain the limited amount of engine overhaul work at present available, particularly if these depots are allowed to undertake service work for only one make of engine, as is generally the case. The solution to the problem of broadening the scope of engine service in advance of a very great increase in the number of airplanes in use would thus seem to be the elimination of the policy of granting dealer franchises only on the basis that the dealer does not accept a similar franchise from a competing engine maker. It is improbable that any one make of engine, or the varying makes produced by any one manufacturing combine, can become sufficiently numerous to produce a really satisfactory volume of service work until a number of years have elapsed. To offer the benefits of expert assistance for any make of engine on any airport will obviously react favorably on airplane sales generally, and this can be accomplished more rapidly by an agreement between manufacturers whereby the best qualified repair depot in a given territory would be authorized to handle a group of varying makes of engines. This would in turn tend to simplify the distribution of spare parts, as the major repair stations would fulfil the functions of jobbers to the whole industry, carrying a large enough stock of spare parts to supply the dozen or more dealers in their territory. At present, the distribution of spare parts is by no means satisfactory to the average engine owner; even the best organized of the service systems suffer from delays and unwarranted expense in getting spares to the point where they are needed. In part, this is due to the past instability of engine production; few distributors or dealers are prepared to carry a large stock of parts that may become obsolete in a few months due to changes in engine design. In addition, few engine factories have been able to supply sufficient spares to build up a reserve in the distributors' stockrooms that is needed for carrying on the work.

With the greater stability now evident in the industry, these conditions no longer obtain, and it should be possible for an engine owner to obtain any normal requirement without having to wait for it to be provided from a far-distant factory.

There is no question that one of the most important factors in the continued growth and prosperity of the aviation industry is the service that engine manufacturers give their customers; it is thus reassuring to see that great combines such as the Curtiss-Wright, United Aircraft, and Bendix Aviation organizations are doing all in their power to establish service facilities wherever airplanes fly.

AER-52-15

Silencing the Airplane[1]

By H. L. DRYDEN,[2] WASHINGTON, D. C.

In this paper the author tells how sound is produced and defines intensity. Experiments and investigations made by the aeronautics branch of the Department of Commerce are described. These deal with the theory of engine mufflers and sound-proofing of airplane cabins. There appears to be no prospect that a device will be developed which at the cost of a pound or two of additional weight will eliminate the noise.

THE problem of silencing the airplane is one of great current interest. The noise nuisance is generally considered as one of the most potent influences tending to discourage the use of airplanes by the general public. Editorials are written about it, some people think there ought to be a law prohibiting airplanes from making noise, and directors of research look with longing eyes toward the problem, the solution of which would bring fame and fortune to them and to their organizations. Unfortunately the problem is not a simple one, and no master key to its solution has been discovered.

The aeronautics branch of the Department of Commerce through its research division in the Bureau of Standards is greatly interested in the possibility of developing practicable methods for the reduction of the noise of airplanes, and has already carried out investigations in two directions—namely, on the theory of engine mufflers and on the sound-proofing of airplane cabins. The work on sound-proofing is conducted in the sound laboratory of the bureau by V. L. Chrisler and W. F. Snyder under the general direction of Dr. Paul R. Heyl, and some of the results have been published in the May, 1929, issue of the Bureau of Standards Journal of Research. The remarks on sound-proofing in this paper are based on the work of Messrs. Chrisler and Snyder.

The purpose of the present paper is not to announce a complete solution of the problem, but rather to present from the viewpoint of a physicist a more or less ordered analysis and a statement of the ideas guiding the work at the Bureau of Standards. In the working out of this analysis the author wishes to acknowledge the wholehearted cooperation and assistance of Dr. L. J. Briggs, Chief of the Aeronautical Research Division.

The sensation of sound is produced by the presence in the air of compressional waves set up by some vibrating body. It is found by experiment that if the motion of the vibrating body is periodic, a musical sensation is produced and the pitch of the sound may be correlated with the number of vibrations of the source in unit time. If the vibrations are not periodic, a sensation of noise is produced. In the case of an airplane, several sources are present, and the resulting motion of the particles of air is an exceedingly complicated one. While it is possible to recognize certain predominant frequencies in the noise of an airplane, and while it is always theoretically possible to consider any complex vibration as the result of a large number of simple vibrations, this paper will be limited for the most part to the second attribute of sound, namely, the loudness.

Experiment shows that when the loudness of a sound increases, the amplitude of the vibration of the source and of the compressional waves in the air also increase. In any small volume of the air through which sound waves are passing there is a certain amount of energy which can be measured by suitable instruments. The amount of energy per unit volume is a measure of the intensity of the wave motion.[3] If the intensity increases, the loudness also increases, but since loudness is a sensation, the loudness cannot, strictly speaking, be measured. No meaning can be given to the statement that one sound is twice as loud as another. Nevertheless it is easily recognized that while the loudness increases with increasing intensity, a given increase in intensity by no means produces equivalent effects on the ear when starting from different levels of intensity. This has led to the construction of a scale of loudness on an "ear scale" which must be understood to obtain the proper perspective on the problem of silencing the airplane.

A certain minimum intensity is required before any sound can be heard. While the magnitude varies considerably from ear to ear and depends to some extent on the frequency, acoustical engineers have determined an average value applicable to the average ear. When the intensity approaches a value in the neighborhood of 100,000,000,000 times the threshold value, the sensation becomes exceedingly painful, and the ear begins to feel the sound rather than hear it. Thus to reduce the most intense sound to inaudibility would require the reduction of the intensity in this ratio.

For intensities between the limits given, the ear is not equally sensitive. We may imagine an experiment in which two sources of sound A and B of the same character are connected alternately to the ear, and the intensities so adjusted that the two seem equally loud to the ear. Then suppose the intensity of A is increased until the ear can just detect that it sounds louder than B. By measuring the change in the intensity, the sensitivity of the ear can be determined. To a first and sufficiently good approximation for most purposes it is found that the intensity must be increased by a certain percentage, approximately 29 per cent, no matter whether the original intensity is 1, 1000, or 1,000,000. The mathematician expresses this observation by the statement that the response of the ear is proportional to the logarithm of the intensity.

Suppose then that a start is made with the threshold intensity which is called 0 loudness and that the experiment progresses in steps such that the intensity of A is increased until a difference is heard, i.e., by a factor 1.26; B is now made equal to A, and the process is continued until the entire range of intensities is covered. The threshold loudness is called 0, the first step 1, the second 2, etc., and thus the ear or loudness scale of units known as sensa-

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[1] Publication approved by the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Chief, Aerodynamical Physics Section, Bureau of Standards, Department of Commerce.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] The exact definition of intensity is the amount of energy transmitted by the wave motion in unit time through a unit area in a plane normal to the direction of travel of the wave.

tion units is built up. It is found that approximately 110 units cover the scale, the exact number being dependent on the frequency. The correspondence between this ear scale, the physical intensity scale, and some familiar sounds is shown in Fig. 1.[4]

Some of the consequences of the facts which have been briefly reviewed are not very well known. Let us suppose that the intensity of the sound produced by the airplane engine is equal to the intensity of the sound produced by all other sources including the propeller, vibration of wires, etc. The complete elimination of the engine noise would reduce the physical intensity in the ratio of 2 to 1, but so far as the ear is concerned the decrease is only 3 sensation units. Unfortunate, but true. The perfect engine muffler under these conditions would improve conditions very little. Of course, the assumption as to the relative intensities may be greatly in error, and the fact has been neglected that cer-

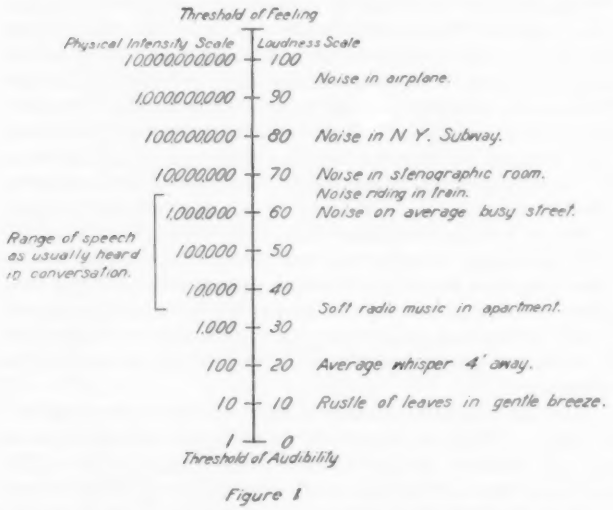


FIG. 1 PHYSICAL INTENSITY SCALE COMPARED WITH EAR SCALE

tain frequencies may produce more uncomfortable sensations than other frequencies.

The second consequence is that if the sound is once produced, the use of absorption as a remedy means that all but a small fraction of the energy must be eliminated. To reduce the sound by 20 sensation units means that 99 per cent of the energy must be eliminated, 30 sensation units 99.9 per cent, 40 sensation units 99.99 per cent, etc. The moral is that the sound should be eliminated at the source, if possible.

The source of sound has been stated to be a vibrating body. The vibrating body may be a column of air confined within more or less rigid walls or even unconfined air flowing in such a manner as to produce pressure changes of appreciable magnitude. The whistle of a bullet traveling at or above the speed of sound is produced by pressure changes in the turbulent flow of the air closing in behind the bullet. In this case the control of the sound is intimately related to the development of forms giving smooth air flow, a point that will be referred to later. The noise produced by a source of this type is usually not very intense unless the air speeds approach or exceed the speed of sound. The sources of sound of this type on an airplane are in the flow around the tips of propellers and in the flow of the exhaust gases from the engine. The only known method which is effective in dealing with sources of this type is to reduce the speed of the flowing air to well below the speed of sound.

[4] This figure is in part taken from Waterfall, *Engineering News-Record*, p. 60, Jan. 10, 1929.

In many instances, pressure waves which in themselves are of small amplitude produce much greater effects through the phenomenon of resonance. Any thin diaphragm such as the engine cowling, the fabric or plywood covering of a fuselage, any air column, as for example the air intake pipe to the carbureter, or any stretched wire has a large number of natural periods of vibration which in general are not greatly damped. Any small pressure fluctuation tends to set up vibrations in these resonating bodies which result finally in building up the intensity to a large value. This procedure may be illustrated by an ordinary tuning fork and resonator. It will be found that it is not necessary that the resonator be tuned to the fork frequency to produce a large effect. The flat top of an ordinary desk serves equally well. Attack on the source is most effective when the trouble is due to resonators, for with patience some of these may be discovered and removed or provided with sufficient damping.

PROPELLER NOISE

For several years the Bureau of Standards in cooperation with the National Advisory Committee for Aeronautics has been engaged in a study of the air flow about airfoil sections such as are used for propeller design. These experiments have been conducted at speeds from one-half the speed of sound to about 1.08 times the speed of sound. At some speed in the region from 0.7 to 1 times the speed of sound depending on the thickness, the flow about the airfoil begins to change from the smooth flow characteristic of good efficiency to the burbling type of flow which ordinarily occurs only at large angles of attack. The air no longer follows the upper curved surface nearly to the trailing edge but breaks away from the surface to form a region of violently eddying flow. In the model tests the change in flow is accompanied by a great increase in the noise. Under the conditions of the set-up, the thin airfoils spanning the stream serve as resonators which amplify the sound considerably. In this case noise is a direct evidence of inefficiency, for the lift is greatly decreased and the drag greatly increased when this type of flow begins.

The onset of the burbling flow depends primarily on the thickness of the section as compared to its chord length. The greater the thickness ratio, the lower the speed at which the inefficient flow begins. The shape of the section seems to be a far less important factor, though not without appreciable effect.

The bearing of these facts on propeller noise is somewhat as follows: For a given propeller the noise will increase as the tip speed increases. This is perhaps an obvious fact, but it is to be observed further than when the tip speed approaches the speed of sound the noise will increase at a rapid rate to an unbearable din. So far as any given propeller is concerned, it is advantageous to keep the tip speed as low as practicable from the standpoint of noise as well as of the efficiency of the propeller. Gearing may be said to be desirable, provided the gears are efficient and comparatively silent.

For propellers of varying thickness it may be expected[5] that at a given tip speed the thicker propeller will give greater noise, at least if the propellers are constructed of the same material. The blade of the propeller in most instances serves as a resonator, and the material of which the propeller is made together with the thickness determines the resonant frequencies and the magnitude of the damping. It is commonly believed that metal propellers make more noise than wooden propellers, but the question may be raised as to whether the metal propellers are or are not generally operated at higher tip speeds than wooden propellers. We know of no comparisons in which two propellers of identical design but of different materials are operated at the same tip speed.

[5] In the absence of measurements of propeller noise, the following statements must be in the form of speculations.

It is in this unsatisfactory state that the question of propeller noise must be left with the only conclusion that the tip speed should be kept as low as possible. It is interesting to speculate as to the part played by the vibration of the propeller blades and to wonder whether a blade so constructed that these vibrations were heavily damped would give considerably less noise.

Engine Noise and Muffling

It has always been a debatable question as to whether an engine muffler absorbs sound energy produced by the explosions in the engine cylinders or whether the muffler acts by modifying the flow of the exhaust gases so that the pressure changes set up are much less intense; in other words, as to whether the muffler takes a sound already existing and dissipates its energy or modifies the source so that a sound of lower intensity is produced. Some experiments have been made at the Bureau of Standards to obtain information on this point. A noise was set up in an air stream passing through a siren, the intensity being comparable with the noise of an airplane engine somewhat throttled. Mufflers of the several common types were introduced, but no appreciable change could be made in the noise unless a muffler of high resistance was used, and even then the reduction in loudness was small. While the apparatus available did not permit the simultaneous matching of air volume, frequency of impulses, and air pressure with the volume of exhaust gases, frequency, and exhaust pressure of an airplane engine, the results indicate that the muffler of an airplane engine functions primarily by so regulating the outward flow of exhaust gases that less sound is produced.

There are two well-recognized and effective principles on which mufflers may be operated, and a third principle is sometimes suggested but which is less effective than the other two. The first is the brute-force method of inserting resistance in the exhaust line so that pressure pulsations and hence the noise are reduced. Most mufflers do function somewhat in this manner, but the method is objectionable since resistance means a back pressure on the engine and hence reduced power. Practically all automobile mufflers are of this type. They are all heavy and impracticable for airplane use.

Most of the successful airplane mufflers, if any may be regarded as successful, operate on the second principle—namely, by reducing the speed of discharge of the exhaust gases. It has already been mentioned that this method is the most successful one for dealing with sources in an air stream traveling near the speed of sound. In practice the muffler utilizes some form of expansion chamber which serves to damp pressure fluctuations to some extent, but more important to cool rapidly the exhaust gases. In some designs the cooling is facilitated by providing devices to introduce a considerable amount of air which is mixed in thoroughly with the exhaust gases. It may be readily seen that the cooling of the exhaust gases produces an increase in density, and hence a given quantity (mass) of exhaust gas is expelled at a lower velocity. If sufficient cooling can be obtained, the velocity may be reduced by a factor of 2 or more. Since the speed of sound in the gas is also reduced in proportion to the square root of the absolute temperature, the ratio of the speed to the speed of sound is reduced in the ratio $\sqrt{2}$.

The third principle often advocated is to apply the methods of interference. In the laboratory, with pure tones it is sometimes possible to divide the sound path and produce at some specified location a low level of intensity. A fact often forgotten is that somewhere the intensity is twice as great. Furthermore a consideration of the relation between loudness and intensity shows that the intensity at the minimum must be of the order of one one-thousandth of the intensity at the maximum, a task which demands an almost perfect match of the amplitudes of the interfering tones. With complex sounds, the method of interference gives extremely unsatisfactory results in the laboratory. If the first thesis is granted—namely, that the larger part of the sound is set up in the air stream discharging from the end of the exhaust pipe—the use of branched pipes could not affect the major part of the sound. The only part of the sound that could be affected is the large and relatively low-frequency pressure changes due to the individual explosions. The adjustment depends on the temperature of the gases, since the speed of sound depends on the temperature, and can be correct for only one engine speed. Attempts to use the principle of interference on the note from a siren met with failure. There appears to be no successful application of this principle to engine muffling, and there are many reasons for believing that success is not likely to be encountered along this path.

Certain auxiliary matters in connection with engine muffling deserve mention. In the first place there are many sources of sound in the engine besides the exhaust. The gears, cams, valves, and other moving parts set up considerable vibration of metal parts leading to diverse kinds of noises. If care is not taken, the engine cowling and even the muffler itself, if made of thin unsupported sheet metal, may be set in vibration, leading to still further noise. In the second place, it is possible in certain types of airplanes to locate the exhaust above the upper wing so that a large mass is interposed in the direct line between the exhaust and the passenger cabin. Such a location leads to a very considerable reduction in noise.

Sound-Proofing the Cabin

The last problem to be considered is that of the sound-proofing of the airplane cabin, in which field Messrs. Chrisler and Snyder of the Bureau of Standards have carried out an extensive series of measurements. It must be recognized at the start that the problem is not alone one of sound transmission through the cabin walls, but also one of sound absorption. No matter how low the transmission of sound through the walls, the sound within the cabin will eventually build up to the same intensity as the sound outside if there is no absorption within the cabin. The low transmission simply delays the attainment of the final intensity for a short time after the starting of the engine.

Consider an airplane cabin of exposed surface area A_w placed in a field of sound energy such that the energy striking the surface per unit area per unit time is E_o. When equilibrium has been established, suppose that the energy striking any surface in the interior of the cabin per unit area per unit time is E. The condition for equilibrium is that the amount of energy entering the cabin per unit time by transmission through the walls equals the amount leaving the cabin by transmission through the walls added to the amount absorbed within the cabin. Obviously, if there is no absorption, the intensity within the cabin must equal the intensity on the outside of the cabin. If the transmission of the wall is t, the amount of energy entering per unit time is tE_oA_w. Similarly the amount leaving per unit time by outward transmission through the walls is tEA_w. If the average absorption coefficient of the exposed surface within the cabin is a, and the area of the exposed surface is A_a, the amount of energy absorbed per unit time is aEA_a. Therefore when equilibrium is reached, $tE_oA_w = tEA_w + aEA_a$

or

$$E = \frac{E_o}{1 + \frac{A_a a}{A_w t}}$$

The preceding elementary exposition, which is far from rigorous and contains several assumptions, will serve to give the general relationships. While without absorption there is no reduction of energy within the cabin, no matter how small the trans-

mission, absorption alone can accomplish little. Thus if the absorbing area equals the area of the walls, and if $a = 1$, so that all energy striking the absorbing surface is absorbed, and if $t = 1$ so that the walls are transparent, the reduction in loudness is only three sensation units. In any practical case some absorption is always present from the passengers themselves if from no other source. Each passenger has an absorption equivalent to a little less than 5 sq. ft. of surface having 100 per cent absorption. Because absorption is always present, a reduction in the transmission can accomplish a great deal. Fig. 2 shows the variation of $\frac{E}{E_o}$ with $\frac{aA_a}{A_w}$ for several values of t. The advantage of low transmission when the absorption is reasonably high is evident.

When the transmission of single sheets of material is investigated, it is found that the intensity on the physical scale of the

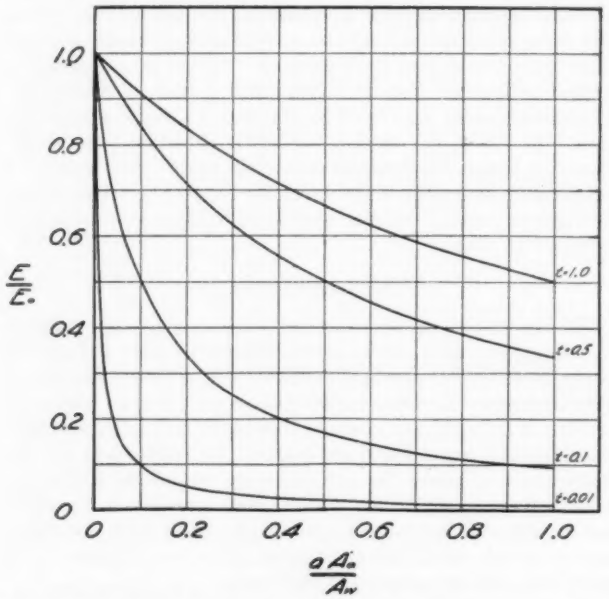


Fig. 2 Graphs to Show Advantage of Low Transmission When Absorption Is Reasonably High

sound received on the side opposite the source is approximately proportional to the reciprocal of the 0.73 power of the weight per unit area—in other words, to the reciprocal of the product of the density of the material and the thickness raised to 0.73 power. Because of the relation of the ear scale to the physical scale, the reduction of loudness in sensation units plotted against the logarithm of the weight per square foot gives a straight line. To obtain a reduction of 10 sensation units requires a material weighing 0.1 lb. per sq. ft., 20 sensation units 0.5 lb. per sq. ft., 30 sensation units about 3 lb. per sq. ft. It may be recalled that a material giving a reduction of 30 sensation units transmits only 0.001 of the energy falling on it.

The relation between the transmission and the weight was found by Messrs. Chrisler and Snyder to hold for wrapping paper, airplane fabric, glass, plywood, aluminum, and building board. The prospect of securing a satisfactory reduction of loudness by simple sheets of a single material is not very bright, and attention was turned to composite structures. In the sound insulation of buildings, air spaces prove to be of great value so that one of the first attempts was to use two sheets with an air space between. With light materials, as for example sheets of aluminum, the transmission through two sheets separated by an air space was greater than through a single sheet. The use of air spaces is entirely ineffective in the sound insulation of airplane cabins.

The next trial was the use of absorbent material between the two panels, a process that is not very effective in building construction. For the much lighter panels which must be used on airplanes, actual test showed that the use of absorbent material between the panels was very effective. With balsam wool, cotton, or hair felt as the filling materials, composite panels were built which gave a reduction as great as for a homogeneous material of twice the weight or more. With a blanket made from kapok fibers, such as are used in life preservers, a panel was constructed which gave a reduction as great as for a homogeneous material of three times the weight. This sound absorbent weighs only 1.14 lb. per cu. ft. A reduction of 30 or 35 sensation units is obtained with a wall weighing about 1 lb. per sq. ft.

The area of the walls of the cabin of modern commercial airplanes is usually from 30 to 50 sq. ft. per passenger. The casual reader might then suppose that 30 to 50 lb. per passenger additional weight would be required to provide the sound protection for each passenger weighing, say, 150 lb., and thus that about one-fifth the useful passenger load must be devoted to sound insulation. This is far from the case. In the first place the cabin must have some kind of walls in any case. If they are of airplane fabric, they will weigh only 0.1 lb. per sq. ft.; if of plywood, about 0.5 lb. per sq. ft.; if of aluminum, about 0.3 lb. per sq. ft. In the second place a modern cabin has some type of interior finish or lining for appearance sake. The weight of the wall that would be provided in any case without thought of sound insulation should be subtracted from the 1 lb. per sq. ft. of the sound-insulating constructing. Experience has shown that the additional weight is likely to be of the order of only 10 to 15 lb. per passenger—say, about 0.3 lb. per sq. ft. At this price the noise in the cabin may be reduced to about that in a railway coach in motion, so that conversation may be carried on with ease.

This concludes the brief survey of the present situation with regard to silencing the airplane. There appears no prospect that any single device will be developed which at the cost of a pound or two of additional weight will eliminate the noise. The imagination of the inventor is to be envied who proposed to have the sound actuate a loud speaker in such a way as to produce a new noise of equal amplitude to the original, but of opposite phase so that by interference no sound at all would be heard, but there appears no hope of progress along this line. By the same token it is unnecessary to resort to ear plugs, telephone sets, or speaking tubes. At the present time a great deal can be done by careful attention to detail to provide greater comfort to the passenger, and at a moderate increase in weight. It will prove more profitable to carry 8 passengers in comfort than 9 or 10 in conditions almost unbearable.

Discussion

S. A. Reed.[6] It is fortunate that this subject is being investigated by the Bureau of Standards and especially by Dr. Dryden, whose good work is recalled in research for the N.A.C.A. on airfoils. Mr. Fokker recently said that the noise from the engine exhaust can be eliminated easily to a great degree, so there is little doubt that propeller noise is the main factor. Dr. Dryden of course did not mean to say that vibration of some object is the only source of sound, which would mean that if a propeller is noisy its blades must be in violent vibration. In a siren there is no vibration, but merely a series of impulses in rapid succession. Other experimental sources of sound also exist without vibration.

In 1922 the N.A.C.A., under "Reports and Memoranda," published an account by the writer of experiments with models leading up to the development of a propeller, and it mentioned

[6] Reed Propeller Co., New York, N. Y.

that a control exactly like the propeller model was usually run, but with zero pitch; also that the writer was working at a range of tip speeds from about 800 up to 1500 ft. per sec. The sound with blades at zero pitch was of great intensity, with a perfectly pure musical note of absolutely definite pitch, and this musical pitch depended solely upon the number of blades and the rotation per second.

In 1925 A. Fage read a paper before the Royal Society in England, in which he said the same thing, viz.: that he found that the acoustic frequency corresponding to the pitch of the sound from a propeller is the number of blades multiplied by the rotations per second. The writer found that in the sound one might detect octaves of the fundamental note, but with blades of zero pitch other overtones appeared not to be present. With a propeller having pitch angles, with thin tips, there may be tones of minor intensity due to vibration of the blade. If there is flutter, of course there may be much confused noise, but fluttering propellers are usually rejected and do not go into service. Thus it may be concluded that the main sound from a propeller is due not to any fault which can be corrected, but is inherent.. As soon as tip speeds exceed about 900 ft. per sec. the intensity increases sharply; this fact is recalled in connection with racing propellers, where tip speeds are usually not less than 1000 ft. per sec.

That the propeller sound is not due to blade vibration to any considerable degree also seems quite clearly proved by the fact that the sound does not correspond in pitch to the estimated frequency of blade vibration. Blade vibration will be either free or forced. The free will be of constant frequency for all speeds, and dependent only upon the blade dimensions and material. The forced, due to thrust, may change somewhat with the speed, but certainly will not follow it accurately, as Fage and the writer have found.

It is not easy to calculate in advance the natural period of vibration of a propeller blade. The usual form of the blades of metal propellers has a rather steady taper in width and thickness from a stout root to a thin tip, and this form is unfavorable to free vibration, due to damping.

A recent paper by Liebers in Germany endeavors to calculate the vibration period of a thin metal propeller blade, and a conclusion similar to the foregoing appears to have been reached. Liebers' experiments were made with relation to the danger of blade fracture, and not to the question of noise.

In the writer's experiments referred to, the tachometer was invariably checked by noting and identifying the musical pitch, and then referring to a table of tone frequencies. A small mouth organ was used marked with the notes of the musical scale, and any one with a fairly correct ear can do this, using any musical instrument, marked with a scale. This was done hundreds of times and never failed to find exact agreement with the tachometer, and in later experiments in an open field with 90 hp. and models of 4-ft. diameter the writer would identify the note, and then predict to an assistant, who was observing the tachometer, exactly what he would find.

In the writer's paper for the N.A.C.A., reference is made to the classic experiments of Mach and Boys, who obtained refraction pictures on a photographic plate showing a longitudinal section of the conical waves spreading from the tip and tail of a rifle bullet, compression at the tip and dilatation at the tail.

These pictures are also now made by U. S. Army engineers at Frankford arsenal, and are very suggestive of conditions which may exist at the sharp leading and trailing edges of thin-blade metal propellers near the tips, and it should be noted that such would occur at zero pitch as well as with a thrust pitch, and that the velocity of propagation through the air would be entirely independent of slipstream velocity.

Some Characteristics of Light Alloys for Aircraft

By H. W. GILLETT,[1] COLUMBUS, OHIO

In this paper the author tells of the various light alloys that might be used to advantage in aircraft construction, and gives their advantages and drawbacks. As fast as the disadvantages are overcome, whether they relate to price fabricability, or other properties, the improved material will find increased use in aircraft. Progress in aeronautics is linked closely with the metallurgy of light alloys.

ACCORDING to the opinions of representative designers,[2] the use of metal in aircraft construction is on the increase. Wood and fabric wings still predominate on the smaller single-engined craft, especially on those makes not in large production, while in large transport planes, wing construction is almost exclusively metal. The wings are about the only major non-metallic part left on most planes, and they show a marked trend toward metal.

Apparently, the time will come before long when if aircraft, both airplanes and dirigibles, were drained of fuel and oil, piled up in one big junk heap, and set fire to, the whole mass would hardly get warm, so little wood and fabric would be present. One could pick out some rubber, glass, upholstery, etc., from the pile, but much the greater part of the weight would be metal. About half of the metal would be light alloys. By volume, the light alloys would greatly preponderate.

Since light alloys bulk so large in aircraft, it is logical that they should be considered in any comprehensive symposium on aircraft, especially one that takes the engineering angle. They have been so considered, time and again, and there are so many recent papers and articles[3] dealing with light alloys for aircraft, that there is nothing especially new or striking to report, nor can one bring to bear any fresh point of view.

It is not necessary to repeat the tables of properties of the light alloys that are available in many publications. Instead the subject will be considered more from the qualitative point of view, using round figures and approximations, when any figures are required. First, what are the light metals that serve as bases for light alloys for aircraft?

The light metals comprise aluminum, magnesium, beryllium, silicon, alkaline-earth metals like calcium, and the alkali metals like sodium, and lithium. The alkali and alkaline-earth metals are so readily attacked by moisture that they do not serve as materials of construction. Tiny amounts of them may find use as alloying elements, but as the base for structural alloys, they are quite out of the picture. There is no present hope for rendering them immune to corrosive attack.

Silicon is brittle, and alloys high enough in silicon to class as light alloys are brittle too. If and when high-silicon alloys can be produced that will be tough and workable instead of brittle and structurally useless, their corrosion resistance would be a factor in their favor. Many metals once considered to be inherently brittle and useless as structural materials, have yielded to the attack of the metallurgist, and become tough under proper treatment, tungsten being an outstanding example. Silicon is relatively cheap and silica is abundant. Some day silicon alloys may be in the picture, but that day will probably not be soon.

Beryllium

Beryllium is an unknown, rather than a rare metal. There is much beryl in the world, and when there is sufficient demand for beryllium so that it is prospected for, there will probably be plenty available for aircraft uses. So far, the metal has been produced on a laboratory scale only. It has been made, and its heavy alloys (those low in beryllium) studied quite exhaustively by Siemans and Halske.[4]

In this country, Cooper made the metal years ago. Hopkins and Meyer,[5] and Vivian,[6] have described the production of the metal in the laboratory. It is alleged[7] that it has also been produced in Austria.

Archer and Fink[8] studied aluminum-beryllium alloys containing only small amounts of beryllium and found them quite unpromising, as did Rawdon. The opportunity for beryllium alloys suitable for aircraft seems to lie in the beryllium-base alloys, i.e., those high in beryllium.

There has been very little done and less published in regard to light beryllium-base alloys for aircraft use, though some hints have been given.[9] Published German data have not dealt with light alloys at all. The Beryllium Development Corporation is alleged to have, in the last few months, put much effort on the making, casting, fabrication, and heat-treatment of light beryllium alloys. An experimental plane with alloys high in beryllium taking the place of those now made of duralumin and other alloys of aluminum, is said to be in contemplation. At the time of writing, however, the situation is as follows: While it is said that most of the difficulties intervening between the existence of raw ingot metal and the existence of useful alloys of that metal in fabricated form ready to be used in the construction of aircraft have been solved, and while it is alleged that many of the engineering data on the properties of the alloys, which must be in existence before aircraft design can make use of the alloys, have been collected, beryllium alloys for the making of aircraft parts are not on the open market, nor are their properties a matter of public record.

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[1] Director, Battelle Memorial Institute.

[2] "1930 Airplane Design" and "The Engine Builders Look at 1930," *Aviation*, vol. 28, Feb. 15, 1930, pages 331–348; Feb. 2, pages 386–390.

[3] See bibliography at end of the paper.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] "Beryllium Arbeiten," various authors, Wiss. Veröff, a.d. Siemens-Konzern, vol. 8, no. 1, 1929, 250 pages.

[5] "Electrolytic Production of Beryllium," B. S. Hopkins and A. W. Meyer, Trans. Am. Electrochem. Soc., vol. 45, 1924, p. 475.

[6] "Beryllium," A. C. Vivian, Trans. Faraday Soc., vol. 22, 1926, p. 211–225.

[7] See *Iron Age*, vol. 125, February 20, 1930, p. 597.

[8] "Aluminum-Beryllium Alloys," R. S. Archer and W. L. Fink, Proceedings, Inst. of Metals Div. Am. Inst. Min. & Met. Eng., 1928, p. 616–643. H. S. Rawdon, Discussion, p. 644.

[9] "Beryllium," M. Merlub-Sobel, *Metals & Alloys*, vol. 1, Aug., 1929, p. 69.

Beryllium alloys must therefore be classed as a potentiality not an actuality, from the standpoint of the aircraft engineer. They form a most interesting potentiality because of certain inherent advantages that will be touched upon later. But unless a surprising amount of success has crowned the efforts of the past few months, there will still be required a long period of patient research before beryllium alloys are available for aircraft construction. When they are, the price will be high in comparison to other light alloys. It is costly to purify the ore, costly to reduce the ore to metal, and the alloys will be costly to fabricate.

Magnesium Alloys

Magnesium alloys are an actuality. Yet they have been singularly slow in finding much use in aircraft. An "all-magnesium alloy" experimental plane was built years ago, and the methods of fabrication and the properties of the alloys have long been known. Their technology is well developed. Facilities for large-scale production of magnesium have long existed in this country, due to its use for flares, etc., in the war. Production could be expanded almost at a moment's notice if a real demand existed. The small demand has been disappointing to those who expected the magnesium industry to duplicate the growth of the aluminum industry. It is very doubtful if, outside of the war production for uses other than structural, the magnesium industry of this country at least, and probably of the world, has as yet made profit enough to take the whole enterprise out of the red.

The mechanical properties of magnesium alloys are quite satisfactory for aircraft use. Were there a large demand so that quantity production were in order, the price should be but little above that of aluminum. Magnesium "ores," whether solid minerals or brines, are plentiful, and while the cost of extraction of the pure metal, and of the fabrication of its alloys, will probably be a bit above that of aluminum, there is little prohibitive in the cost situation if and when real large-scale uses appear. The cost under relatively small production has been high enough so that when one merely considers the replacement of iron, steel, brass, or bronze by some light alloy, he normally chooses the cheaper aluminum alloys. Such a substitution accomplishes a big jump in weight reduction. The specific gravities are approximately: Steel 7.85, aluminum 2.7, magnesium 1.75, beryllium 1.85. The smaller jump from aluminum to magnesium is seldom considered worth the extra cost, when either group of alloys would serve. Pistons for automotive engines made of aluminum alloys do pretty nearly all that can be expected of magnesium-alloy pistons. Only when the last refinement in weight reduction is imperative, as in the rotors of a supercharger, has magnesium shown a great tendency to replace aluminum. There is likely to be more of a drift toward magnesium for such parts as are automatically protected from corrosion, such as engine parts that are constantly covered with oil.

That structural members, wing coverings, etc., are not ordinarily made from magnesium alloys is due to an inherent deficiency of magnesium and its alloys. They are prone to corrode. Considerable advance has been made toward, but not to, incorrodibility. The impure magnesium of not so many years ago, would corrode away in storage unless protected from atmospheric moisture. Present-day purified magnesium is far more resistant.

The older alloys containing copper as a strengthening element, were highly corrodible. These are now almost obsolete, and the chief alloys used today, one containing about 4 per cent aluminum and $^1/_4$ to $^1/_2$ per cent manganese, and others alloyed with zinc, are quite fairly corrosion-resistant in moist air. But few aircraft designers would dare use even these alloys around salt water or in salt air for parts whose structural strength must be maintained.

Chlorides are not healthy for magnesium; they promote corrosion very badly, and magnesium sheet, tubing, and other thin-section materials used in the presence of chlorides would require the use of an impervious, durable, protective coating. Such coatings are not yet available.

In an extensive investigation by the Bureau of Standards it was found that, in Washington, D. C. air, bare magnesium alloys of the more corrosion-resistant types were scarcely damaged after six month's exposure, but on the other hand, sheet specimens exposed at Coco Solo (Canal Zone), although protected by the most promising protective coatings now known, "resembled lace" after less than three months exposure, so severe was the corrosion.

So far, the best advice that can be given in respect to the use of sheet magnesium alloys is a slight modification of the old song, to read, "Hang your clothes on a hickory limb, but don't go near salt water." The Army Air Service confines it uses of magnesium to castings, since any army plane may have to be used near the sea coast.

On an inland mail or transport line, use of sheet magnesium alloys might be considered if there were a real protective coating. Hicks[10] says, "The ideal large plane is one in which the geometric extremities of the wings, fuselage, and control surfaces are of properly protected magnesium alloy, the intermediate and adjacent sections of the wings and fuselage of duralumin, and the middle portions of the fuselage and wing root of alloy steel." So far, no "proper protection" is known for conditions of service near salt water. Hence, the use of magnesium in aircraft will be pretty much limited to heavy sections, such as crank-case castings and experimental propeller forgings, or to such things as meter cases and small parts not requiring structural strength. Use of magnesium for such purposes is increasing, but hope for the satisfactory protection of sheet magnesium seems to be about abandoned.

Cost and Production

The price of ingot aluminum in 1892 was a little less than that of magnesium today. About 1900, it dropped to the present price level, and while it sold in 1909–1911 for about half its present price, for the last seven years it has stuck at 23–28 cents per lb., now being around the lower end of that range. The price of ingot magnesium for the last 10 years has run from 70 cents to $1 per lb.

The price of what little beryllium has been produced, has been around $200 per lb. When the time comes that one can buy beryllium at all, it is expected to start in at say $50 per lb., and then fall. Optimists look for a $5 price some day, and superoptimists for a price of the order of the present price of magnesium.

The probable world production for the present year appears to be somewhere about 250,000 tons of aluminum, 2500 tons magnesium, and 2.5 tons of beryllium. Knowledge of the alloys of aluminum, magnesium, or beryllium is in inverse proportion to their present price, and in direct proportion to the present tonnage.

Only one designer,[11] among those who replied to the questionnaire sent out by *Aviation*, mentions beryllium at all in discussing materials for aviation engines. Rosenhain,[12] however, takes an optimistic attitude in regard to the probable utility of beryllium in aircraft.

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[10] See *Aviation*, vol. 28, Feb. 15, 1930, p. 336.

[11] *Ibid.*, p. 345.

[12] W. Rosenhain (abstract of lecture by), "An Important New Metal for Aircraft," *Iron Age*, vol. 125, Feb. 27, 1930, p. 692.

Alloy Steels

Strong, heat-treated alloy steels, and the strong light alloys compete with each other for aircraft construction. Alloy steels heat-treated to around 200,000 lb. per sq. in. have a somewhat better strength-weight factor than the wrought strong light alloys, though the strongest magnesium alloys are about equal. The alloy steels heat-treated to 125,000 lb. per sq. in. have a somewhat poorer strength-weight factor.

On a yield-point-weight basis, the steels are superior because the yield point is much nearer the ultimate. Although the yield point in both the light alloys and in heat-treated steels is more or less a matter of definition, on almost any basis of allowable permanent set the steels have the edge. Stream-line wire, control cables, etc., are therefore never made of light alloys.

From the point of view of the modulus of elasticity, the steels are three times stiffer than the aluminum alloys, and five times stiffer than those of magnesium, while an alloy high in beryllium will have almost the modulus of steel. One does not make springs of the light alloys, but a beryllium spring could be made. On the other hand, the greater bulk of the light alloys gives a tube or an I-beam of a given strength a greater stiffness as a structure when made from light alloys than from steel. Even when one draws a steel tube so thin that it has the same weight as a given light-alloy tube, it would not have the same resistance to crumpling, though it might have identical tensile strength.

The exigencies of manufacture require certain minimum dimensions. A casting can be made only so thin. A boss must have a certain thickness to have enough thread area for a cap screw to hold, and so on. The new rigid airship in which the metal skin is the gas container, and transmits some of the stresses, uses aluminum-alloy sheet 10 mils thick. This serves the purposes better, and is easier to fabricate than would be the case with the excessively thin stainless-steel foil that would give the same strength-weight ratio.

When the minimum dimensions for fabrication give sufficient strength in the light alloys, the light alloys replace the heavy ones. Hence, there will be mighty few steel castings in aircraft, and it is in castings that magnesium will give the aluminum alloys something of a battle. When the minimum dimensions allow the use of either, other factors decide. The new British dirigibles use either duralumin or stainless steel in much of their frame work, and a final choice has yet to be made. In the 12-engined, 170-passenger Dornier, the main spars are said to be of extruded magnesium alloy. That plane is described as chiefly made of aluminum alloys.

Metal Propellers

For aerodynamic efficiency, metal propellers need a more or less definite external form and size. If it were possible to make a hollow, heat-treated, steel propeller, it ought to be better than one of a light alloy. Its higher modulus would make it hold its shape better under load. Its endurance limit would be better since the endurance-limit-weight ratios of the steel could easily be made double that of the light alloy propeller, and the steel would have a real endurance limit while there is a question whether the light alloys do have a true endurance limit. Between the heat-treated 96 per cent Al, 4 per cent Cu forging alloy and the heat-treated 95½ per cent Mg, 4 per cent Al, ½ per cent Mn forging alloy suggested for propellers, there is apparently not much choice in actual endurance limit, but the endurance-limit-weight ratio is probably in favor of the magnesium alloy. Endurance data on the magnesium alloy are as yet none too conclusive.

But, as a matter of fact, whether the propeller is steel, aluminum alloy, or magnesium, it has to be, so far, a solid forging. Built-up propellers using a high-tensile steel framework inside, and a welded light-alloy covering outside, have been used, and some work is being done on welded, hollow-steel propellers. The hollow-type propeller has theoretical advantages, but one hesitates to put any sort of joints in a propeller. That being the case, the light alloy is the better bet since it will stand up, and the actual weight of the solid forging is less.

Effect of Welding and Heat Treatment

On the impact-weight basis, the heat-treated steels show up better than the light alloys unless the steels are used in a very hard condition. One of the drawbacks of the light alloys is that their optimum strength is obtained by heat-treatment, or by cold working, or both. Heating in welding ruins the strength. Since welding is so important in airplane fabrication today, steel has the edge over light alloys for parts to be welded.

The light-aluminum alloys respond to heat-treatment much more readily in the wrought state, with the grains smashed up, than they do in the cast condition. Weld metal is, of course, in the cast condition. Castings must be heated many hours in order to effect solution for "precipitation hardening," while in wrought metal, far less time is required. The light alloys do not undergo a grain refinement on cooling as is the case when steel cools through the A_1 point. With carbon steel or stainless steel, while the weld may not be of just the composition desired, metal can be built up at the joint so that there is enough strength there. The weak spot is a bit away from the weld where the tubing itself is softened. Reheat-treatment of the whole welded assembly can repair this damage rather better than in the case of the aluminum alloys.

Chromium-molybdenum steel tubing has sufficient air-hardening properties so that air-cooling after welding regenerates a good grain structure, and the welded structure will be 80 to 90 per cent as good as the original heat-treated tubing. This great advantage accounts for the almost universal use of chromium-molybdenum steel tubing for fuselages.

Dirigible girders are built up in light aluminum alloys by bending and punching sheets into channels, and assembling these and the stamped cross members into box girders by riveting, not by welding. Airplane-wing spars are made either in steel or light aluminum alloys with some tendency toward a construction using both materials.[13]

One point in favor of modern duralumin is its uniformity of properties in different thicknesses. Tubes of different wall thicknesses, for example, appear to run more uniform in properties than would be expected in steel.

While magnesium alloys can be welded, it is a tricky operation, and welded-magnesium aircraft structures would be difficult to build up. Moreover, magnesium-alloy tubing is hard to make, and structural use of the magnesium alloys would have to rely chiefly on extruded shapes. Nothing is known about the weldability of beryllium alloys, nor of their ability to be made into tubing or extruded shapes.

Corrosion

The difficulties and disabilities of magnesium and its alloys in resisting corrosion where chlorides are present were mentioned above. Duralumin, the mainstay among the wrought-aluminum alloys for aircraft use, is also attacked in a fashion that is dangerous because the attack is not necessarily indicated by the surface appearance. This attack is intercrystalline in nature, and the alloy gets brittle. The attack may be greatly reduced by proper heat-treatment, i.e., cold-water quenching and the avoidance of accelerated aging at an elevated temperature. The attack is of no structural import on heavy sections like propellers. The thinner the material, the worse the damage.

[13] See footnote 10.

Thin sheet, say 10 mils, in contact only with rain water or distilled water, and with rather pure air, in locations far from the sea shore, and not badly contaminated by industrial gases, loses its ductility at an alarming rate. This trouble has been experimentally overcome through the application of thin layers of unalloyed aluminum by the metal-spray process, and practically, at least for fairly thick sheet, by casting a duralumin ingot between two unalloyed aluminum sheets to which the duralumin welds, and rolling the composite ingot. The resultant "Alclad" duralumin sheet is very resistant to intercrystalline embrittlement, and such coatings are far and away ahead of any non-metallic coating. Unfortunately, Alclad tubing has not yet become available, and outside of the metal-spray process, there seems little hope of putting an aluminum coating on magnesium alloys which so badly need protection.

The endurance limit of Alclad sheet duralumin is lower than that of ordinary duralumin, but no worse than that of duralumin that has corroded somewhat. The ductility and resistance to impact of the Alclad duralumin is far superior to that of the uncoated, somewhat corroded material. It has a slightly lower strength-weight ratio than uncorroded duralumin, but a far better toughness-weight ratio than somewhat corroded duralumin. The Alclad material starts out with certain properties and maintains them. This is one of the many cases where sustained mediocrity is preferable to shortlived perfection.

Protective Coatings

Alclad duralumin is rapidly displacing bare duralumin for wing coverings and the like, and is sometimes satisfactorily used without any sort of paint or varnish coating, though in tropical climates even Alclad is the better for some protection, and the thinner it is, the more it needs protection. Even though the thinnest Alclad sheet may not have a sufficient thickness of pure aluminum on the surface to give as much protection as is had in thicker sheets, its substitution for uncoated duralumin in the new metal-clad airship *ZMC-2* has probably materially increased the life of the ship.

That aluminum or the aluminum-manganese alloy used in making hard sheet is not materially attacked by chlorides is evident from the performance of the cooking utensils in every kitchen. But intermittent immersion in seawater is pretty severe service, and even Alclad sheet, although probably the best light material available for seaplane floats, has a limited life in such service.

The duralumin parts that cannot be given an aluminum skin, and any parts made of magnesium alloys have to be given some sort of a protective coat. To make the metal surface of such texture that coatings will stick, it is advisable to produce a matt surface. Anodic treatment in chromic acid (and recently promising use is being made of sulphuric acid) for aluminum alloys, some analogous chemical treatments without the use of electrolysis, in carbonate-chromate mixtures, or in milk of lime, and a chromate or phosphate treatment for the alloys of magnesium, produce a tough thin skin over the surface, with a matt texture that serves well as a "priming" coat for other coatings. One aircraft maker characterizes the anodic coating as of "immense benefit as a base for the lacquer or varnish." The anodic or similar skin, despite claims for inherent corrosion resistance is in itself not of very great value in presence of chlorides, according to the author's opinion. Its chief value is as a primer.

The paints, varnishes, lacquers, and what-not suggested for protection of light alloys are simply legion. Their protective value is far from perfect, and, from some points of view, largely illusory. Greasing over one of the matt "primer" skins, with constant renewal of the grease, preferably with aluminum powder in the grease, the use of a flexible linseed-oil carbon-black coating, or, under some conditions, rubber-type coatings, especially if pigmented with aluminum powder, do help. Brittle paint films that check or become pervious to moisture only localize the damage and delay it a bit. Some synthetic resin lacquers and the zinc chromate primers offer some promise among the non-flexible coatings.

Exposure tests of coated magnesium alloys in the tropics, in which the metal surface has been phosphate-treated, and a wide range of protective coatings tried, should be illuminating to those who believe that paint-and-varnish-type coatings afford true protection. If these coatings could really remain impervious, they would really protect magnesium. They do not. The coating doesn't know whether it is on magnesium or on duralumin. It is obviously unsatisfactory on magnesium because magnesium itself corrodes so rapidly. That it is really unsatisfactory on duralumin is less apparent because duralumin corrodes more slowly. That the non-metallic coatings afford some protection and that even partial protection is desirable, cannot be denied, but a non-metallic protective coating in which implicit faith can be placed for the complete protection of duralumin is still a long way off.

In Navy practice, bitumastic coatings are used where practicable, otherwise, aluminized spar varnish. Anodic treatment is required in order to secure proper adhesion.

Metallic beryllium is very resistant to atmospheric attack. It, and the aluminum alloys high in beryllium are said[14] to be more resistant to chloride attack than any other light alloys. Results of some cursory tests which the author has seen, make him want very definite proof of this before accepting the statement, though he can well believe that they will be far ahead of the magnesium alloys.

Castings

For many years, the mainstay in the aluminum sand-casting business was the "No. 12" alloy, with 8 per cent copper, which is nearly as brittle as cast iron. The alloy still serves for many purposes, but as the castings became more complicated, there was a need for an alloy less prone to cracking in the mold. The alloys with silicon then came in, and combined an almost foolproof resistance to mold-cracking, and a good ductility in the casting at room temperatures. But they had a very low proportional limit. Non-heat-treated alloys with both copper and silicon strike a rather happy medium. For aircraft use, still greater strength combined with good ductility is afforded by an alloy with 4 per cent copper given a very long time heat-treatment, or by alloys of copper and silicon, also heat-treated. In these latter, the properties can also be varied by varying the iron content. The more complex alloys with copper, magnesium, and silicon as alloying elements, are also used as heat-treatable casting alloys. They give higher strength, but lower ductility than the alloy with copper alone. A still more complex alloy, the so-called "Y" alloy, to which nickel is added as well as the other elements, finds minor use in sand castings. Rolls Royce is reported[15] as using a "Y" type alloy with high silicon which may possibly offer some of the casting advantages of the silicon alloys plus an approach to the properties of the oth r heat-treatable casting alloys when heat-treated.

Magnesium is attacked by steam so its alloys are not cast in ordinary green sand. The sand is bonded with glycerine or sulphur is added so that some other atmosphere than steam comes in contact with the hot casting. Other difficulties are present, but all have been well overcome so that almost any-

[14] See footnote 9.

[15] Some Recently Introduced Aluminum Alloys, W. C. Devereux, *Metal Ind.* (London), vol. 36, Jan. 24, 1930, p. 117.

thing that can be cast in aluminum alloys can be cast in magnesium alloys if one is willing to pay the price. On a strength-weight basis, they are superior to the best aluminum alloy castings so their increased use in aircraft is probable.

The alloys high in beryllium are not easy to cast, although sound castings of simple shapes have been made. There are no data available on the properties of such castings. Permanent-mold castings, made in an iron mold under gravity flow, are much used for pistons. Here ductility and corrosion resistance (except to exhaust gases) are not as important as strength at high temperatures, and resistance to wear. The greater thermal coefficient of expansion of aluminum alloys over cast iron is a drawback, partially cured by Invar struts, but the higher thermal conductivity is an advantage. In automobiles, many manufacturers change from aluminum to cast iron from year to year. In aircraft, the aluminum piston remains standard.

On account of the low melting point of aluminum, most of its alloys get pretty weak at engine temperatures, so the alloys have to be chosen with high-temperature strength in view. The probability of still higher temperatures focuses attention on this point.

Although magnesium pistons will work, the strength of the magnesium alloys is still lower than that of the aluminum alloys, and their thermal conductivity is lower so that in order to drain off enough heat, the piston has to have greater volume. Unless magnesium becomes as cheap as aluminum, there seems little chance for the magnesium piston as there is no real weight reduction. Attack by exhaust gases would probably be less than with aluminum.

Beryllium is of great interest here. Its thermal coefficient of expansion is not far from that of cast iron. Its high melting point encourages one to expect good strength at high temperatures, and beryllium itself is so hard that good wear resistance may result. On the important matter of thermal conductivity there seems to be no data at all on beryllium, but one would hardly expect it to be as high as in aluminum or even magnesium. It will take some time to develop casting methods that will produce sound pistons of beryllium or alloys high in beryllium.

Die Castings

Pressure die castings in aluminum alloys are useful outside of aircraft for small parts not highly stressed. They can be produced close to size and require little machining. They are likely to consist of a honeycomb of blow-holes on the inside, and a nice sound skin on the outside. As long as the part is still strong enough even though its interior is cheesy, this merely helps on weight reduction, but inspection difficulties are multiplied to be sure that each piece is strong enough. Material with internal defects such as are likely to be found in die castings, is most dangerous under repeated stress, and hence, there is almost no use of die castings in stressed aircraft parts.

There is a great difference between a die casting made by one who knows how to make it, and one who does not. The efforts of the die-casting committee of the American Society for Testing Materials are helping this situation materially, but it is doubtful if the castings can be made satisfactory for highly stressed parts of aircraft. On the whole, for aircraft use one prefers a forging to any casting, both because of soundness and because the forgings heat-treat to better properties.

Forgings

Here the aluminum alloys stand out. The technique of making heavy aluminum forgings is pretty well understood. The magnesium alloys harden up more rapidly on forging and are therefore more difficult to make. Yet, satisfactory magnesium-alloy propellers have been made experimentally, and have decided advantages over heavier alloys. They may have a future.

Beryllium-alloy forgings are still a closed book. As the metal is only amenable to hot-working, development of such forgings would be even harder than with magnesium. The high modulus of elasticity might give a material advantage to a beryllium-alloy propeller. What the endurance properties would be is entirely unknown.

Heat-treatable light-aluminum wrought alloys have all been spoken of as a class. The chief alloy, duralumin or "17 S," contains around 4 per cent Cu, $^1/_2$ per cent Mg, and $^1/_2$ per cent Si. The mechanism of heat-treatment, according to the usual theory, is to heat to about 500 deg. cent. long enough for most of the previously separated constituents to be taken into solution, and to quench, retaining the solution at room temperature. As time goes on hard compounds like $CuAl_2$ and Mg_2Si form and precipitate at room temperature in very tiny particles which strengthen the alloy. The alloy can be severely worked immediately after quenching while it is soft without making it brittle. Rivets must be driven in about $^1/_2$ hr. after quenching.

Duralumin hardens at room temperature. If copper and silicon are the only alloying elements as in the alloy "25 S," or if only magnesium and silicon are used as in "51 S," then we get, respectively, $CuAl_2$ or Mg_2Si. Without the presence of the other, these compounds do not precipitate at room temperature. Heating is required. Hence where cold forming is to be done, it can be carried out at leisure with these alloys instead of in a hurry as with duralumin. The alloys are more readily handled than duralumin in some types of hot forming. Propellers are generally made from the 25 S (copper) alloy.

The strength of these alloys is just a little below that of duralumin. Hence, when the best strength-weight ratio is needed, duralumin is used unless some problem of fabrication outweighs the need for strength.

There is naturally a desire for material of still better strength-weight ratio than ordinary heat-treated duralumin. In dirigible girders, carefully heat-treated duralumin may be slightly cold worked after aging has occurred or progressed part way, thus adding something in tensile strength, considerable in tensile yield point, and considerable in endurance limit, at the expense of some ductility. There is not so much use of heat-treated and cold-worked duralumin in airplane tubing, since the yield point in compression is not raised, and probably lowered by the cold work.

Duralumin, higher in silicon than the normal, can by aging at a slightly elevated temperature be brought up to 75,000 lb. per sq. in. with 13 per cent elongation instead of the 60,000 lb. per sq. in., 25 per cent elongation, of very good duralumin of ordinary composition. This "super-duralumin" is strong enough so it has a better strength-weight ratio than the magnesium alloys of comparable ductility. There seems to be hope of making beryllium-aluminum alloys that will have a materially better strength-weight ratio than the alloys of the better known light metals. So many factors enter the selection of a material for a given purpose in aircraft construction that in a paper of this sort, one can deal only witeh gneralities.

Conclusion

An attempt has been made in this discussion to set forth some of the outstanding drawbacks of the light alloys as well as their outstanding advantages. Competition among light alloys and with other alloys will go on. Every time a drawback is overcome, whether it be of price, of fabricability or of properties, the improved material will find wider use in aircraft. The survival of the fittest materials will doubtless bring light alloys

even more to the fore in aircraft construction. Without them aircraft could not have reached their present usefulness. Aeronautics owes much to the science of aerodynamics and to engine designers, but it also owes much to metallurgy. Metallurgical research has provided materials with properties suitable for the service demanded. Further progress in aeronautics is closely linked with further progress in the metallurgy of the light alloys. Fortunately, so much continued metallurgical research in light alloys is in progress that one can confidently look forward to continued improvements.

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Some Economical Aspects of Airplane Design

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Airplane design requires a very clear idea of the duties to be performed. Specifications should include full information as to engine power, load, top speed, landing speed, climb, range of operation, restrictions of dimensions, special arrangements for docking or storing, and factors of safety desired. The paper contains computations and graphs for planes with one, two, or three engines. It shows gains in pay load with reduction of top speed and increase in power loading. The three-engined plane suffers in comparison due to greater resistance from the greater division of power.

DESIGN of modern airplanes introduces many problems of a purely economic nature. These are factors that have a profound influence on the performance of the airplane, but are not directly connected with its engineering aspects. For example, it is simple enough to design and build an airplane that will fly, but not so simple to design and build one to meet specific conditions.

For military service these conditions are so different that distinct types have been evolved for pursuit fighting, observation, photography, bombing, transport, etc. Hence the military airplane is designed for a single purpose, all conflicting requirements being compromised to meet that end best.

The commercial airplane on the other hand has not, up to the present at least, been required to meet such specific requirements, the same make and type of airplane often being used for passenger hopping, aerial taxi, commercial photography, and cross-country flying. This promiscuous use of any type of plane for any duty has been responsible for large operating costs and a more or less mediocre performance, the airplane being compromised in so many places to make it suitable for the many kinds of service.

The public attitude has contributed its bit to this haphazard procedure by hesitating to ride in certain types of planes, believing them to be less safe than some other form which may have received considerable publicity and extensive advertising. Thus the airline operator has in many cases been forced to buy equipment that he would otherwise not think of purchasing if it were not for the public sentiment involved. This has in many instances led to incorrect investment, high operating cost, and less efficient service.

While public interest was running high and almost any kind of an aeronautical enterprise could be financed, there was no particular need for the air operations companies to show a profit, but such days cannot last forever and sooner or later all users of aircraft must establish their operations on an economically sound foundation. Thus the present need of aviation, particularly the transportation companies, is to make money. To do this, they must realize the lack of suitability of much of their present equipment and purchase airplanes designed to meet their particular requirements. This leads to a study of the factors governing the design of suitable equipment.

The first important point, then, is that the designer should have a very clear conception of the duties which the airplane has to perform, and for this purpose a specification is absolutely essential. The authority for whom the airplane is to be constructed should draw up this specification from a careful study of his requirements. It should include such principal points as:

1 The engine power available for the design
2 The amount and nature of the load to be carried
3 The top speed which is required from the airplane, and more particularly, the cruising speed required
4 The landing speed required
5 The climb which is necessary
6 The range of the airplane
7 Restrictions as to overall dimensions
8 If multimotored, the performance with engines out of action
9 Any special arrangements for docking or storing
10 Factors of safety required
11 Accommodations, etc.
12 All other points that the purchaser feels need particular attention.

The specification should also include such points as type of finish, materials of construction, type of construction, and any special provisions required.

If the specification is sufficiently detailed a capable designer can study the project and decide whether he can design an airplane to meet the requirements or not. If he can, he should have no fear of drawing up a rigid contract to build to the specification or provide a forfeit. If he cannot, it is better that he abandon the project because the contractee presumably knows what he wants and is only delayed by further negotiations.

The operator may need a land plane, sea plane, amphibion, or flying boat. He may need single- or multiple-engined equipment. He may need to operate a fast schedule to make connections with other lines, or the range may be large because of the necessity for non-stop flights. Speed in the extreme may be unnecessary, but the transportation of large loads may be of paramount importance. There is no end to the cases that may possibly arise. It is, therefore, interesting to study some of the economical factors in airplane designs.

Passengers Versus Speed

A passenger is chosen as a unit of load because it is more comprehensible to the average mind. The average person has a much better idea of the load if one says 150 passengers than if one says 27,000 lb. Computations have been made, and curves drawn for installations with one, two, or three engines each of 200, 300, 400, or 500 hp.

Since each combination of engines was studied for seven different values of the speed, the results are obtained for 84 hypothetical airplanes differing only in scale, because they are assumed to be geometrically similar. That is, all the three-engined airplanes are geometrically similar, and all the single-engined planes are also, although there is, of course, no similarity

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

between the planes having a different number of engines. This difference is weighted by the use of the appropriate constant in the speed formula.

A wing loading of 15 lb. per sq. ft. of wing surface has been chosen as a basis for these calculations, for it is considered the maximum that may be used and still give a landing speed that present airports and emergency fields will permit. With this loading an airplane with a maximum lift coefficient of $C_l = 1.40$ will have a stalling speed of 65 m.p.h. at sea level when referred to standard conditions.

It is well known that the most economical speed of operation, namely, most miles-per-gallon speed, occurs for most conventional airplanes, at approximately half their maximum lift coefficient. Thus the most economical speed is about 41 per cent greater than the stalling speed. $V_e = V_s \sqrt[3]{2}$. While this is the most miles-per-gallon speed, it is not necessarily the best

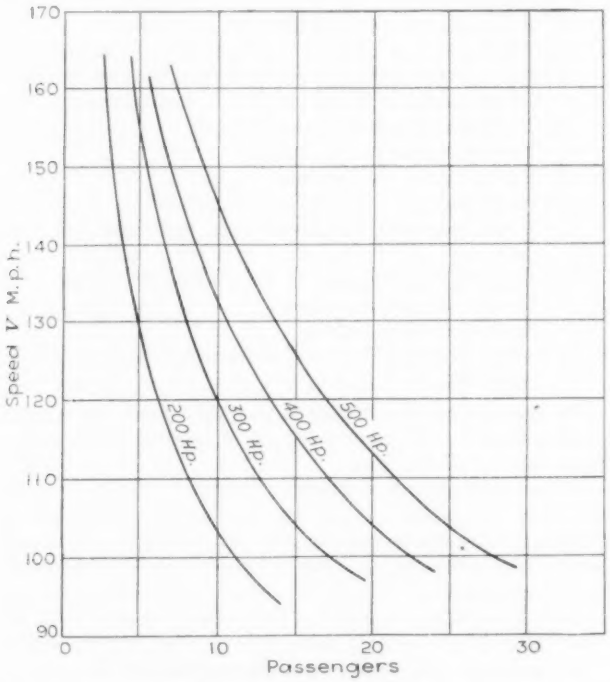


FIG. 1 RELATION OF PASSENGERS AND SPEED IN PLANES WITH SINGLE ENGINES

cruising speed for the airplane because there are a number of direct and indirect charges placed upon the airplane that have no direct connection with the plane except by the number of hours it is in the air. That these charges may be kept as low as possible, it is necessary for the airplane to complete the flight in the minimum time. Thus there are two conflicting factors which mutually affect one another and demand a compromise performance. An investigation of operating costs of a number of different enterprises seems to show that the best compromise is to operate the airplane at a speed approximately 25 per cent in excess of the economical speed. This then is the best cruising speed considering both direct and indirect expenses.

With a stalling speed of 65 m.p.h. the economical speed will be 92 m.p.h. and the best cruising speed 114 m.p.h. This speed is independent of the maximum speed or the amount of engine power installed. This is an important point which has not received the attention which it deserves. The reserve speed required of course depends upon average climatic conditions, terrain operated over, length of line, importance of schedule, etc.

Hypothetical airplane performance and characteristics were computed in the following way:

1 By assuming the top speed required

2 By solving for the area by the formula $V = k\sqrt{\frac{\text{hp.}}{A}}$

where V = maximum speed in m.p.h., hp. = horsepower of engines, A = wing area, and K = a constant. It is 140 for single-engined airplanes, 130 for twin-engined airplanes, and 125 for three-engined airplanes.

3 By solving for the gross weight by multiplying the area by the wing loading

4 By solving for the pay load which is assumed to be 25 per cent of the gross weight

5 By solving for the number of passengers, assuming a passenger and his baggage to weigh 180 lb.

The calculations are given in tabular form in Tables 1 to 12 inclusive. The results are plotted by groups in Figs. 1 to 3 inclusive. Fig. 1 is for single-engined planes, Fig. 2 for twin-engined planes, and Fig. 3 for three-engined planes.

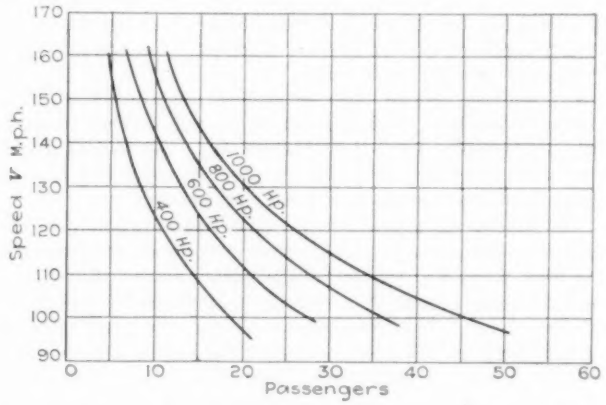


FIG. 2 RELATION OF PASSENGERS AND SPEED IN PLANES WITH TWIN ENGINES

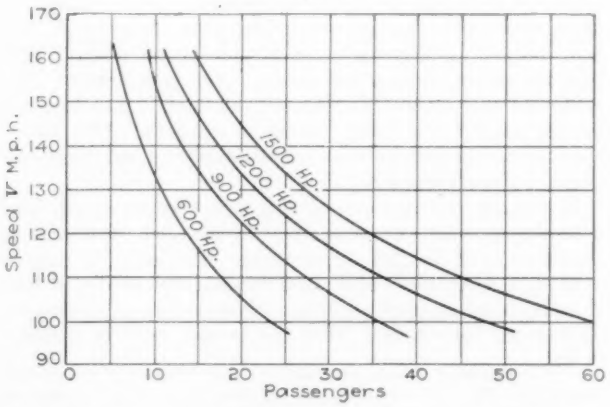


FIG. 3 RELATION OF PASSENGERS AND SPEED IN PLANES WITH TRIPLE ENGINES

As would be expected the results are very similar for the three different groups. They all show large gains in pay load with reduction in high speed and increase in power loading. The smaller pay load per horsepower for the multiengined planes is a direct consequence of the smaller constant in the speed formula. The three-engined plane suffers from a greater resistance because of the greater division of power. Thus the

speed constant for the three-engined plane is considerably less than that for a single-engined installation, and a little less than that for the two-engined plane. Most of this difference comes from the fact that engine nacelles are usually not well "streamlined," and even if artistically done are still of rather high resistance. Some of the newer cowling methods promise a large increase in the speed constant for both two- and three-engined planes.

In drawing conclusions from these curves and figures, it must be remembered that the results are no more reliable than the assumptions they are based on. It is true that some of the hypothetical airplanes analyzed are probably not capable of flight because of the extremely high power loadings. They have been included, however, because they border on the edge of practical design and fall in the class of some of the initially heavily loaded record-breaking airplanes. It should also be remembered that these results apply only to conventional airplanes of modern design, and do not indicate future possibilities. The constant reduction of resistance will gradually permit higher and higher speeds and greater pay loads. In brief, the reduction of resistance will be shown by an increase of the constant in the speed formula.

TABLE 1 AIRPLANE CHARACTERISTICS WITH ONE 200-HP. ENGINE

Hp.	V	A	Lb. per sq. ft.	W	Pay load	Passengers
200	100	550	15	8250	2062	11.5
200	110	414	15	6200	1300	7.2
200	120	318	15	4770	1192	6.6
200	130	250	15	3750	938	5.2
200	140	200	15	3000	750	4.2
200	150	162	15	2430	608	3.4
200	160	134	15	2010	502	2.8

TABLE 2 AIRPLANE CHARACTERISTICS WITH ONE 300-HP. ENGINE

Hp.	V	A	w	W	Pay load	Passengers
300	100	825	15	12400	3100	17.2
300	110	615	15	9200	2300	12.8
300	120	477	15	7150	1788	10.0
300	130	375	15	5120	1280	7.1
300	140	300	15	4500	1125	6.3
300	150	244	15	3660	915	5.1
300	160	201	15	3010	752	4.2

TABLE 3 AIRPLANE CHARACTERISTICS WITH ONE 400-HP. ENGINE

Hp.	V	A	w	W	Pay load	Passengers
400	100	1100	15	16500	4125	22.9
400	110	820	15	12300	3075	17.1
400	120	635	15	9500	2375	13.2
400	130	500	15	7500	1875	10.4
400	140	400	15	6000	1500	8.4
400	150	325	15	4880	1200	6.8
400	160	268	15	4020	1005	5.6

TABLE 4 AIRPLANE CHARACTERISTICS WITH ONE 500-HP. ENGINE

Hp.	V	A	w	W	Pay load	Passengers
500	100	1375	15	20600	5150	28.6
500	110	1025	15	15400	3850	21.4
500	120	795	15	11900	2975	16.5
500	130	625	15	9400	2350	13.1
500	140	500	15	7500	1875	10.4
500	150	407	15	6100	1525	8.5
500	160	336	15	5050	1262	7.0

TABLE 5 AIRPLANE CHARACTERISTICS WITH TWO 200-HP. ENGINES = 400 HP.

Hp.	V	A	w	W	Pay load	Passengers
400	100	880	15	13200	3300	18.3
400	110	662	15	9930	2480	13.8
400	120	508	15	7630	1910	10.6
400	130	400	15	6000	1500	8.4
400	140	320	15	4800	1200	6.7
400	150	260	15	3900	975	5.4
400	160	215	15	3220	805	4.5

TABLE 6 AIRPLANE CHARACTERISTICS WITH TWO 300-HP. ENGINES = 600 HP.

Hp.	V	A	w	W	Pay load	Passengers
600	100	1320	15	19800	4950	27.5
600	110	995	15	14900	3730	20.7
600	120	765	15	11500	2880	16.0
600	130	600	15	9000	2250	12.5
600	140	480	15	7200	1800	10.0
600	150	391	15	5870	1465	8.2
600	160	322	15	4825	1206	6.7

Wing Loading

From the foregoing discussion it would seem that there is no end to the benefits of increasing the wing loading. There are, however, serious drawbacks, if the loading is excessive for the size of the machine, for if the loading exceeds a particular value dependent on the size of the plane the airplane loses what might be called its more or less buoyant feeling when in a powerless glide. Another drawback is the resultant high landing speed, although this is not as serious as it seems on first thought. Briefly the advantages and disadvantages are as follows: Among the advantages are: (1) It permits the airplane to be operated nearer its economical speed, (2) it gives the pilot more control of the plane in landing, (3) the airplane is influenced less by air disturbances, (4) it reduces the structural weight, and (5) it reduces the resistance of the plane. The disadvantages include: (1) A slightly longer take-off run, (2) a longer landing run, and (3) a greater amount of kinetic energy stored when landing.

TABLE 7 AIRPLANE CHARACTERISTICS WITH TWO 400-HP. ENGINES = 800 HP.

Hp.	V	A	w	W	Pay load	Passengers
800	100	7160	15	26400	6600	35.7
800	110	1325	15	19850	4950	27.5
800	120	1020	15	15300	3825	21.2
800	130	800	15	12000	3000	16.7
800	140	640	15	9600	2400	13.3
800	150	520	15	7800	1950	10.8
800	160	430	15	6450	1610	9.0

TABLE 8 AIRPLANE CHARACTERISTICS WITH TWO 500-HP. ENGINES = 1000 HP.

Hp.	V	A	w	W	Pay load	Passengers
1000	100	2200	15	33000	8250	45.9
1000	110	1650	15	24800	6200	34.5
1000	120	1270	15	19000	4750	26.4
1000	130	1000	15	15000	3750	20.8
1000	140	800	15	12000	3000	16.7
1000	150	650	15	9750	2440	13.5
1000	160	535	15	8000	2000	11.1

TABLE 9 AIRPLANE CHARACTERISTICS WITH THREE 200-HP. ENGINES = 600 HP.

Hp.	V	A	w	W	Pay load	Passengers
600	100	1170	15	17500	4375	24.3
600	110	880	15	13100	3275	18.2
600	120	675	15	10100	2525	14.0
600	130	540	15	8100	2025	11.3
600	140	425	15	6400	1600	8.9
600	150	345	15	5200	1300	7.2
600	160	275	15	4100	1025	5.7

TABLE 10 AIRPLANE CHARACTERISTICS WITH THREE 300-HP. ENGINES = 900 HP.

Hp.	V	A	w	W	Pay load	Passengers
900	100	1755	15	26300	6600	36.7
900	110	1320	15	19800	4950	27.5
900	120	1015	15	15200	3800	21.1
900	130	800	15	12000	3000	16.7
900	140	638	15	9600	2400	13.3
900	150	520	15	7800	1950	10.8
900	160	430	15	6450	1615	9.0

TABLE 11 AIRPLANE CHARACTERISTICS WITH THREE 400-HP. ENGINES = 1200 HP.

Hp.	V	A	w	W	Pay load	Passengers
1200	100	2340	15	35100	8800	48.0
1200	110	1760	15	26400	6600	36.7
1200	120	1350	15	20200	5050	28.1
1200	130	1065	15	16000	4000	22.2
1200	140	850	15	12750	3190	17.7
1200	150	690	15	10350	2580	14.3
1200	160	570	15	8550	2140	11.9

TABLE 12 AIRPLANE CHARACTERISTICS WITH THREE 500-HP. ENGINES = 1500 HP.

Hp.	V	A	w	W	Pay load	Passengers
1500	100	2930	15	44000	11000	61.0
1500	110	2200	15	33000	8250	45.8
1500	120	1700	15	25250	6325	34.6
1500	130	1330	15	20000	5000	27.8
1500	140	1065	15	16000	4000	22.2
1500	150	870	15	13200	3300	18.3
1500	160	715	15	10700	2680	14.9

The first point has been discussed in the preceding paragraphs, and needs no particular discussion here. The matter of control is simply a consequence of the fact that the air forces vary as the square of the speed while the inertia forces remain constant. The influence of air disturbances is merely one of effective incidence of the airplane. For example, suppose a lightly loaded airplane to be flying at an absolute angle of attack of 3 deg., and to meet a disturbance that increases the angle of

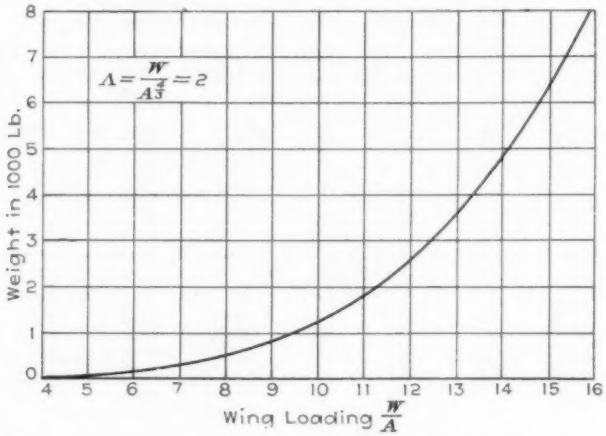


FIG. 4 RELATION OF WING LOADING TO GROSS WEIGHT UP TO 5000 LB.

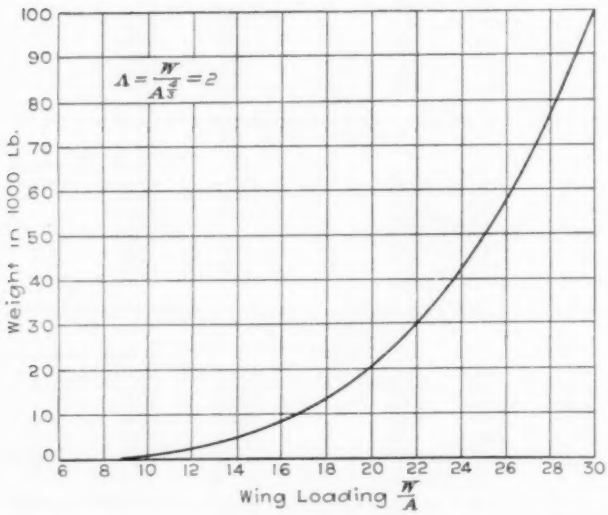


FIG. 5 RELATION OF WING LOADING TO GROSS WEIGHT UP TO 100,000 LB.

attack by 3 deg., then the lift will be doubled and the acceleration will be 2 g.

Now suppose a heavily loaded airplane to have twice the wing loading of the lightly loaded machine. The incidence will now be 6 deg. absolute angle of attack. The disturbance will again cause a 3 deg. increase in the incidence, and will result in an incidence of 9 deg. This is one and one-half times the original incidence, and will produce one and one-half times the original lift. The acceleration will then be 1.5 g. This effect is particularly noticeable when flying the same airplane, first light then fully loaded, under the same relative conditions.

The saving of structural weight is particularly noticeable where metal covering of the wing surfaces is to be used. The minimum thickness and weight of material is often the determining factor in the design so the same material in the same sizes will often carry the increased load. Also it is possible to build a structure to carry a much greater load without increasing the weight of the structure proportionately. The reduction of resistance is a consequence of the smaller wing area required to sustain the same load. The reduction in profile resistance of the wing may be a substantial saving, since the total wing profile of modern airplanes represents about 25 per cent of the parasite resistance.

The longer take-off run is serious only where the space for taking off is limited. Flying fields and airports are increasing their runways almost constantly so there will, in general, be ample space to take off with even the heaviest loaded machines. The problem of taking off from small emergency fields should not receive particular attention, for it is not the airplane's normal working environment. A railroad train may be required to run along the ties, but it certainly is not designed to do so. If it becomes necessary to take the plane off from some such limited space, it should be flown light. Geared engines are a decided help in reducing the take-off run. Their more general use seems to be merely a matter of time.

Wheel brakes are the principal mechanism used to reduce the landing run. If properly designed and installed they are very effective. Wind brakes have often been proposed, but they add complication, weight, and are effective only at the higher speeds. Since an airplane can in general land in a smaller field than from which it can take off, the provision of the ordinary wheel brake seems sufficient.

It is the author's observation that it is the way the plane lands that is much more important than the speed of the landing. A particular incident comes to mind. A small biplane, landing at about 40 m.p.h., was particularly hard to manage when once on the ground, while another larger and heavier machine landing at about 70 m.p.h. was so easy to control that it could be landed by the use of the stabilizer adjustment alone, and would roll to a normal stop without the necessity of any rudder control. This last machine although landing 30 m.p.h. faster than the first was certainly preferable even in cases of forced landings in small fields.

The kinetic energy stored in the two machines based on equal weight is more than three times as great for the fast-landing plane just mentioned than for the slower one. In case of a mishap, this most certainly would have been responsible for far greater damage to the larger than to the smaller machine, but the probability of a mishap with the small machine was out of all proportion to this difference in kinetic energy. Also, the larger machine was of such rugged construction that the final outcome would most probably have been in its favor if an accident had overtaken them both. Most certainly if the small machine had possessed the landing qualities of the larger one, the tables would have been reversed; still the heavier loaded machine will be of a more rugged construction than is economically possible in planes with relatively larger wing areas.

The point mentioned previously of the machine losing its buoyant feeling if it is excessively loaded, deserves considerable attention. The word buoyant is here used incorrectly because an airplane is sustained by dynamic forces and not by static ones as the word buoyant implies. It is, however, exactly the way the machine feels, and is used for lack of a better expression.

Early in the twentieth century it was observed by some naturalists that, although the wing loading of birds varied through quite large limits, there was a steady increase in loading, with increase in the size of the birds. Also, that sea birds had loadings much higher than did land birds of the same size. A careful investigation of the relationship of weight and area showed the land and sea birds to fall in two distinct classes, and the relationship to be

$$\Lambda = \frac{W}{A^{4/3}}$$

where W = gross weight of the bird
A = wing area of the bird
Λ = a constant
$^4/_3$ = an exponent.

When the weight was given in kilograms, and the area in square meters, the value of Λ was 7.6 and 8.4 for land and sea birds, respectively.

As birds have been man's model for flying machines down through the ages, it seemed logical to apply this relationship to airplanes as they now exist. The author has done this over a period of several years with considerable success. It was found that when the weight was taken in pounds and the area in square feet the constant Λ had values between 1 and 2 for most existing machines. The next point to determine was what was the upper limit of Λ beyond which some sort of trouble was sure to occur. Since airplanes were of much larger scale and powered mechanically and not by muscular effort, there was reason to believe that the value of Λ would be much larger when applied to airplanes than when applied to birds. This belief was justified and the upper limit of Λ was found by observations taken on a number of flight tests.

For monoplanes the maximum value of Λ was found to be very close to 2, although there was some small difference in different types and makes of machines. The limit was definitely shown by a decided change in type of airflow and a consequent loss of lift and increase in drag, although the plane had not yet reached its calculated stalling speed. It was found necessary to glide the machine at rather fast speeds to maintain the normal type of airflow. The phenomena is generally spoken of by pilots as "squashing" through the air.

At first, the results with biplanes were not so promising. The values were much lower and did not at first appear to follow the same law. It looked for a time as though extrapolation from monoplane bird to monoplane airplane was legitimate, but that extrapolation from monoplane bird to biplane airplane was not so. It was soon noticed that the results from equal-span equal-chord biplanes checked nicely with the results of monoplanes of approximately half the gross weight. This suggested the idea of considering each plane of a biplane as a monoplane and using for the gross weight the share of the load carried by each plane. When all the biplane data were reconsidered with this in mind the values checked those of the monoplane tests and were all in fairly satisfactory agreement. As is usually the case, the matter looked very simple in retrospect, and one wondered it had not been thought of in the first place.

The relationship of the equation

$$\Lambda = \frac{W}{A^{4/3}} = 2$$

is shown graphically in Fig. 4. This shows a graph with the maximum allowable wing loading plotted against the gross weight for gross weights up to and including 5000 lb. Fig. 5 is a graph of the same equation, but to a different scale and shows the maximum permissible wing loading, plotted against the gross weight for weights up to and including 100,000 lb.

It must be remembered that for biplanes the weight is not the gross weight but the weight carried by the wing in question. Some sesqui-planes by disregarding the small lower wing may reach values approximately those of monoplanes.

Conclusion

In conclusion it is well to mention the necessity of designing the airplane as a finished piece of machinery equipped with all manner of devices designed to minimize the maintenance, and keep the machine in the air long hours each day so that it has some chance of paying its way and eventually taking its stand along with other more highly developed means of transportation. In this paper it is seen

1 That very high speed is not economical for weight-carrying airplanes
2 That the pay load is greatly increased by the use of high power loadings with its resultant decrease in maximum speed
3 That high wing loadings accompanied by high landing speeds are not objectionable if the plane is designed to land properly.
4 That the maximum permissible wing loading is given by the equation $\Lambda = \frac{W}{A^{4/3}} = 2$
5 That biplanes must be treated as monoplanes each plane carrying its share of the weight.

And it should be remembered that he who undertakes a careful and rigorous analysis of the problems at hand, and designs accordingly, will eventually be rewarded with success.

AER-52-18

Airplane Servicing From the Operator's Point of View

By RICHARD C. MARSHALL,[1] CLEVELAND, OHIO

The necessity for periodic overhauls of equipment and the large financial outlay necessary to do this work properly form one of the operator's greatest problems. The author tells why centrally located shops equipped to do this work at lowest possible prices constitute the answer to the problem. This is an able and well-informed paper on airplane servicing, drawn from actual experience, and should be helpful to airplane operators.

THE difference between profit and loss, all other factors being equal, depends upon the efficient operation of the equipment involving the investment. In the case of the airplane, from a commercial viewpoint, it becomes primarily a question of ship air hours. A ship on the ground is not earning anything unless it is on display in a show or in a salesroom. It was built to fly and to carry a certain pay load, at a certain average speed, with a certain degree of safety. It behooves every operator, whether he operates one or more planes, to minimize his ship ground hours and to maximumize his ship air hours. Inasmuch as efficiency and ship air hours are in direct proportion to the condition of his equipment, it is of vital necessity that he maintain his mechanical equipment at the highest possible state of efficiency.

The problem of airplane servicing from the operator's viewpoint has grown in many instances more or less like Topsy. By the word operator is meant the air-transport operator of the mail, passenger, and express varieties, as well as the fixed-base operator, be he large or small.

It has been the author's experience that the first requisite of knowing what the equipment is doing is that there be kept an accurate log of each plane, each motor, and each propeller. The planes and motors are kept track of rather easily, but one is forced to number the propellers as to the hub and the individual blades. It is essential that a complete inspection chart be used each time a plane is flown and that each space on the inspection chart be used each time a plane is flown and that each space on the inspection chart be initialed by the mechanic who makes the inspection. Log books, too, are a source of worry. All log books are kept by the field clerks, who are responsible for their correctness.

The length of time that an airplane motor should be run is debatable, for even the manufacturer says that, with the different types of work, in the various climates the motor will function satisfactorily for varying lengths of time.

We began with using Whirlwind J-5 motors, which we ran 225 hours, including the ground time for running up and for idling for other purposes in the 225 hours. Then we raised the maximum time to 250. Experience teaching us that we were still far on the safe side, we made the maximum limit 275 hours and cut off the idling or running-up time entirely. As this time on our run amounts to about 10 per cent of the total time, as the log books were formerly kept, we have been able to boost our motor time in perfect safety from 225 to over 300, using the original method of computing the time.

The operator, again, is faced with the question of how long he shall run his plane before he has it completely overhauled and recovered. There does not seem to be any definite time at which one can say this work should be done. However, we have a number of planes that have been in constant service for two years and which are just now being completely overhauled. They have an average flying time of about 1400 hours.

One of the major problems of the operator is that of keeping his ships at peak efficiency through the periodical overhauling of planes and motors. The climate, of course, has a great deal to do with this, as stated before. However, the maintenance is much more important. The little things done from day to day in the way of maintaining the plane and motor, if done correctly, will do more to lengthen the life of the plane and motor than will any other one thing. We will take a Whirlwind J-5 Wright 225-hp. motor as an illustration.

After the engine has attained the age of 250 to 300 hours since its last overhaul, it is again in need of a complete tear-down and inspection. However, it may be true that the engine is functioning perfectly and turning up as well as ever. Nevertheless, certain vital parts have had considerable wear and, if not replaced, may cause motor failure at a critical time. Then, too, a more or less insignificant trouble may have been developing, which in a short time may turn into a major disability.

Because of this need for frequent overhauls on planes and motors, it is necessary for the operator either to maintain his own overhaul and service shops or to send his planes or engines back to the home factories, or to depend upon some other aeronautical organization which operates airplane- and motor-overhaul bases. It is the author's belief that the last-mentioned method is the most satisfactory one for the smaller operator to follow. Inasmuch as the smaller operators are greatly in the majority now and probably will be for a long time to come, the need for well-equipped, dependable, and economical repair bases for the general public's use has become imperative. Very few operators can afford to maintain their own overhaul shops and obtain satisfactory results; first, because of the large investment necessary in maintaining a fireproof building, in having specially designed equipment, in maintaining a large corps of technically trained mechanics, re-coverers, and ship builders, and then in the need for carrying costly inventories of parts and supplies. Perhaps it is just as well that the cost of the establishment of overhaul bases is well-nigh prohibitive, for the advantages offered by a few conveniently located, highly dependable aero repair depots, owned and operated by organizations of sound reputation, outrule whatever advantages a small operator might find in maintaining his own shops.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

One of the most important advantages is that the larger aeronautical repair-depot operator who has a constant stream of work going through his shops can effect many economies due to this large volume of business. He can cooperate closely with the Department of Commerce and with the manufacturers of planes and motors. He can afford to employ a higher salaried and more skilled type of workman. He can afford to carry a complete stock of parts and supplies at all times, thus eliminating the time that is lost by the average operator who often has to send to the home factory for the parts needed because he cannot afford to carry them in stock. These parts and supplies can be purchased in larger quantities and at a greater discount and are always of the latest production types and materials. And, too, the operator who specializes in overhaul work can so standardize all operations through the use of specially designed equipment that the repair costs are cut to the lowest possible figure. By adopting a bonus

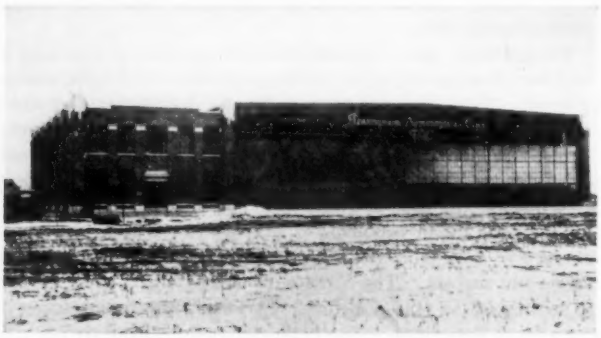


FIG. 1 AERO REPAIR DEPOT AT PONTIAC, MICH.

system, put in operation through a close study of time, prices, and labor, the charges can be lowered greatly without lowering, but even by constantly raising, the standard in workmanship. The cost can be made lower than could be secured at the factory where the original products were manufactured, due to the specialization of the man in this one line of work at the aero repair depot and the non-interruption in production by manufacturing operations, as usually happens at the home factory when outside overhaul work is taken in.

The Thompson Aeronautical Corporation, having its activities spread over four states, was faced with this problem of servicing its equipment. We found that our biggest problem was the question of how, when, and where to do this all-important major overhauling. So we established two aeronautical repair depots, one in Pontiac, near Detroit, and one at Cleveland. (See Fig. 1.) Then, we thought, if we could do our own work satisfactorily, we could lower our own costs and other operators' costs by doing as large a volume as possible, and so these depots were built to handle efficiently a large quantity of just this kind of work. Since these are excellent examples of the large overhaul bases which we believe the industry needs, it would perhaps be helpful to describe these bases and to outline the procedures followed in servicing planes and motors.

At both bases there are well-lighted buildings of brick and steel construction, because without the proper temperature and the proper lighting, the workman cannot do his best. Only the highest trained mechanics are allowed to work on motors and planes brought into these shops, because we have found that there is real economy in employing the higher class of workmen rather than having more lower-paid workmen supervised by a few experts.

Over a long period we have been taking as a beginning point or basis the time allowed by the manufacturers in various operations on their equipment and then having a time-study expert check each operation, and thus we have arrived at a constantly lowered number of hours needed to do a given job. This means, of course, economy.

Let us follow a motor through one of our depots and see just how we have tried to create efficiency and a saving in money for operators and how we have tried to solve our own problem of the greatest safety and reliability at the smallest cost per flying hour.

Again we will take the Whirlwind type of motor as an example. When the motor is brought into our depot, the first step is to receive the authorization to perform the necessary work from the owner or his representative. This is done by securing the owner's signature on a "request for repair" blank. This form, which serves as a contract is turned over to the service clerk, who in turn makes out a "job card." After these are filled out, the motor is placed in the shop. The foreman fills out a "receiving report," in which he lists all parts received and missing. After this has been done, the engine is turned over to the mechanics for dismantling. The Thompson Aeronautical Corporation has found that the greatest efficiency is obtained in tearing down or in assembling the motors by having two mechanics work together.

These workmen first place the motor on a stand and wash it down with a gasoline spray gun, to remove all grease or dirt that has collected on the outside. As the engine is disassembled, the parts are laid out on a specially constructed triple-decked parts rack, which has compartments so arranged for each group of engine parts that they can be found easily and quickly. These racks are on castors, which permit them to be wheeled over to the washing stand, the inspector's bench, and along the assembly line as the motor is being reassembled. Special tools have been developed which make a comparatively simple matter of the assembly or the taking apart of such a finely finished and accurately fitted mechanism as an aviation motor.

Following the initial tear-down, such integral parts as the front section, rear section, intermediate section, oil pump, fuel pump, and carburetor are taken apart.

Before the various parts can be inspected they receive a thorough cleaning. Therefore the parts rack is wheeled over to the washing stand, where each piece is sprayed with a gasoline air gun and cleaned with a stiff brush. Many of these parts can then be buffed and scraped to remove carbon deposits, in order that the inspection may be more accurate.

The inspector, who, like the other men, is a specialist in his line, goes over each part separately, checking for marks, crystallization, cracks, and excess wear. Practically all parts are checked carefully with a 30-diameter microscope. Many gages and micrometers are used in checking for excess wear. All parts found to be defective or worn beyond the toleration allowed are branded with red paint by the inspector to forestall any chance of their being used again. The inspector makes out an "Inspection Report," listing all machine work to be done, parts to be replaced, and component sections to be overhauled. (See Fig. 2.) Of course, such parts as the carburetors and magnetos are taken apart by men who specialize on these parts, with the special equipment necessary to inspect and thoroughly check and test them. These men must take a special course of training at the home factories of these assemblies and have all the equipment furnished by the home factories so that the job may in truth be as good as new when they are through repairing it. Nowhere in our shops do we use any equipment or parts for replacement not authorized by the maker of the original machine.

The machine work is given to the machinists. The cylinder barrels have previously been checked for tolerance, wear, etc., by the inspector. If the cylinders are beyond the toleration al-

lowed by the maker of the motor, the cylinders then are ground and honed in the machine shop.

It may be interesting to note that few operators who do their own overhaul work and maintain shops manned by skilled mechanics do their own cylinder grinding and honing, preferring to send this work out to regular machine shops. The main reason for this is the large investment which must be made in equipment and the expense of carrying an experienced grinder and honer, as there is a limited amount of this work in the average operator's overhaul requirements.

In connection with the honing process, the Thompson Aeronautical Corporation experienced considerable difficulty during the latter part of 1928, when it first started honing air-cooled motor cylinders, due to the cylinder walls being of a different texture after several hundred hours' use. It took several months of close cooperation with the makers of honing stones and the honing machine to get the right composition in the stone and the right speeds for the machine to prevent the pulling or "ringing" of the cylinders. The first stones we attempted to use made the metal ridge up or cut deep marks in the cylinder barrel. However, a satisfactory stone was developed, and for the past year and a half no trouble has been experienced.

Of course, if the cylinder is of the type that has a steel barrel and an aluminum head shrunk or screwed on, and the walls of the cylinder barrel are so badly marked or worn that the barrel cannot be used again, it must be removed and a new sleeve inserted in the aluminum head, if the owner so desires; otherwise a complete new cylinder is necessary.

Another problem which we have solved since entering the overhaul and service business was that of removing the cylinder head in the composite type of cylinder, without damaging it, from the steel barrel. The Thompson Aeronautical Corporation overcame this problem by developing a special gas furnace, in which the cylinder head alone is heated. The principal reason that this has been a difficult task is because of the improper control of the heat used; and, too, there has resulted the pulling up or wrinkling of the aluminum head when an attempt was made to remove the steel insert or sleeve.

Another operation that has caused an excessive amount of trouble and loss of time, but which has been solved, has been the removal and proper replacing of valve seats in the aluminum heads of the modern air-cooled cylinder. Previously, all of this work was done at the home factory of the motor builder, and many of the operators still send work of this nature to the home factory, but because of the great amount of this type of work that was coming to us, we had to develop equipment to do the job ourselves.

It might be asked why we did not secure the same type of equipment employed by the motor manufacturers. The reply is that we did not consider their methods satisfactory or fast enough, as they usually had very little of this kind of work to do. The loss of material which was injured and thereby lost was too great in most cases, due to the damage which resulted in attempts to remove the worn valve seat.

With cutters and jigs designed for the job, we use a large drill press to ream out the old valve seat. The cutters are designed to leave 0.002 to 0.003 in. of stock of the old seat, which can be easily removed with a sharp-pointed tool. This does a quick clean job. After the seat has been removed, the aluminum part of the cylinder head is placed in the gas furnace, and a stopper, which our mechanics have developed, is inserted in the barrel so that it is in line with the top of the steel barrel. Thus the heat is kept from the inside of the cylinder and therefore is prevented from distorting or damaging it.

When the aluminum head has been heated to the proper temperature, the cylinder is removed from the furnace, and the valve seat is quickly dropped in place in the expanded aluminum head. A tool, also of our own development, is then used to expand and roll the seat firmly into place. Through this equipment we are able to save 30 minutes on a cylinder, or $4^1/_2$ hours on a nine-cylinder motor. This of course is a big saving to the operator whose motor we are overhauling.

While the parts are being machined and reconditioned, a "material requisition" blank is filled out by the shop foreman, who lists all the new parts necessary to complete the assembly of the motor. This is turned over to the stock room, where the new material list is made up. This eliminates delay, and work can proceed without loss of time.

In the assembly operations, the greatest care is taken to keep all clearances at the precise exactness necessary. The assembly of the motor is done in the most practical and well-proved method, each case depending upon the motor type and size. Each operation is passed on by the shop foreman or an equally competent man. When completely assembled, the engine is washed down with a gasoline spray gun and dried, in order that any oil leaks developing during the block test may be positively detected and identified. The engine is then mounted on a test stand, located in the test building. This building is arranged so that the engines run under "outdoor conditions" at all times.

A mechanic, usually the foreman or the chief mechanic, watches the engine at all times while it is on the test stand, from a point in a room adjoining the test room where he can observe the engine and put it through various tests. Before him is an instrument board most completely equipped. Among the instruments on this board he has an oil-pressure gage, an oil-in and oil-out temperature gage used for indicating the temperature of the oil as it enters and leaves the motor, a recording tachometer which registers the r.p.m., a carburetor air-intake thermometer, and a thermometer to show the temperature of the water through which the oil coils pass. He also has a thermometer on the oil tank and a clock and a stopwatch at hand. Conveniently located at the bottom of the board is the throttle, spark, and switch control of the type employed in an airplane.

The test stand is built to withstand a 1000-hp. motor, and there is a practically complete absence of vibration. With the aid of a four-bladed calibrated propeller, the conditions are ideal for testing.

The standard block test for the Wright Whirlwind, Cyclone, Kinner, and other popular makes of motors in their class includes a four-hour run-in. The run-in consists of one hour at 700 r.p.m., increasing the speed by 100 r.p.m. every fifteen minutes thereafter, until the engine has reached the speed at which it is developing nine-tenths of its full rated horsepower.

During the run-in, readings are taken at the start of each change in speed of the oil temperature in and out, the oil pressure, the manifold vacuum in inches of mercury, and the carburetor air temperature in degrees fahrenheit.

Readings are taken and recorded every ten minutes during the nine-tenths full-rated-load test, as during the run-in, to observe the fuel and oil consumption.

Fuel-consumption tests are made at nine-tenths full rated load and at full rated load with mixture control at full-rich and at best setting. Two pounds of fuel are used in computing the fuel consumption. Oil consumption is figured on the same basis as the gasoline and at the same time the gasoline consumption rating is being taken.

In addition to the time required to make these tests, the engine is run at full throttle for a sufficient length of time to determine the maximum horsepower, with the mixture control at best setting.

The motor is given an idling test and an acceleration test. The idling test consists of running the motor for five minutes at 300

r.p.m., recording the time of start and finish for this run, the r.p.m. and the oil pressure. The acceleration test consists of opening the throttle slowly to idling speed and then quickly wide open, noting whether the engine accelerates satisfactorily.

When the run is completed, a careful inspection of the engine is made before removing it from the test stand to detect any sign of oil leakage or loosening up of any parts.

All test data are listed on the standard test sheets, together with information concerning any leaks or peculiarities that might be noted.

The engine is then dismounted and cleaned up. All exposed parts are greased and various other parts oiled and given a final check. The motor is now finished.

Of course, it is necessary thoroughly to calibrate each metal propeller each time the motor is completely overhauled, or about every 300 flying hours. We have in our aero-repair depots propeller-balancing stands and mandrels for the more popular propellers and hubs.

The propeller is first put on a pitch-setting table, and the pitch of each blade is checked, and it is then inspected for cracks or other flaws. It is then carefully balanced and polished, and it is ready for service again. Once every 1000 hours we have the blades pickled, or put in a solution that will show up any defects in the metal. This is a precaution against failure.

It is a fairly well-accepted rule that at about every 1000 hours of running time or 18 months of elapsed time a fabric-covered plane needs a general overhauling and checking up. For the average operator this is indeed a problem, because to do this work he needs blueprints of the airplane from the manufacturer. He needs to maintain a well-heated and well-ventilated dope

INSPECTION REPORT

PLANE NO. ..... STATION .....

MOTOR NO. ..... DATE .....

MOTOR

Mags. Cleaned: RH..... LH..... | Carburetor Tight..... Oiled..... | Oil Pump..... Tight.....
Mag Points set: RH..... LH..... | Rocker Push Rods..... Covers..... | Fuel Pump..... Tight.....
Spark Plugs Cleaned ..... | Rocker Axles Greased (25 hrs.)..... | Oil Filter Cleaned .....
Spark Plugs Set..... Tight..... | Valve Springs..... Clearance..... | Oil Drain Safetied.....
Wiring..... Connections..... | Valve Timing .....

Generator Cleaned ..... | Starter Cleaned ..... | Starting Motor .....
Brushes..... Commutator..... | Attachments ..... | Brushes..... Commutators.....
Oiled..... Connections..... | Operates..... DO NOT LUBRICATE | Oiled..... Connections .....

Motor Cleaned..... Mount Inspected..... Nuts Tight ..... Cottered.....
Propeller Lock Nuts..... Saftd..... Clamps..... Nuts Tight ..... Cottered.....
Propeller Pitch ..... Pitting Smoothed .....

POWER PLANT

Fuel Tank (wing)..... Connections..... Lines..... Fuel Strainer Cleaned.....
Fuel Tank (body)..... Connections..... Lines..... Fuel Relief Valve .....
Engine Fuel Pump ..... Connections..... Lines.....
Hand Fuel Pump..... Connections..... Lines.....
Oil Tank ..... Connections..... Lines.....
Oil Pump ..... Connections..... Lines.....

CONTROLS (CHECK OPERATION OF EACH)

Engine: Throttle..... Spark..... Mixture..... Starter.....
Hand Fuel Pump..... Fuel Cock..... Seat Position Controls .....
Ailerons ..... Rudder..... Elevators..... Stabilizer Adjustment.....
Control Stick ..... Bearing Lubricated .....

EQUIPMENT

Landing Lights..... Switch..... Lenses Cleaned..... Leads.....
Battery Terminals..... Leads..... Specific Gravity..... Liquid.....
Hand Fire Ext..... Pressure F. X..... Pressure..... lbs. Connections..... Lines.....
Running Lights..... Switch..... Compass Light..... Switch..... Cabin Light.....
Parachute Flares..... Loaded?..... Release Mechanism .....

WINGS

Alignment..... Attachment..... Flying Wires..... Landing Wires..... Wire Spacers.....
Struts..... Nuts..... Saftd..... Are all inspection doors in place and secured?.....
Ailerons..... Hinges..... Saftd..... Control Wires..... Pullies.....
Fabric: Upper..... Lower L. H..... Lower R. H..... Is dope OK?.....

FUSELAGE

Cowl: Engine..... Body..... Front Mail Comp..... Rear Mail Comp..... Pilots.....
Fabric..... Inspection Doors..... Secured? .....
Cabin Doors..... Glass..... Cushions..... Cabin Cleaned.....

EMPENNAGE

Fin..... Attachments..... Braces..... Alignment..... Inspect. Doors..... Fabric.....
Stabilizer..... Attachments..... Braces..... Alignment..... Inspect. Doors..... Fabric.....
Rudder ..... Hinges..... Horns..... Cables..... Inspect. Doors..... Fabric.....
Elevators ..... Hinges..... Horns..... Cables..... Inspect. Doors..... Fabric.....

TAIL SKID

Shoe..... Bearing Lubricated..... Shock Absorb. Cord..... Skid Tube.....

LANDING GEAR

Axles L. H..... R. H..... Struts L. H..... R. H..... Bolts..... Saftd.....
Oleo Units..... Is oil chamber filled to proper level?..... Strut Fairing..... Brakes.....
Wheels..... Tires and Tubes..... Pressure..... lbs Streamlines..... Wheels Lubricated.....

TEST RUN ON ARRIVAL

RPM: R.H..... L.H..... Both..... Idling.....
Oil Pressure..... Oil Temp..... Gas Pressure .....
Generator: Amperes..... Volts.....

NOTICE: Do not open motor up until oil temperature has reached 100° F. (40° Cent.). Do not run motor at full throttle on ground for more than a moment.

SERVICE

Gas: Main Tank..... Gals. Wing Tank..... Gals. Filled by.....
Oil: Drained out..... Qts. Drained by..... Put in..... Qts. Filled by.....

FINAL TEST

Run by Mechanic..... Date.....

Accepted ..... Report by .....

Pilot ..... Mechanic

WORK TO BE DONE LISTED ON BACK OF PAGE

FIG. 2 INSPECTION REPORT OF THOMPSON AERONAUTICAL CORPORATION

room, with the latest type of spraying and painting equipment. He also needs a welding room where the most difficult jobs can be done by experienced welders, because most of the modern planes are of tubular-steel construction. And then, of course, the average airplane mechanic is not competent to do a ship rebuilding job satisfactorily.

It is possible for the aeronautical repair-depot operator to secure from the various manufacturers blueprints of their types of machines and to arrange for the frequent inspection by the Department of Commerce inspector, so that the work will not be held up during process, as in our motor-overhaul jobs we have endeavored to cut down the man-hours on each plane rebuilding job.

In our aero repair depots the procedure in completely overhauling a plane is as follows, the routine changing very little, regardless of whether the ship is a monoplane or a biplane:

When the plane is brought into the ship rebuilding department, it is carefully checked over by the expert ship builder, and a "work order" is made out for the job, on which is itemized as nearly as possible all work to be done. The work is then planned accordingly. The first work, of course, consists of completely dismantling. The wings and all control surfaces are sent to the wing room for stripping and careful inspection of the structures. Here all repairing and truing-up that are necessary are performed.

Following the dismantling, the fuselage is taken into the fuselage department, where it is stripped. All bearings are examined and repaired, and replaced if necessary. The struts and members are cleaned, and a minute inspection is made so as to locate cracks, damaged parts, or structural defects. If of dural construction, it is sandpapered and cleaned up completely, following which it is moved into the paint shop, where a coat of clear weatherproofing protective material, such as Lionoil, is sprayed on.

If the fuselage is constructed of steel welded tubing, it is thoroughly cleaned, and two coats of anti-rust material are sprayed on. On some members it is necessary to apply several coats of enamel over the priming paint.

Members of the steel fuselage are drilled at points where the structure will not be weakened, and a special preservative liquid, usually some sort of oil, is forced into the interior of the tubes. The holes are then closed by welding. In addition to serving its purpose as a preservative, this liquid, when forced into the tubing, will disclose defects in the welding.

By this time the linen covering has been fabricated in the cover department, and as soon as the paint shop is through with the bare fuselage, the upholsterers are busy with the interior finish and the recovering is done.

The next step is doping and finishing the fuselage. In this work six coats of dope are applied, followed by two or three finishing coats of paint or lacquer, if desired. In the meantime the upholstering is also being completely renovated or made new as is required.

The control surfaces are completed, together with the wings, and are brought into the dope room for weatherproofing by using varnish or anti-rust paints. The wings and control surfaces are then covered and returned to the dope room for final finishing.

The plane is then reassembled, and while the riggers are busy, the upholsterers are completing their ork in the cabin or cockpit.

It is understood that only the highest grade of materials is employed in all work done. The reasons for this are obvious. It has been found that only the best dopes and paints should be used, the small additional cost being many times offset by the long life given to the plane in protection and appearance. It is the same with the linen and with any other materials.

The Department of Commerce inspectors have been making various inspections as the plane has progressed from dismantling to reassembling, and they are present or in the plane when it is test-flown and finally pronounced satisfactory.

The building and its arrangement are such that work can be moved from point to point with a minimum amount of lost motion, which is very important, more so in this particular kind of work than it is in the manufacturer's home establishment.

Usually the plane is called upon by its owner to fulfil a definite job, and when it is not in use it is a loss to its owner. Consequently, every effort must be made to expedite the work on it.

The procedure and methods used in overhauling motors and planes have been discussed at length, but one of the major items to an aircraft operator is the complete servicing of his aircraft. The line operations or day-by-day maintenance problems are not nearly so difficult to the operator as are his overhaul problems. The solution of the operator's problem along these lines is, as stated before, the providing of reliable, competent aero repair depots, located at convenient points, so that through the volume of work which they do they are enabled to keep the cost down to the minimum and which are every day striving to lower their own costs, so that they in turn may lower the cost to the airplane operator.

Discussion

D. W. Tomlinson.[2] From the standpoint of the large air-line operator it is essential as well as desirable that maintenance work of motors and planes be an integral part of the operations department of the organization. Usually in air-lines operations of any consequence there is a sufficient amount of motor and plane overhaul to justify shops for that purpose. When such work is done within the organization, the work can be better planned and executed in such a way that the entire plan of operations is most efficiently served.

Log books, both plane and motor, are in this organization, kept up by a chief clerk at division headquarters. Each plane carries an inspection form on which is recorded in abstract form the total time of the plane and all motors, with any other special information the log books contain on that particular ship or its motors. In addition we employ in our hangar a motor board. This is a large blackboard on which is kept up to date the total number of motor hours since overhaul of each engine then in service. With this data there is also recorded for each motor the time it has run since the several routine operations required in service have been performed, such as oil changed, valves checked, pushrods greased, magnetos checked, strainers cleaned, etc. This provides a visible record, which may be seen at a glance, of the status of all engines in active service.

Regarding the number of hours which any engine may safely be run, the writer believes it depends, as the author states, on the conditions under which that particular engine operates. The greatest factor in the operating life of the modern engine is the care with which it receives its routine service checks. This assumes, of course, that the engine at its initial point of operation was in perfect condition. Experience has shown us that engine wear, and the resulting life of the engine between overhauls, depends also on the amount and kind of dirt which goes through the engine with the fuel; in other words, dust off the field. If present-day engines could be assured of a fuel supply free of grit, their operating life would in many instances be practically double. Up to certain limits the writer does not believe in dismantling an engine for inspection and overhaul so long as it is functioning perfectly. Invariably a motor will give timely and sufficient warning of wear due to use. We have found it practicable to run Wasp engines 500 hours between overhauling.

[2] Manager of Operations, Western Division, T.A.T.-Maddux Air Lines, Los Angeles, Calif.

The points brought out by the author in detailing the procedure followed in the motor-overhaul shops are excellent. The routine outlined is basically that employed in our own overhaul department.

The author makes one point that is well worth consideration by all operators, which is that it is far more economical to employ only higher-paid skilled mechanics than to carry one or two such men with a larger number of helpers. In our shops we use only the most experienced and reliable men. The only place where we have found it advisable or economical to employ helpers is for such work as changing oil in ships at night, fueling planes, running oil in the cleaning machine, cleaning the motor parts in the teardown room, etc.

Only all-metal Ford monoplanes have been used so far in our operations. As yet there has been no necessity for major overhauling of any of these planes. Routine maintenance and incidental repairs are made by a special metal force. Planes are very carefully inspected as soon as they arrive at the base field. An inspector goes through the interior of the wings and examines the ship generally for any defects. The life of these planes before an overhaul would be a debatable question, and resolves itself down to the life of the metal of which the planes are built. We have several planes with well over 2000 hours of operation. These planes have, to date, shown no indication whatever of any failure or deterioration. Unless crystallization or fatigue of the metal develops it is the writer's opinion that these ships may give perfect service without major overhaul for some five to ten years. This presupposes painstaking routine maintenance throughout this period.

The problem of renewing cylinder heads, valve seats, and regrinding cylinder barrels has been taken care of by us in returning the parts to the factory for such operations, with the exception of cylinder regrinding, which is done locally, the volume of motor overhaul not having developed to the point at which it would be economical for us to perform these operations in our own shop.

To facilitate the testing of motors we have employed a test stand mounted on the rear end of a truck. All test instruments are mounted on a board between the driver's seat and the motor mount. This instrument board, with all engine controls, is closed for the convenience of the mechanics making a test. This stand has been found very satisfactory. It greatly simplifies the mounting and dismantling of the engines from the test stand, as the engine is backed up to the motor overhaul shop itself. As minor failure occurs during the test the motor can be brought without delay directly to the overhaul shop for the adjustment or change of parts.

Alfred C. Hurlbut.[3] Maximum performance at a minimum maintenance cost is becoming one of the most important factors in the aircraft industry. A manufacturer can design a ship so that all parts can be easily serviced, but the ship must be inspected and serviced constantly to insure continuous dependable performance.

Like any other piece of machinery, an airplane must have constant care and cleaning to keep it functioning properly. However, it differs in that the need for repairs must be anticipated before they become serious. This, of course, emphasizes the need for trained airplane mechanics who know what parts are most likely to show excessive wear, how to look for them, and how to repair them.

For example, the bolt holes in certain places may become elongated due to vibration. It is an easy matter to enlarge these holes and put in a larger bolt, and this is often done, with the result that sometimes there is not enough stock left in the member so drilled to give it sufficient strength, and it gives way. If this member had been filled, and then drilled to its original size, it would have been as good as new.

Another example is improper cleaning. Many ships are cleaned with gasoline, as it is the quickest way, or they are cleaned with nothing but water. Either of these methods will cut the life of the finish in half. The ship should be thoroughly scrubbed with castile soap and water, and then carefully rinsed to remove the soap. If the ship is finished with lacquer, it should be gone over with a good lacquer polish to bring the life back. Thorough cleaning is also important, because only when the ship is clean can small cracks or failures be discovered and repaired before they become serious. These are only two of the many examples which show the importance of good servicing.

Airplane operators are demanding planes that can be kept in the air a maximum number of hours per year and a ship that is designed so that inspection, lubrication, and replacement of parts may be easily accomplished. Such ships can usually be operated at a lower cost than a plane of more complicated construction. The Detroit Aircraft Corporation realized this fact, and all models are so designed that there are a minimum number of working parts, and all of these are easily inspected and serviced by removing several cotter keys from an inspection plate or opening a zipper fastening. This enables the pilot to make his own inspections or to have them made quickly at a minimum cost.

The service departments of manufacturing concerns, operating lines, and general service hangars are faced with problems which in most cases are similar. Through years of experience, we have evolved service departments which operate smoothly and efficiently.

Any ship coming up the runway to the hangars is met by a representative of the service department with his order pad and pencil in his hand, who takes the customer's signed order for any work he wants done and who is held responsible for the ship until it leaves. He makes out the order, sees that all jobs are done, that they are paid for, and that the customer is satisfied. Anything which he can do personally for the pilot, such as arranging hotel accommodations if the pilot is staying overnight or getting weather reports, is gladly done.

The personnel of the service hangar must comprise experienced mechanics not specialists in one line, but men who can do various kinds of work. The service hangar is continually running into new problems which a crew of skilled mechanics can solve and repair with a minimum amount of lost motion. Many times the service department is called upon to do work in a field miles away from the hangar, and usually the telegraph or telephone call which orders them out to the job does not give complete information. Upon arriving at the scene, the mechanic may find that, besides the broken parts he is prepared to replace, he must also patch fabric or splice a control cable. Valuable time is lost and extra expense to the customer is incurred if the service department does not realize these possibilities and anticipate them by sending an experienced man and necessary materials to take care of any damage which might be caused by such an accident.

For example, the Ryan Division service department in St. Louis received a long-distance call for service. The pilot reported an axle washed out in overshooting a field and running into a ditch, and asked that a mechanic be sent out with a new axle by airplane at once, as they must get the ship in the air as soon as possible to fill an important engagement. He stated nothing else was needed. The mechanic and axle were sent in an airplane, along with some extra parts which might possibly have torn loose. Upon arriving at the scene of the accident the mechanic discovered the fuselage axle fittings badly bent and the bottom fairing broken, but he was able to make the necessary repairs because

[3] Service Manager, Ryan Division, Detroit Aircraft Corporation, St. Louis, Mo.

the possibilities had been anticipated. If he had not taken these parts along, an important engagement would have been unfilled and an additional expense to the customer incurred by ordering more parts shipped.

Too often ground crews lose sight of the responsibility they undertake when servicing a ship; therefore, the mechanics should be under the direction of a foreman who understands the problems which face a pilot and who realizes his responsibility when he inspects the finished work and approves the ship for flight.

A service hangar should have a machine shop, a dope shop, a sheet-metal shop, welding equipment, woodworking tools, and miscellaneous equipment necessary for checking pitch, balance and track of a propeller; vises, jacks, hoists, and all the special tools required for a particular motor. A manufacturing plant has an advantage in this respect, as many of the departments can be used for service work without duplicating equipment in the service hangar.

If the service hangar is specializing in the maintenance of any particular type of ships, a supply of the parts which need replacing from time to time should be carried in stock. It is our policy to carry in stock a supply of parts for all models, obsolete and present, to enable the service department to make immediate delivery of any part. This policy saves much time and money for customers.

If a service station is connected with a sales agency, it can be of great value through its contact with visiting pilots who often furnish leads which result in the sale of new ships. If it is connected with a manufacturing plant, it can be of assistance in many respects. It is a great help to the production and engineering departments, as reports on how various parts perform or stand up will help the engineering department to design many improvements.

Our service departments are operated with the fact in mind that a satisfied customer becomes a booster and that operating at a profit, although important, is a secondary consideration.

We feel that when a ship is sold, our responsibility is just beginning. The customer is given help and advice on the care and maintenance of the ship and an invitation to write or come in for suggestions at any time. He is particularly asked to come in for an inspection before his ninety-day guarantee on defective parts runs out.

AER-52-19

Vibration Characteristics of Aircraft-Engine Crankshafts

Report on a Series of Tests in Which a Recording Torsiometer Was Used

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IT HAS long been realized by designers of internal-combustion engines that calculation of stresses in crankshafts was incomplete without an investigation of the torsional-vibration characteristics of the engine. Marine designers have been compelled to take this phase of design into account because of the nature of the drives used and the extremely favorable conditions for large-amplitude torsional vibrations. It is only because of the shorter drives and the comparatively stiffer crankshafts used in aircraft engines that the torsional periods have not forced themselves on the designers of such engines at an earlier stage. There has also been a feeling among those actively engaged in such work that calculation of torsional period was too involved and complex and too little was known of the phenomenon to make it worth while to take account of it. As a result, engines are built and run regardless of critical periods, and when a "rough spot" is observed in the operating range, the engine speed is varied to avoid running at that particular point. An example in point is the Liberty-12 aircraft engine. Calculation of crankshaft stresses does not reveal any marked weakness and yet there are speeds within the operating range where crankshaft failures occur. It has been found by numerous tests that there are three critical speeds at or near which torsional vibration is encountered. These speeds, as will be pointed out in connection with test records, are 1000, 1333, and 1715 r.p.m. with a frequency of 100 vibrations per second. Two of these periods, most unfortunately, occur at speeds which are encountered in operation. The normal rated speed of the Liberty-12 is 1700 r.p.m., and when fitted with a propeller for supercharging, but taking off without supercharge, the speed is pulled down to 1350 or 1400 r.p.m. A consideration of this design from the torsional-vibration standpoint would have indicated the desirability of removing these periods to a little-used part of the speed range. Because of the smoothness of operation of the Liberty-12 at 1450 to 1550 r.p.m., the pilot prefers to use this speed range at the expense of output. Another design of large 12-cylinder Vee engine has a critical period of vibration at 1800 r.p.m., its rated speed. The roughness of this engine, which was partially due to poor distribution, caused numerous failures of pipes, accessories, and water joints. To remedy this serious defect, a spring coupling was incorporated in the geared models. This seems to have the effect of making all speeds about equally bad in respect to torsional vibration. (See Figs. 11 and 13.) The coupling did, however, reduce the impact load on the teeth of the reduction gearing.

It would appear to be ideal to design a crankshaft so that no torsional periods would occur within its operating range. This is readily done in the case of the radial engine because of the short, stiff crankshaft used. In the case of the 4- or 6-cylinder line engine and the 8-cylinder Vee, it is still possible to avoid the primary critical period. The 4- and 6-cylinder line engines, however, will have torsional periods at one-half the primary critical speed by reason of the double-frequency inertia torque which is characteristic of these engines.

The inertia torque characteristics of a number of engine types have been computed and published in the *Automobile Engineer*, September, 1929. In the curves, the fact is brought out that the torque due to inertia is, in the 4- and 6-cylinder arrangements, greatest in the propeller-end crankcheek and main journal. In the 12-cylinder Vee, the stress due to this factor is greater in the journal between cylinders 5 and 6. This would seem to explain why the Liberty crankshaft almost always breaks at one of the cheeks between 5 and 6 instead of at the last cheek. In the case of the radial engine, inertia torque is a negligible factor.

In the case of the 8-cylinder and 12-cylinder Vee, the frequency of the inertia torque is equal to that of the power impulses. This frequency is too high in the 12-cylinder Vee to permit of designing the crankshaft so as to eliminate torsional vibration even up to a speed of 2000 r.p.m., since the equivalent flywheel effect of the necessary reciprocating and rotating parts combined with that of the crankshaft itself would make necessary a stiffness which cannot be incorporated in a practical design. It therefore appears that the best that can be done is to place the rated speed just below one of the critical ranges. This allows a considerable throttling range before the next lower period is reached. The reduced torque impulses at this latter period are not sufficiently serious to cause dangerous torsional vibration.

The study of torsional vibration by the Power Plant Branch, Matériel Division of the U. S. Air Corps, began at McCook Field prior to July, 1927. It was not, however, seriously undertaken until the summer months of that year. The project was carried through the summers of 1927 and 1928 and from June, 1929, until the present. It is felt that there is still much to be learned in spite of the mass of data already taken.

Instrumentation

The question of an instrument suitable for recording the vibrations actually encountered proved to be a serious one. Many types were considered and discarded as impractical for various reasons. The electromagnetic type of instrument gives a beautiful record of the relative velocity of the crankshaft and some uniformly rotating reference member. This record must be integrated in order to show the displacement record and of course does not give the amplitude of vibration. In the case of a vibration of harmonic form, the velocity record also gives the frequency, but the rather complicated displacement curves observed are not readily studied by any record other than the displacement-time record. It was therefore decided that the desired instrument should draw a graph of torsional displacement against time. An ingenious electrical device has been developed by Dr. H. Thoma, Karlsruhe, Germany. The novel feature of this instru-

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

ment was the use of toothed condenser plates, one attached to the crankshaft and the other to a uniformly rotating member. The variations of capacity of this condenser were used as in the condenser microphone to modulate a vacuum-tube circuit, the plate current of which passed through an oscillograph. This arrangement is capable of being calibrated and gives a displacement-time record. Another device which has been used successfully is Dr. Geiger's well-known torsiograph. It was by means of these instruments that the failures of the power plants on the *Graf Zeppelin* were analyzed.

An early design of torsiometer used by the Matériel Division incorporated links and pins to impart motion to an indicating hand. The latter moved over a scale graduated in degrees and was read by means of a neon stroboscope. Rapid wear proved to be fatal to this type of instrument, in spite of case-hardened

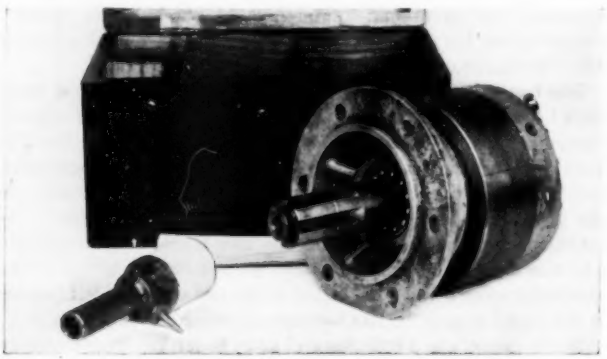


FIG. 1 VIEW OF INSTRUMENT SHOWING METHOD OF ATTACHMENT TO ENGINE-STARTER MOUNT

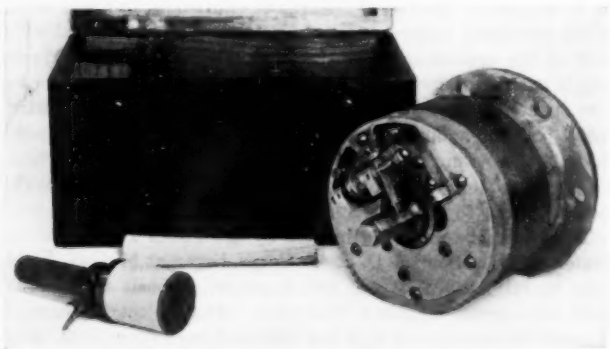


FIG. 2 VIEW OF INSTRUMENT SHOWING DETAILS OF RECORDING MECHANISM

parts. In the spring of 1928, the idea occurred to the author that a flexible cable of small size could be used, and this was accordingly incorporated in the instrument used during the summer of 1928. Figs. 1 and 2 show general views of the instrument. The arrangement makes use of pulleys and a drum on the shaft carrying the pencil. This has proved very successful and is quite free of backlash. The cable eventually wears out but is readily replaced and, under the severest conditions so far met with, the life of the cable is several hours of continuous operation. The record is made on ordinary indicator paper. A drum is provided which retains the paper, and since the pencil revolves with the instrument, the drum is held in the hand, inserted over the pilot provided for it, and quickly pressed in and withdrawn. The last $^1/_{32}$ inch of motion of the drum operates a bell-crank system and presses the pencil against the paper. A pointer is provided on the drum, and with the engine set on top center of one cylinder, the beginning of the card is lined up with the pencil, the pointer being vertical. This serves to locate the crank positions on the records. The two revolutions in the engine cycle are superposed, hence the complete cycle crosses the card twice before it repeats. In the present design, the records show a displacement of $^1/_8$ inch for each 1 deg. of torsional displacement of the crankshaft. The natural frequency of the spring-driven flywheel is 10 to 12 vibrations per second. This frequency is necessitated by the high accelerations to which aircraft engines are subjected. Fig. 3-*d* shows the calibration record obtained at a speed of 30 r.p.m. The flywheel was plucked and released to obtain a record of the damped vibration of the spring-driven flywheel.

TEST RESULTS ON VARIOUS ENGINES

A number of records from various engines are presented in Figs. 4 to 18. In Fig. 4 is presented a series of cards taken with a standard Liberty metal propeller with blades set at an angle of 25 deg. at 42-in. radius. This held the full-throttle speed down to 1330 r.p.m. Two synchronous periods are evident, one at 1000 r.p.m., card (*c*), and one at 1330 r.p.m., card (*j*). In both cases, the frequency is very close to 100 vibrations per second.

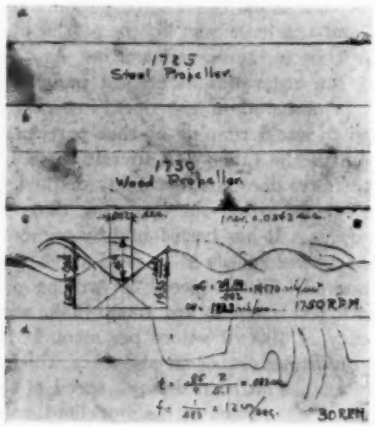


FIG. 3 (*a*) AND (*b*) RECORD OF TORSIONAL VIBRATION AT NOSE END OF LIBERTY CRANKSHAFT WITH METAL AND WOOD PROPELLERS. (*c*) METHOD OF DETERMINING ACCELERATION FROM RECORDS. (*d*) RECORD OF NATURAL VIBRATION OF FLYWHEEL OF INSTRUMENT

At 1000 r.p.m. there are 12 vibrations corresponding with the 12 power impulses per cycle or 6 per revolution. At 1330 r.p.m there are 9 vibrations per cycle, or $4^1/_2$ per revolution. These two periods would be accounted for by the presence of twelfth and ninth harmonics in the single-cylinder torque curve. Cards (*l*) and (*m*) show the effect of cutting out one set of plugs. In the Liberty, both plugs were near the center of the combustion space, hence little effect is observed other than the loss of r.p.m. Card (*n*) shows the effect of retarded spark at full throttle. The speed dropped to 1000 r.p.m., but since maximum pressure was reached later in the cycle, the deflections were greater than those in card (*c*). Vertical lines were drawn indicating approximately maximum torque (20 deg. after top center) on the cylinders marked at the bottom. It is observed that all 12 vibrations at 1000 r.p.m. and 6 out of 9 at 1330 r.p.m. show maximum acceleration at about the time of maximum torque.

Fig. 5 presents the records taken on the same engine with a propeller setting of 17 deg. at 42-in. radius. Here the maximum speed was 1700 r.p.m., the rated speed of the Liberty-12. In this set of records, three resonant periods are observed. That at 1000 r.p.m. is of small amplitude, due to the small power impulses.

The period at 1330 is apparent in cards (*g*) and (*h*), followed by a comparatively smooth range, cards (*i*) to (*l*). The third resonant period occurs at about 1700 r.p.m., as shown in cards (*o*) and (*p*). The frequency is the same as at the two lower periods. It is to be noted that 6 out of 7 of these vibrations show maximum acceleration close to points of maximum torque. This period would indicate a seventh harmonic in the resultant torque curve. No records have been taken at speeds in excess of the third resonant period on a 12-cylinder Vee engine.

Fig. 6 shows a set of cards taken on a 12-cylinder Vee engine of about 1150 cu. in. displacement. These were taken on the dynamometer using a rather heavy thermoid coupling. The compression ratio of this engine was 7.3 to 1 and the engine was operated at the maximum load it was believed safe to carry at each speed up

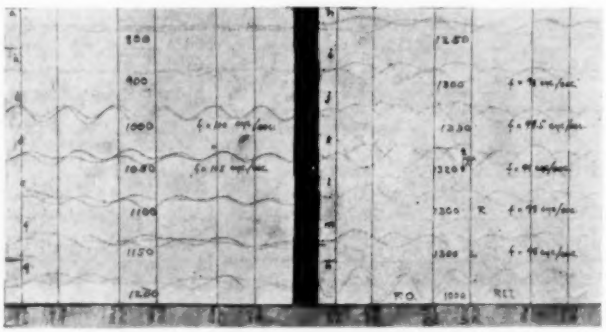


FIG. 4 TORSIONAL VIBRATION IN LIBERTY-12, WITH PROPELLER SET FOR 1330 R.P.M. AT FULL THROTTLE

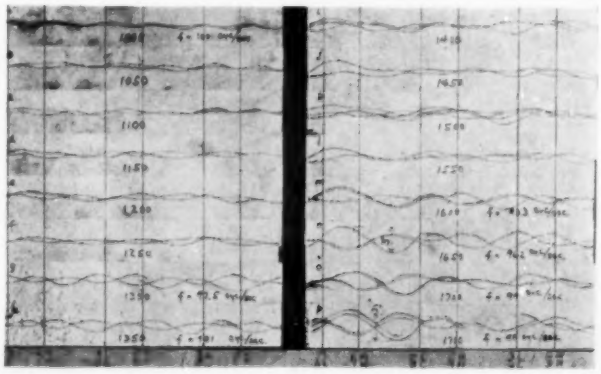


FIG. 5 TORSIONAL VIBRATION IN LIBERTY-12, WITH PROPELLER SET FOR 1700 R.P.M. AT FULL THROTTLE

to 1800 r.p.m. From this point, cards were taken at full throttle up to 2700 r.p.m. This was done in order to record the worst conditions obtainable in normal operation. Vertical lines indicate cylinders at top center. It is observed that definite resonance appears at 1600, 2100, and 2700 r.p.m. These periods bear the same relation to each other as did 1000, 1330, and 1700 r.p.m., in the case of the Liberty-12. If the frequency is accurately 160 vibrations per second, these resonance points should occur at 1600, 2130, and 2740 r.p.m. It is evident from the records that 2500 r.p.m. should be an excellent full-throttle speed, since this would allow a throttling range of 400 r.p.m. before encountering resonance, reducing the power output on propeller load to 59 per cent of that at 2500 r.p.m. It is doubtful if the effect of the resonant vibrations at 2100 r.p.m. could be felt at this reduced output. In climbing, the engine might be called upon to operate within this critical range at full throttle, but the duration of such full-throttle operation would not be great enough to cause crankshaft failure. Unfortunately, no records are available on this engine with propeller load. It will be pointed out in connection with another set of cards that it is unsafe to draw conclusions from dynamometer records, because of the effect of the coupling used. In order to approximate propeller loading, the moment of inertia of the coupling should be large.

Fig. 7 presents a very interesting record of a 6-cylinder line engine of about 470 cu. in. displacement. In this engine, resonance with the power impulses could not be observed, since it would not occur until speeds in the neighborhood of 3200 r.p.m.

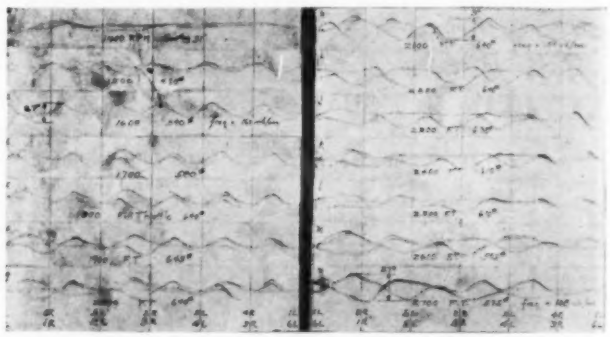


FIG. 6 RECORDS FROM ENGINE NO. 2, TABLE 1, TAKEN ON DYNAMOMETER COMPRESSION RATIO 7.3 TO 1, FULL THROTTLE, FROM 1800 R.P.M. TO 2700 R.P.M.

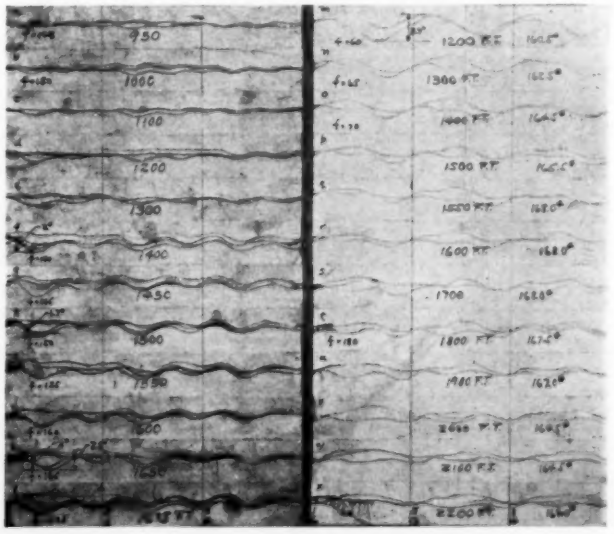


FIG. 7 RECORDS FROM ENGINE NO. 4, TABLE 1, TAKEN ON PROPELLER LOAD, IN THE AIRPLANE (*a*) TO (*l*) AND ON DYNAMOMETER, USING THERMOID COUPLING (*m*) TO (*x*)

were reached. This is far beyond the operating range of the engine. There should also appear a range of resonance at a speed of about 2130 r.p.m., but the available propellers would allow the engine to turn up only to 1875 r.p.m. Cards (*a*) to (*l*) were taken on propeller load in the airplane, while cards (*n*) to (*x*) were taken on the dynamometer. In the latter case, a thermoid coupling was used to couple the engine to the dynamometer. Since it was of small size, its effect as a flywheel was negligible and the actual crankshaft vibration was only slightly apparent in the records. These cards would seem to indicate that a coupling with appreciable hysteresis, such as that offered by the thermoid disks, might be a desirable type to incorporate in a geared engine. The chief objection to the thermoid coupling would be, of course, its size.

Figs. 8 and 9 show cards taken on a large 9-cylinder radial engine. Fig. 8 is from the direct-drive engine and Fig. 9 is from the 2 to 1 geared engine. The gearing is of the Farman type and does not incorporate any type of spring coupling. There is, however, indication of a considerable amount of flexibility in the reduction gearing, since the resonance period is lowered from about 2100 r.p.m. to about 1450 r.p.m. Assuming the same moment of inertia in both engines, the ratio of torsional stiffness from center of cylinders to edge of propeller in the direct engine is to that of

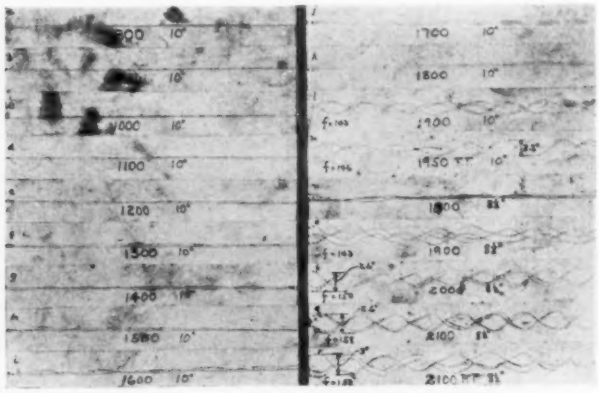


FIG. 8 RECORDS FROM A LARGE DIRECT-DRIVE 9-CYLINDER RADIAL ENGINE, ON PROPELLER LOAD, WITH TWO PROPELLER SETTINGS, 10 DEG. AND 8½ DEG. AT 42-IN. RADIUS

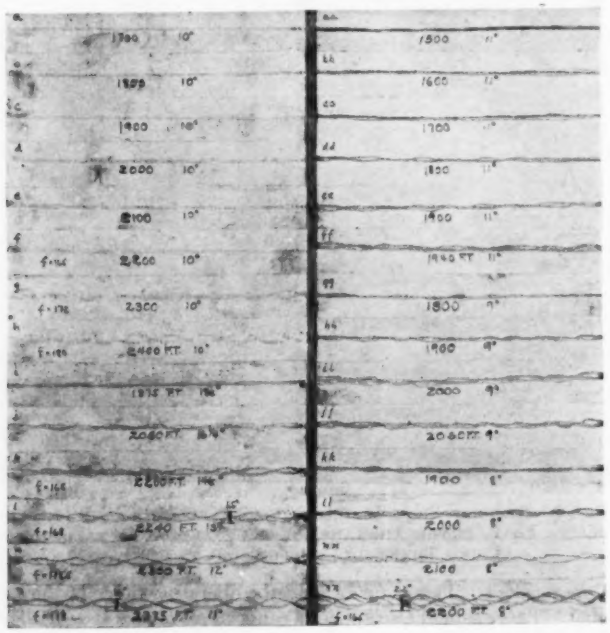


FIG. 10 (*a*) TO (*n*), RECORDS OF LARGE RADIAL ENGINE (DIRECT DRIVE) CHOSEN FOR SAMPLE CALCULATION OF RADIAL ENGINE, (*aa*) TO (*nn*), RECORDS OF ANOTHER LARGE RADIAL ENGINE OF 9 CYLINDERS

the geared engine as 2100^2 to 1450^2, or 2.1 to 1. It is hard to say where this flexibility is introduced, as this type of reduction gear is very much like an automobile differential in which the spider drives the propeller and one of the bevel gears is held stationary, the other being driven by the engine. The frequency of the direct crankshaft is about 158 vibrations per second, while that of the geared shaft is reduced to about 109 vibrations per second. It is observed that torsional vibration occurs in the direct engine at normal operating speed of 1900 r.p.m., while in the geared engine, the operating range from 1600 to 1900 r.p.m. is free of resonant vibration. The range from 1300 to 1550 r.p.m. is practically never used. Hence, this condition is satisfactory in the geared engine. In the case of the direct engine, however, stiffening of the crankshaft is indicated in order to eliminate possible trouble due to critical crankshaft vibration.

Fig. 10 shows records from two other large 9-cylinder radial engines, both of them direct drive. Both are free of critical vibration up to about 2100 r.p.m., which is in excess of normal rated

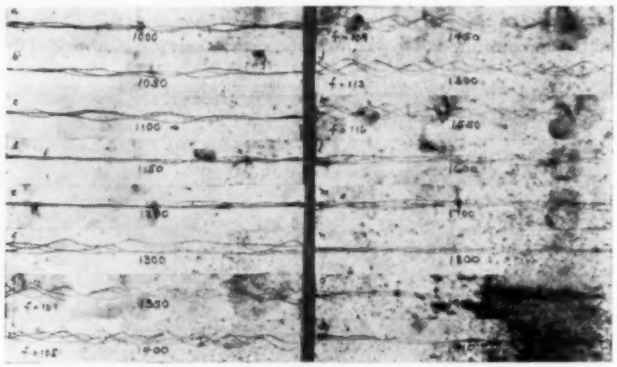


FIG. 9 RECORDS FROM ENGINE SAME AS IN FIG. 8, BUT WITH 2-TO-1 FARMAN TYPE REDUCTION GEAR

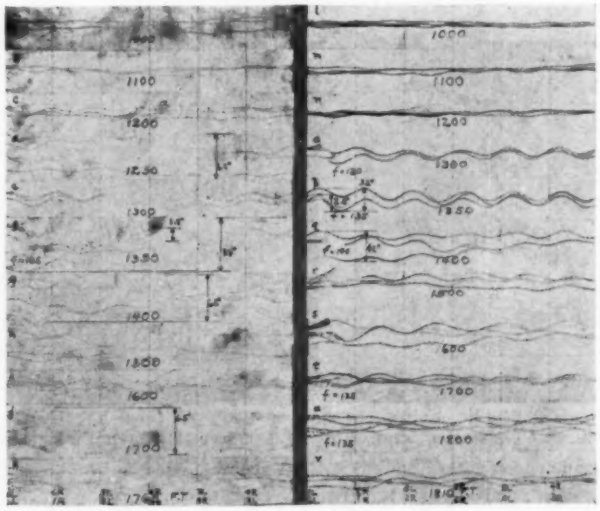


FIG. 11 (*a*) TO (*k*), ENGINE NO. 6, TABLE 2, ON PROPELLER LOAD, SHOWING EFFECT OF BAD DISTRIBUTION. (*l*) TO (*v*) SAME ENGINE WITH CASINGHEAD GASOLINE TO SMOOTH OUT DISTRIBUTION

speed. The critical frequency of the first appears to be about 175 vibrations per second and of the other, about 165 vibrations per second. These records constitute proof that the large radial engine can be so designed as to be free from trouble due to torsional vibration up to a speed as great as can be utilized before valve-gear trouble makes itself felt, and where inertia loading of the crank bearing becomes excessive.

Fig. 11 is an interesting set of records from a large 12-cylinder Vee engine. Cards (*a*) to (*k*) were taken, using domestic aviation gasoline and 5½ cc. of ethyl fluid. It is to be noted that while the period of the crankshaft is apparent at 1350 r.p.m., the variations from cycle to cycle are much greater in amplitude.

This was thought to be due to faulty distribution, hence the fuel was changed to highly volatile casinghead gasoline and cards (*l*) to (*v*) were taken. The engine operated much more smoothly on this gasoline, showing that a large part of the roughness of the engine was due to poor carburetor and manifold arrangement. It is to be observed here that the crankshaft has a distinct torsional period at 1350 r.p.m. and another at 1800 r.p.m. Both of these show a frequency of 135 vibrations per second, there being

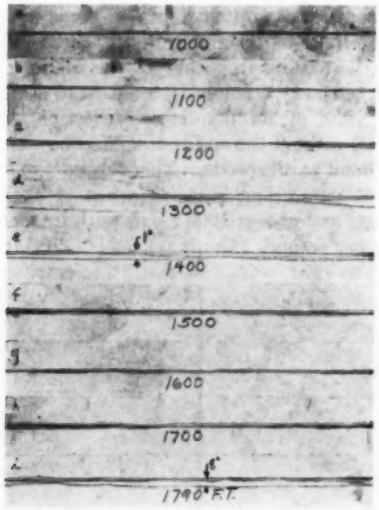


FIG. 12 SAME ENGINE AS IN FIG. 11, BUT RECORDS TAKEN FROM NOSE END OF SHAFT

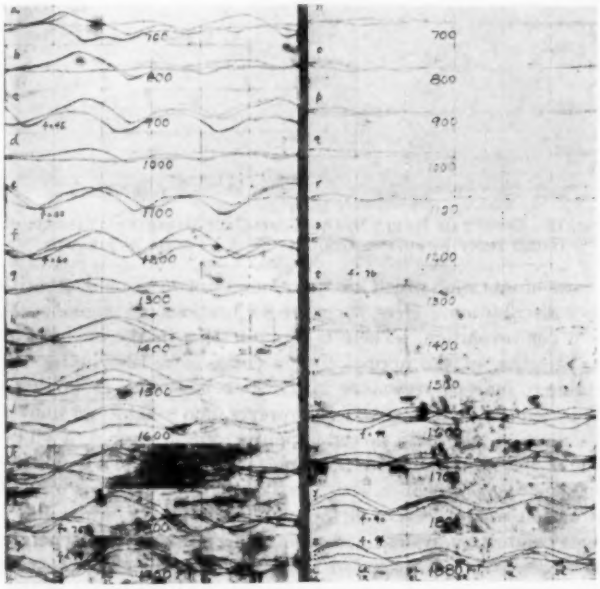


FIG. 13 (*a*) TO (*m*), ENGINE NO. 7, TABLE 1, SAME ENGINE AS IN FIGS. 11 AND 12 BUT WITH REDUCTION GEAR INCORPORATING A SPRING COUPLING. (*n*) TO (*z*), SAME AS (*a*) TO (*m*) BUT WITH RIGID-GEAR DRIVE

12 vibrations per engine cycle at 1350 r.p.m. and 9 at 1800 r.p.m. Since the rated speed of this engine is 1800 r.p.m., the engine is called upon to operate for extended periods within this second critical range. Fig. 12 shows a set of records from the propeller end of the engine, using the same fuel as was used in cards (*a*) to (*k*), Fig. 11. These records show quite definitely that deflections occurring at the rear end of the crankshaft do not indicate accelerations of the shaft and propeller as a unit. In resonant vibration, the shaft probably deflects according to an elas-

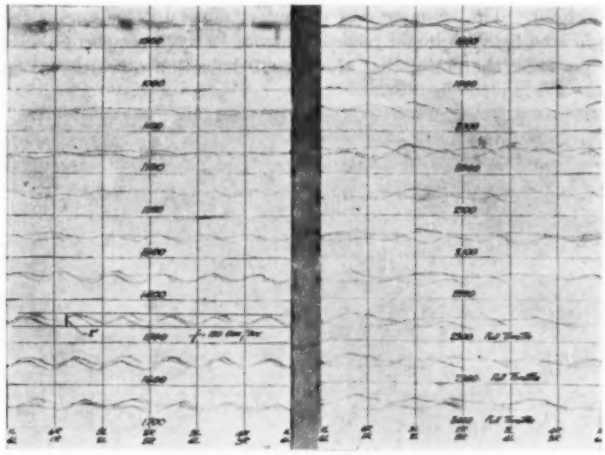


FIG. 14 ENGINE NO. 3, TABLE 1, ON DYNAMOMETER. DIRECT DRIVE THROUGH HEAVY THERMOID COUPLING

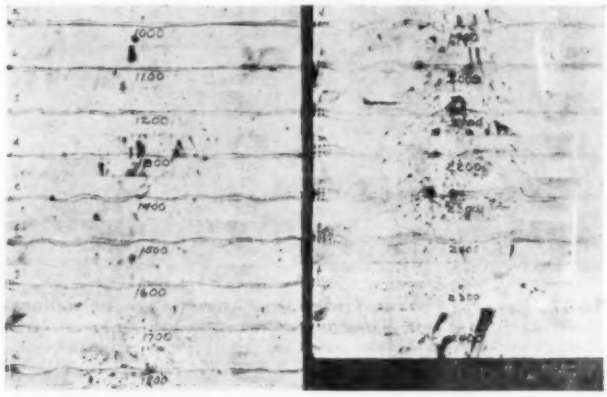


FIG. 15 ENGINE NO. 3, TABLE 1, ON PROPELLER LOAD, CRANKSHAFT CRACKED AT MIDDLE MAIN JOURNAL

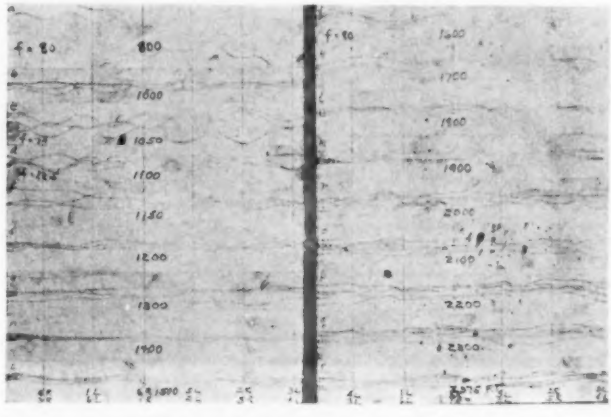


FIG. 16 SAME ENGINE AS IN FIGS. 14 AND 15, EXCEPT INCORPORATING 7-TO-5 REDUCTION GEARS AND A UNIQUE TYPE OF SPRING COUPLING. RECORDS TAKEN ON PROPELLER LOAD

tic curve, showing one node at a point close to the edge of the propeller hub. In this case, the torque on the most stressed section is nearly double that which, applied at the rear, would pro-

duce the same angular deflection as observed in operation. The large asynchronous deflections due to distribution may or may not follow this same type of elastic curve, depending on which cylinders receive the surges of pressure, or ramming effect of the manifold. The frequency of the latter effect in this engine, as indicated by vibrometer records, is from 12 to 21 per second. It is possible, therefore, that the actual deflection of cards (*a*) to

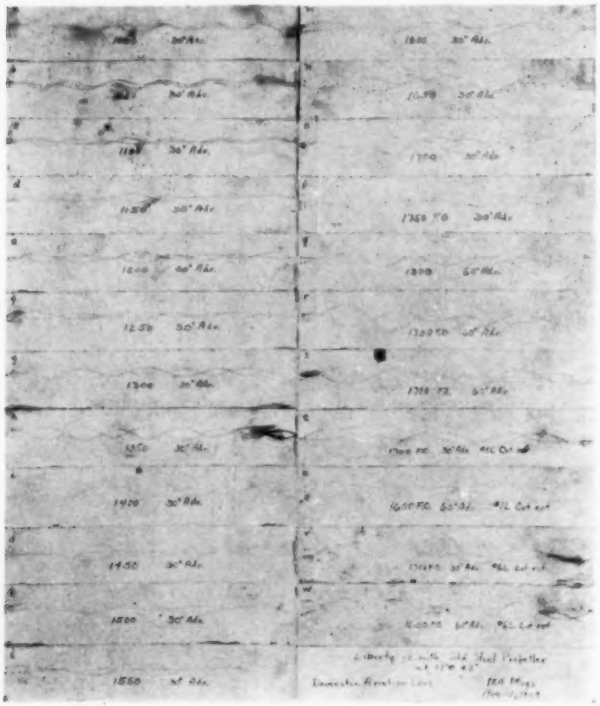


FIG. 17 EFFECT OF SPARK TIMING AND CUTTING OUT OF CYLINDERS ON TORSIONAL VIBRATION

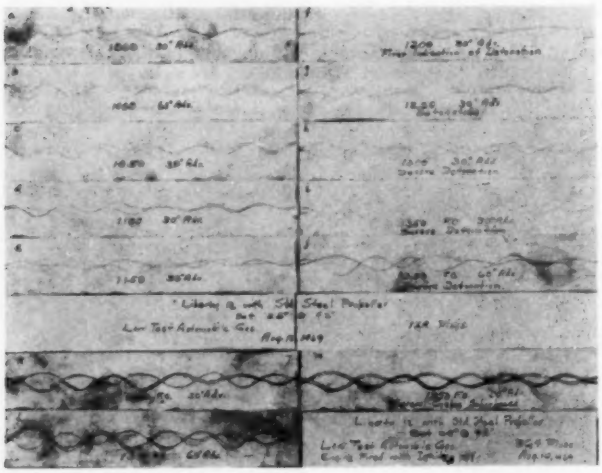


FIG. 18 EFFECT OF DETONATION AND PRE-IGNITION FROM HOT SPARK PLUGS ON TORSIONAL VIBRATION

(*k*), Fig. 11, were even greater than shown by the records, since the instrument frequency was about 10 vibrations per second.

In connection with this test, a second engine was available, which incorporated a reduction gear and a generally accepted type of spring coupling. It was also possible to replace some parts of this reduction gear with others, and run without any spring coupling. Fig. 13 cards (*a*) to (*m*) show the records obtained with the spring coupling, while (*n*) to (*z*) show those obtained with the solid-gear drive. These later records should bear some relation to those shown in Fig. 11, (*a*) to (*k*). The fuel was the same, and aside from the gearing, the engines were alike. Little can be said as to frequency of vibration with the spring coupling, except to again observe that all speeds from 700 to 1900 r.p.m. were about equally bad in respect to vibration, with no apparent tendency to damp out oscillations, as was observed in the case of the thermoid coupling, Fig. 7, cards (*m*) to (*x*). It appears that the natural frequency of the coupling is about 55 vibrations per second, although it could have a variety of frequencies, in view of its non-linear deflection characteristic. Cards (*n*) to (*z*) without the spring coupling show that much less vibration is found at all speeds. There appears to be some resonance at about 1800 r.p.m. at a frequency of about 90 vibrations per second and also at 1600 r.p.m. with frequency of about 99 vibrations per second. Cards (*c*), (*e*), (*f*), (*y*), and (*z*) show

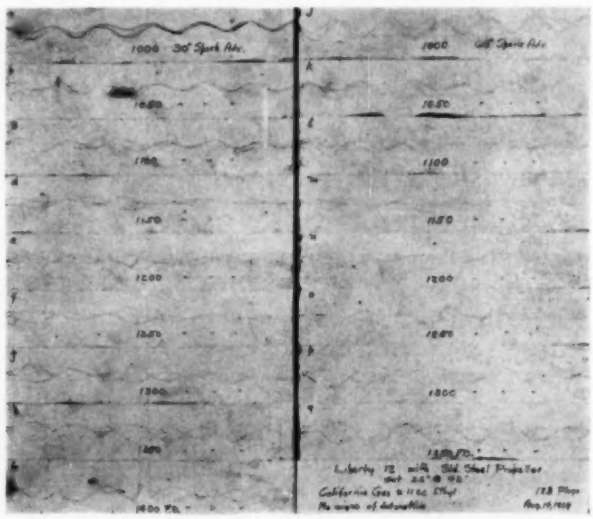


FIG. 19 EFFECT OF EARLY SPARK TIMING ON TORSIONAL VIBRATION, USING BEST FUEL AND COOLEST SPARK PLUGS AVAILABLE

a form of vibration which has been found in two Vee engines with poor distribution. Here, there are 6 vibrations per engine cycle, or 3 per revolution. There is no indication in the analysis by A. Stieglitz, of the normal torque curve of a 12-cylinder Vee engine to indicate resonance with a sixth harmonic, as would be indicated by these records. If, however, due to faulty or uneven distribution, alternate power impulses were abnormal, a vibration of this type would occur, much as is observed in a 6-cylinder line engine at resonance with the sixth harmonic. This form of vibration was observed on the air-cooled Liberty, which has a rotary induction system and plain gallery manifolds. There is no reason why the air-cooled Liberty should differ greatly from the water-cooled in character of vibration with the exception that pressure surges in the gallery type manifold are undoubtedly present, due to overlap of suction strokes. This form of vibration is not found on engines using four carburetor barrels, so that each barrel supplies three cylinders at 240-deg. intervals.

Fig. 14 is the record of another large 12-cylinder Vee engine. Here resonance occurs at about 1550 and 2000 r.p.m. On the basis of a frequency of 155 vibrations per second, resonance would occur at 1550, 2060, and 2660 r.p.m. This engine passed a 50-hour full-throttle test at 2400 r.p.m. but was found to have a cracked crankshaft at the conclusion of the test. Fig. 15 is a

set of records on the engine at conclusion of the test. The character of vibration is different from that observed with a perfect shaft, as might be expected. An attempt to run 50 hours at 2600 r.p.m. terminated at 12½ hours with a broken crankshaft.

Fig. 16 shows a set of records on the same engine as in Figs. 14 and 15, but with 7 to 5 reduction gears of the spur type. A spring coupling is incorporated in which coil springs are under an initial compression and no motion of the coupling occurs until a definite torque is exceeded. The rate of increase of torque with deflection is also different after the coupling starts to yield than before. The torsional rigidity of the propeller-crankshaft system is very greatly reduced, giving a natural frequency of only about

In Figs. 17 to 19 are presented some records taken on a Liberty-12 engine to study the effect of fuels, detonation, and pre-ignition on crankshaft torsional vibration. Fig. 17 shows the effect of changing the spark advance from 30 to 60 deg. This was undoubtedly accompanied by severe detonation, since the fuel was domestic aviation gasoline. Cards (*o*) and (*s*) serve to show the effect of premature ignition. Cards (*t*) to (*w*) show the effect of cutting out cylinder No. 1 or No. 6 combined with 30 deg. and 60 deg. advance. It appears that No. 6 (nearest the propeller) is the more serious loss, giving greater deflections than when cylinders near the rear end of the shaft are cut out. It was then decided to try the worst fuel obtainable—low-test automobile gaso-

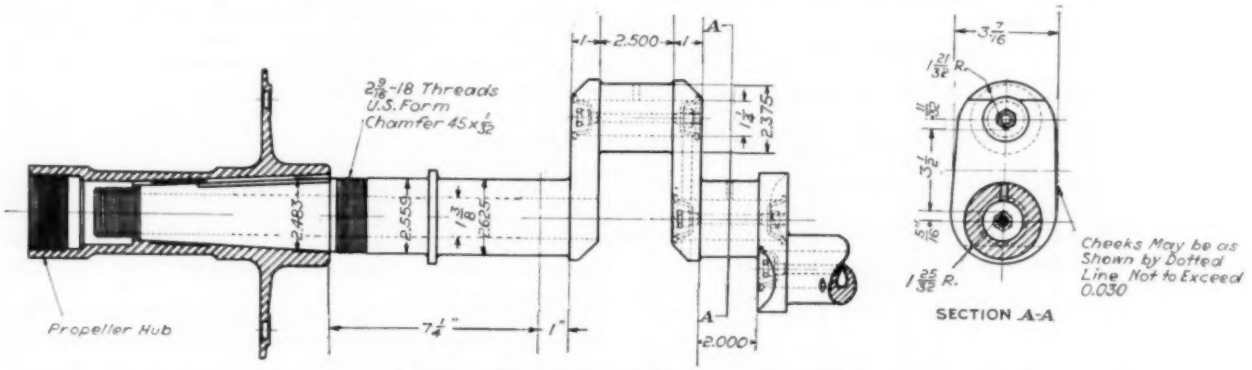


FIG. 20 DETAIL OF IN-LINE-ENGINE CRANKSHAFT

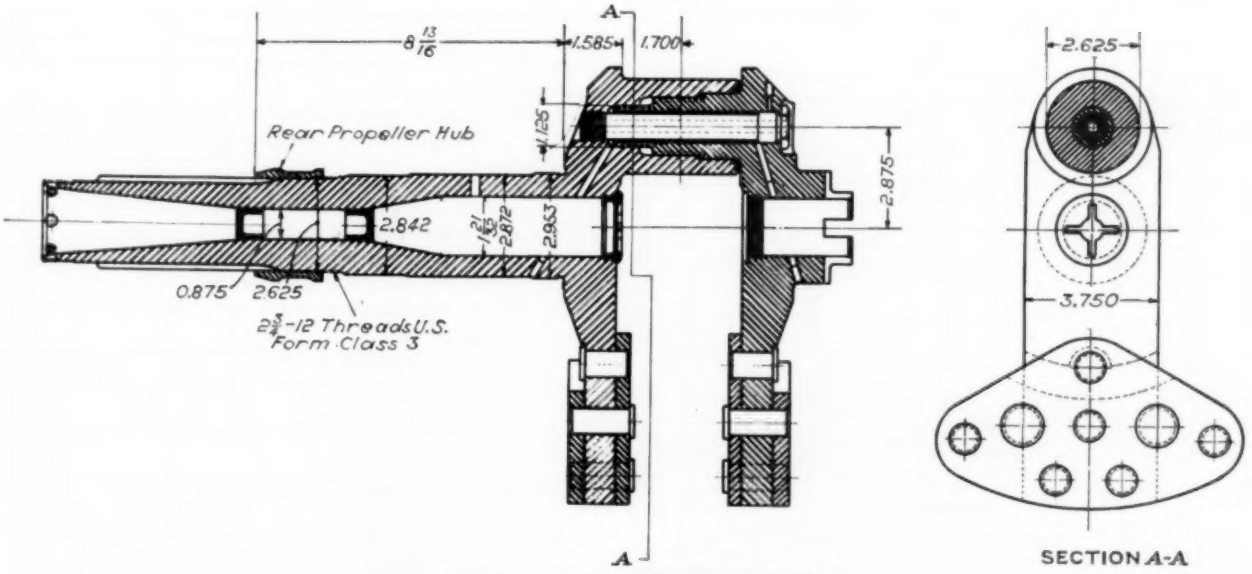


FIG. 21 DETAIL OF RADIAL-ENGINE CRANKSHAFT

80 vibrations per second. This frequency indicates the first torsional period at 800 r.p.m., the second at 1070 r.p.m., the third at 1370 r.p.m., and the fourth at 1600 r.p.m. The third period at 1370 r.p.m. does not appear in the records, but the fourth shows plainly at 1600 r.p.m. This period with three vibrations per revolution is not caused by distribution, since this engine is particularly good in this respect. It would indicate a sixth harmonic for a 12-cylinder Vee engine, contrary to the analysis previously referred to by A. Stieglitz and reproduced in Fig. 22*a*. This engine, however, has the articulated-rod construction whch may cause the torque curve from one bank to differ from the other.

line such as is used for washing purposes. To make matters as severe as possible, the plugs used were the hottest obtainable and cards were then taken at 30-deg. and 60-deg. spark advance. The detonation must have been terrific as evidenced by dense black puffs from the stacks. It is interesting to note, however, that no one was able to hear detonation above the other noises present. At another time, silencers were fitted to a Liberty in the effort to make detonation audible, but without success. These silencers were effective enough to make unnecessary the use of ear defenders and the fuel was low-test gasoline. However, no one could be sure he could detect detonation audibly. When a Liberty single is running, exhausting into an exhaust duct, deto-

nation may be heard for half a mile. In Fig. 18 are shown several cards taken under various conditions. These serve to show that premature ignition does increase the amplitude of torsional vibration, while detonation at normal spark advance does not affect torsional vibration. In case, however, of extended periods of detonation, hot spots may appear and cause pre-ignition, and thus indirectly cause increased vibration. Fig. 19 shows the best conditions obtainable. California gasoline was used with 11 cc. of ethyl fluid and cool plugs were installed. There were no visible signs of detonation, even with 60 deg. spark advance. The surprising feature of this run is that the amplitude of torsional vibration is greater for a given spark setting and power output than when using low-test gasoline with pre-igniting spark plugs and extreme detonation.

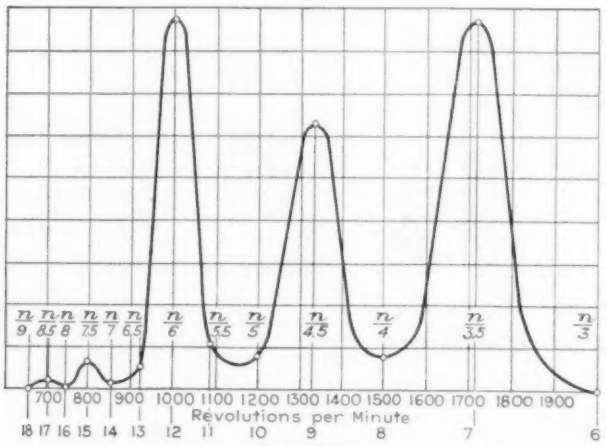


FIG. 22a

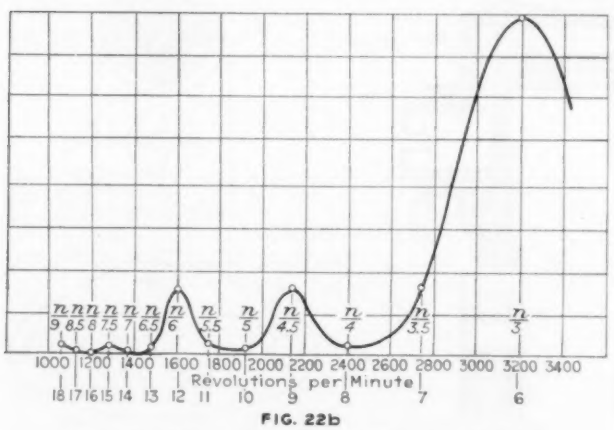


FIG. 22b

FIG. 22 HARMONICS OF SINGLE-CYLINDER TORQUE

(Fig. 22*a* Comparative resonance deflections—12-cylinder Vee, $f = 100$ vibrations per second, $n = 60f$; Fig. 22*b* Comparative resonance deflections —6-cylinder line, $f = 160$ vibrations per second, $n = 60f$.)

Figs. 22*a* and 22*b* show comparative resonance deflections for 12-cylinder Vee and 6-cylinder line engines. Fig. 22*a* is calculated on a frequency $n = 6000$, as found in the Liberty-12. These periods check with great accuracy the observed periods as indicated in Figs. 4 and 5.

Fig. 22*b* shows the 6-cylinder resonance curve on the basis of $n = 9600$ as found in the engine which gave the records shown in Fig. 7. Unfortunately, only one of the periods predicted in Fig. 22*b* could be observed.

These curves are reproduced from a paper on "Torsional Vibrations in Vertical Engines," by Albert Stieglitz, published in *Luftfahrtforschung*, July 24, 1929.

METHOD OF CALCULATING FREQUENCY OF TORSIONAL VIBRATION

Several methods of attacking the problem of torsional vibration have been proposed and two of these in particular have given excellent results. B. C. Carter, in *Engineering*, July 13, 1928, has given an empirical formula based on numerous experiments. In his method the crank is reduced to a torsionally equivalent length of shaft of the same cross-section as the main journal. S. Timoshenko in his book, "Vibration Problems in Engineering," has given a rational method of reducing all cranks to a length of shaft having one torsional rigidity. Once having the crank replaced by a uniform shaft, and the inertia masses reduced to an equivalent flywheel system, the frequency of vibration is readily calculated. In aircraft work, the moment of inertia of the propeller is from 10 to 20 times the total equivalent moment of inertia of the engine. It is therefore permissible to consider the nodal point as being located at the edge of the propeller hub. It could hardly be taken at the center of the hub, since the hub reinforces the shaft end and prevents it from deflecting. In the case of the radial engine, the equivalent moment of inertia of a given number of cylinders is approximately twice that of the line engines, because of the necessary balance weights. However, it is found that little error is introduced in considering the propeller moment of inertia as infinite, that is, assuming a node at the edge of the propeller hub.

Professor Timoshenko has stated in his book, "Vibration Problems in Engineering," that little error is introduced in calculating the primary or fundamental frequency, if all the equivalent flywheel masses at the crankpins are considered as adding together at the longitudinal center of symmetry of the engine. The problem is therefore reduced to that of finding the torsionally equivalent length of shaft from the edge of the propeller hub to the center of symmetry of the engine, and the equivalent moment of inertia of the crankpin systems.

A typical aircraft-engine crankshaft is shown in Fig. 20. The notation used is given below substantially as used by Professor Timoshenko.

a = actual length of crankpin, in.
a_1 = effective length of crankpin $= a + 0.9h$ in.
$2b$ = actual length of main journal, in.
$2b_1$ = effective length of main journal, $= 2b + 0.9h$ in.
B_2 = flexural rigidity of crankpin $= I_2E = 1.474 \times 10^6 (D_2^4 - d_2^4)$
B_3 = flexural rigidity of crankcheek $= I_3E = 2.5 \times 10^6 hw^3$
C_1 = torsional rigidity of main journal $= J_1G = 1.178 \times 10^6 (D_1^4 - d_1^4)$
C_2 = torsional rigidity of crankpin $= J_2G = 1.178 \times 10^6 (D_2^4 - d_2^4)$
C_3 = torsional rigidity of crankcheek $= J_3G = 3.33 \times 10^6 w^3h^3/(w^2 + h^2)$
D_1 = outside diameter of main journal, in.
d_1 = inside diameter of main journal, in.
D_2 = outside diameter of crankpin, in.
d_2 = inside diameter of crankpin, in.
E = Young's modulus of elasticity for steel $= 30 \times 10^6$ lb./in.^{2}
F_2 = cross-sectional area of crankpin $= \frac{\pi}{4}(D_2^2 - d_2^2)$
F_3 = cross-sectional area of crankcheek $= hw$
G = modulus of elasticity in shear or torsion $= 12 \times 10^6$ lb./in.^{2}
h = thickness of crankcheek, in.
I_1 = moment of inertia of main journal section $= \frac{\pi}{64}(D_1^4 - d_1^4)$ in.^{4}

I_2 = moment of inertia of crankpin section = $\frac{\pi}{64}(D_2^4 - d_2^4)$ in.^{4}

I_3 = moment of inertia of crankcheek section = $hw^3/12$ in.^{4}

J_1 = polar moment of inertia of main journal section = $2I_1$ in.^{4}

J_2 = polar moment of inertia of crankpin section = $2I_2$ in.^{4}

J_3 = polar moment of inertia of crankcheek section = $w^3h^3/3.6(w^2 + h^2)$ in.^{4}

K = factor of complete constraint by main journals (Timoshenko) =

$$\frac{\frac{R(a+h)^2}{4C_3} + \frac{aR^2}{2C_2} + \frac{a^3}{24B_2} + \frac{R^3}{3B_3} + \frac{1.2}{12 \times 10^6}\left(\frac{a}{2F_2} + \frac{R}{F_3}\right)}{\frac{aR}{2C_2} + \frac{R^2}{2B_3}} \text{ in.}$$

L = length of connecting rod, in.

l_1 = equivalent length of shaft from edge of propeller to first crank throw minus b

l_2 = equivalent length of shaft, center to center of main journals if shaft is unconstrained = $2b_1 + a_1 \frac{C_1}{C_2} + 2R\frac{C_1}{B_3}$ in.

l_3 = equivalent length of shaft, center to center of main journals if shaft is constrained by main bearings = $2b_1 + a_1\frac{C_1}{C_2}\left(1 - \frac{R}{K}\right) + 2R\frac{C_1}{B_3}\left(1 - \frac{R}{2K}\right)$ in.

L_f = $l_1 + \frac{n}{2}l_2$ (unconstrained) in.

L_c = $l_1 + \frac{n}{2}l_3$ (constrained) in.

m_1 = rotating weight per crankpin, lb.

m_2 = reciprocating weight per crankpin, lb.

n = number of crank throws

R = crank radius, in.

w = width of crankcheek, in.

The equivalent moment of inertia of one crank system is, according to Professor Timoshenko:

$$i = \left[m_1 + \frac{m_2}{2}\left(1 + \frac{R^2}{4L^2}\right)\right]R^2 + i_c$$

Since the average ratio of crank to connecting rod is about 1/4, this value of R/L may be used, making the term $\frac{1}{2}\left(1 + \frac{R^2}{4L^2}\right)$ equal to 0.508. This may be used for all practical purposes, hence we may take for the whole engine

$$I = n[(m_1 + 0.51m_2)R^2 + i_c] \text{ (lb-in.}^2\text{)} \ldots\ldots [1]$$

in which i_c is the calculated or experimentally determined value of the moment of inertia of one crank, about the main journal center, lb-in.^{2}

The frequency of oscillation of a torsion pendulum is $f = \frac{1}{2\pi}\sqrt{\frac{C}{I}}$ in which C is the torque in pound-feet per radian, required to deflect the torsion member, and I is the moment of inertia of the pendulum in slug-ft.^{2} In design calculations, it is convenient to use C_1 in pound-inches per radian per inch length and I in lb-in.^{2}, in which case $C = \frac{C_1}{L}$ and $f = \frac{1}{2\pi}\sqrt{\frac{12 \times 32.2 \times C_1}{IL}}$ or $3.125\sqrt{\frac{C}{IL}}$

For the unconstrained case, $f_f = 3.125\sqrt{\frac{C_1}{IL_f}}$ . . . . . [2]

For complete constraint, $f_c = 3.125\sqrt{\frac{C_1}{IL_c}}$ . . . . . [3]

Major Carter's method, see *Engineering*, July 13, 1928, is purely an empirical one, and is given only for the case of the shaft in its bearings. The equivalent length of shaft from center to center of main journals is given as

$$l_3 = (2b + 0.8h) + \frac{3}{4}\frac{D_1^4 - d_1^4}{D_2^4 - d_2^4}a + \frac{3R}{2}\frac{D_1^4 - d_1^4}{hw^3} \ldots\ldots [4]$$

in which the symbols have the same meaning as before

The above methods will now be applied to the Liberty-12 engine, where

a = 2.5 in.

a_1 = 2.5 + 0.9 × 1 = 3.4 in.

$2b$ = 2 in.

$2b_1$ = 2 + 0.9 × 1 = 2.9 in.

B_2 = $1.474 \times 10^6(2.375^4 - 1.25^4) = 43.1 \times 10^6$

B_3 = $2.5 \times 10^6 \times 1 \times 3.4375^3 = 101.2 \times 10^6$

C_1 = $1.178 \times 10^6(2.625^4 - 1.375^4) = 51.8 \times 10^6$

C_2 = $1.178 \times 10^6(2.375^4 - 1.25^4) = 34.5 \times 10^6$

C_3 = $3.33 \times 10^6 \times 3.4375^3 \times 1^3/(3.4375^2 + 1^2) = 10.6 \times 10^6$

D_1 = 2.625 in.

d_1 = 1.375 in.

D_2 = 2.375 in.

d_2 = 1.25 in.

F_2 = $\frac{\pi}{4}(2.375^2 - 1.25^2) = 3.2$ sq. in.

F_3 = 1 × 3.4375 = 3.4375 sq. in.

h = 1 in.

w = 3.4375 in.

m_1 = 6.3 lb.

m_2 = 12.4 lb.

i_c = 92.5 lb-in.^{2}

n = 6

R = 3.5 in.

$$K = \frac{\frac{3.5(2.5+1)^2}{4 \times 10.6 \times 10^6} + \frac{2.5 \times 3.5^2}{2 \times 34.5 \times 10^6} + \frac{2.5^3}{24 \times 43.1 \times 10^6} + \frac{3.5^3}{3 \times 101.2 \times 10^6} + \frac{1.2}{12 \times 10^6}\left(\frac{2.5}{2 \times 3.2} + \frac{3.5}{3.44}\right)}{\frac{2.5 \times 3.5}{2 \times 34.5 \times 10^6} + \frac{3.5^2}{2 \times 101.2 \times 10^6}} = 9.35 \text{ in.}$$

l_1 = 7.25 in. (actual) or 7.6 in. (equivalent length)

$$l_2 = 2.9 + 3.4\frac{51.8 \times 10^6}{34.5 \times 10^6} + 2 \times 3.5\frac{51.8 \times 10^6}{101.2 \times 10^6} = 11.58 \text{ in.}$$

$$l_3 = 2.9 + 3.4\,\frac{51.8 \times 10^6}{34.5 \times 10^6}\left(1 - \frac{3.5}{9.35}\right) + 2 \times 3.5\,\frac{51.8 \times 10^6}{101.2 \times 10^6}\left(1 - \frac{3.5}{2 \times 9.35}\right) = 9 \text{ in.}$$

$L_f = 7.6 + 3 \times 11.58 = 42.36$ in.
$L_c = 7.6 + 3 \times 9 = 34.6$ in.
$I = 6\,[(6.1 + 0.53 \times 12.4)\,3.5^2 + 92.5] = 1482$ lb-in.^{2}

$$f_f = 3.125\sqrt{\frac{51.8 \times 10^6}{1482 \times 42.36}} = 90 \text{ vib. per sec.}$$

$$f_c = 3.125\sqrt{\frac{51.8 \times 10^6}{1482 \times 34.6}} = 99.5 \text{ vib. per sec.}$$

According to Major Carter,

$$l_3 = (2 + .8 \times 1) + \frac{3}{4}\,\frac{2.625^4 - 1.375^4}{2.375^4 - 1.25^4}\,2.5 + \frac{3 \times 3.5}{2}\,\frac{2.625^4 - 1.375^4}{1 \times 3.4375^3} = 11.53 \text{ in.}$$

$L = 7.6 + 3 \times 11.53 = 42.19$ in.

$$f = 3.125\sqrt{\frac{51.8 \times 10^6}{1482 \times 42.19}} = 90 \text{ vib. per sec.}$$

The observed value is about 100 vib. per sec. as shown by the records, Figs. 4 and 5.

Table 1 shows the calculated and observed frequencies for several engines.

TABLE 1

Engine	Timoshenko: Unconstrained	Timoshenko: Constrained	Carter	Observed: Freq.	Observed: Crit. R.p.m.		
(1) Liberty 12.5-in. × 7-in. Vee	90	100	90	100	1000	1330	1715
(2) 12-cyl. 4½-in. × 6-in. Vee	147.5	161	152	160	1600	2130	2740
(3) 12-cyl. 5⅛-in. × 6¼-in. Vee	139.3	153	149.5	155	1550	2070	2660
(4) 6-cyl. 4½-in. × 5½-in. Line	151	161	155	160	1600	2140	3200
(5) OX5, 8-cyl. Vee	132	135.5	133.5	Not obs.	2036	(Not observed)	
(6) 12-cyl. 6⅜-in. × 6½-in. Vee	129	137	138	135	1350	1800	2320
(7) 12-cyl. 6⅜-in. × 6½-in. Vee (Geared)	70.5	71.8	71.8	Ext. vib.	700 to 1900		

In applying the foregoing calculations to radial engines, it is found that Professor Timoshenko's unconstrained case applies with sufficient accuracy. This is probably because in the usual radial engine designs the after crank throw is supported by only one bearing and the shaft has a certain degree of freedom. The equivalent shaft length is calculated only to the center line of the engine, hence the after crankcheek need not be considered. The crankshaft is counterbalanced in all single-row radial engines, hence its moment of inertia may readily be found by treating it as a compound pendulum, the only data necessary being the crank radius, reaction of counterbalances at crank radius and time of one swing right to left about the main journal axis. The moment of inertia of the shaft is then

$$i_c = W_c R t^2\,\frac{144g}{12\pi_2} = 39.2\,W_c R t^2 \text{ lb-in.}^2 \quad \ldots\ldots\ldots [5]$$

where W_c = weight of counterweights at crank radius, lb.
R = crank radius, in.
t = time of ½ vibration (one swing), sec.

No appreciable error is introduced in thus including the shaft end which projects into the hub.

$a^1/2 = 1.78 + 0.45 \times 1.475 = 2.44$ in.
$b = 0.4375$ in.
$b_1 = 0.438 + 0.45 \times 1.475 = 1.098$ in.
$B_3 = 2.5 \times 10^6 \times 1.475 \times 3.75^3 = 194.3 \times 10^6$
$C_1 = 1.178 \times 10^6(2.874^4 - 1.656^4) = 71.7 \times 10^6$
$C_2 = 1.178 \times 10^6(2.685^4 - 1.125^4) = 59.1 \times 10^6$
$D_1 = 2.874$ in.
$d_1 = 1.656$ in.
$D_2 = 2.685$ in.
$d_2 = 1.125$ in.
$h = 1.475$ in.
$w = 3.75$ in.
m_1 (rotating weight of master and link rods) = 19.8 lb.
m_2 (reciprocating weight of pistons and rods) = 56.55 lb.
W_c = weight of counter weights at crank radius = 48.27 lb.
$R = 2.875$ in.
t = time of one swing of crankshaft = 0.517 sec.
$i_c = 39.2 \times 48.27 \times 2.875 \times 0.517^2 = 1460$ lb-in.^{2}
$i_p = (19.8 + 0.51 \times 56.55)\,2.875^2 = 402$ lb-in.^{2}
$I = i_c + i_p = 1862$ lb-in.^{2}
l_1 (actual, since torsional rigidity approximately uniform) = 8.4 in.

$$L_f = l_1 + b_1 + \frac{a_1}{2}\,\frac{C_1}{C_2} + R\,\frac{C_1}{B_3} = 8.4 + 1.098 + 2.44\,\frac{71.7 \times 10^6}{59.1 \times 10^6} + 2.875\,\frac{71.7 \times 10^6}{194.3 \times 10^6} = 13.56 \text{ in.}$$

The equivalent moment of inertia of the rotating and reciprocating parts is, as in the line engine

$$i_p = (m_1 + 0.51\,m_2)R^2, \text{ lb-in.}^2 \quad \ldots\ldots\ldots [6]$$

Fig. 21 shows the crankshaft of a very successful 9-cylinder radial engine of 1340 cu. in. displacement, in which

$a/2$ = (center of engine to crankcheek) = 1.78 in.

$$f_f = 3.125\sqrt{\frac{71.7 \times 10^6}{1862 \times 13.56}} = 166 \text{ vib. per sec.}$$

The critical speed, therefore, will be for nine cylinders,

$$N = \frac{166 \times 60}{4.5} = 2210 \text{ r.p.m.}$$

Fig. 10, (*a*) to (*n*), shows the torsiometer records on this engine at throttling loads from 2400 r.p.m.; also full throttle from 1875 r.p.m. to 2375 r.p.m.

Fig. 10, (*aa*) to (*nn*), shows the records of a large 9-cylinder radial engine of 1860 cu. in. displacement on propeller load. The calculated frequency is 180 vibrations per second and the critical speed

$$N = \frac{180 \times 60}{4.5} = 2400 \text{ r.p.m.}$$

A similar calculation on a small 5-cylinder engine of 370 cu. in. displacement shows the frequency to be 156 vibrations per second and critical speed

$$N = \frac{156 \times 60}{2.5} = 3750 \text{ r.p.m.}$$

No test records are available on this engine.

Fig. 3 (*a*) and (*b*) show torsiometer records of the nose end of a Liberty-12, using two types of propeller, metal, and wood. A small displacement undoubtedly occurs, but due to the relatively large moment of inertia, it is insignificant; hence in aircraft work it appears sufficiently precise to record only the motion of the rear end of the crankshaft. These records also serve to show the desirability of driving accessories, and particularly superchargers, form the propeller end of the shaft where torsional vibrations are of low amplitude. Slipping clutches, spring couplings, etc., could then be omitted, making a saving in weight. Ignition and valve timing would also be more uniform, resulting in smoother operation of the engine.

Card (*c*), Fig. 3, shows a method of arriving at the accelerations which occur in actual accessory drives, using the conventional rear-end drive.

From the foregoing it is seen that calculation of torsional periods from design data is sufficiently precise to make such a study worth while in any new design as well as in existing engines. Data are not available on all types of engines, but the leading types have been covered by experiment so that the character of vibration to be expected is fairly well established. It is the aim of the Power Plant Branch to continue this study until all types of engines and reduction gears have been analyzed.

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Discussion

J. P. DenHartog.[2] Vibration troubles occurring in the rotating parts of machinery can be grouped in two distinct classes, (1) torsional and (2) flexural or whirling vibrations. In most types of machines the whirling vibration causes most grief while the torsional type is comparatively unimportant. Exceptions to this are the gas and Diesel engines, where due to the great non-uniformity of torque the torsional vibrations are more important than the flexural.

The present state of the art of predicting by calculation the behavior of a machine in this respect is in quite different stages of development for the two types of vibration. As regards the determination of the frequency or the "critical speeds," this can be done in the torsional case with an error of 5 per cent or less. In the flexural case, however, the frequency depends on the flexibility of the bearings and the frame, so that in a complicated machine errors of over 50 per cent are not uncommon.

Determination of the amplitude of vibration at a critical speed is possible in the torsional case with an accuracy, making such calculations practically useful. The methods for this have been worked out by Holzer, Wydler, and Lewis, and are described in S. Timoshenko's book mentioned in the paper. For the flexural case, however, a determination of amplitudes is as yet not possible.

The present paper gives a rather rough method of precalculating the frequency of torsional vibrations. The author reduces an engine with many cranks and pistons to a system with a single equivalent inertia in the center of the engine. This introduces an inaccuracy which in practical cases runs up as high as 10 per cent.

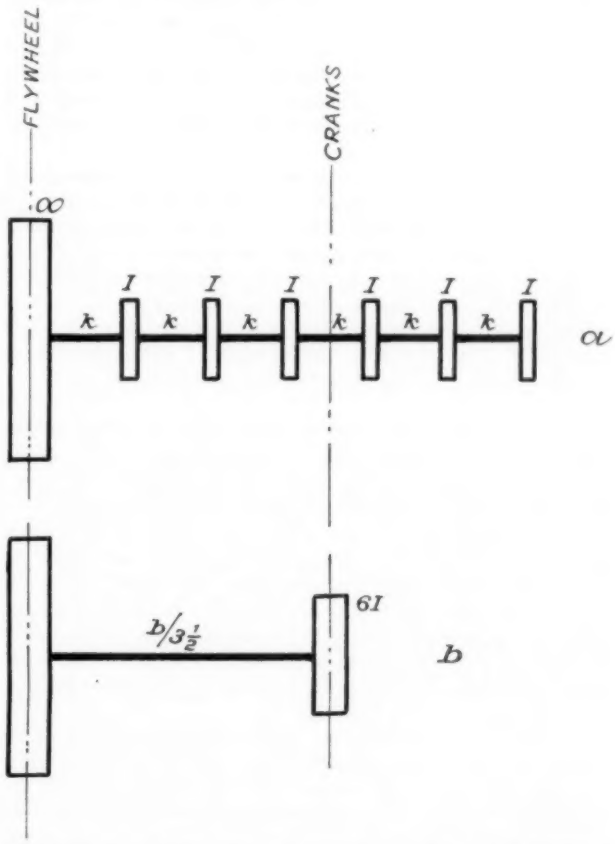


Fig. 23 System With Heavy Flywheel and Six Equal Inertias

For instance consider a system consisting of a "heavy" flywheel, and six equal inertias I connected by six pieces of shaft all of the flexibility k lb-in. per radian. (Fig. 23*a*.)

The exact calculation (for instance by means of Holzer's method[3]) gives:

$$f = \frac{1}{2\pi}\sqrt{0.0580\,\frac{k}{I}}$$

while the author's approximation as shown in Fig. 23*b* gives

[2] Research Laboratories, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

[3] "Vibration Problems in Engineering, by S. Timoshenko, pp. 150 and 151.

$$f = \frac{1}{2\pi}\sqrt{0.0476\,\frac{k}{I}}$$

being about 10 per cent smaller. In case a coupling of great torsional flexibility is inserted between the engine and the flywheel the error becomes less. It is seen in Table 1 of the paper that the observed frequencies are between zero and 10 per cent greater than the ones predicted by calculation by means of Timshenko's unconstrained or Carter's formulas. This discrepancy is due to the lumping of all masses into a single one. It again can be regarded as a confirmation of the experience that the effect of constraint of the main bearings does not play an important part.

In the paper the author states that certain of his test results seem to contradict the analysis by Stieglitz shown in Fig. 22a. However, it seems that the engine tested is not exactly like the one analyzed, and since the amplitude in the third order vibration ($n = 3$) for a 12-cylinder engine depends on the crank-shaft structure and on the firing order of the cylinders, no comparison is justified.

The author does not give any calculations of the amplitudes of vibration at the critical speeds. It would be of considerable interest to do this yet and to compare the calculated results to the observed ones. In this manner the accuracy of calculations of this type can be improved. In cases where thermoid couplings are used, first a test has to be run on the coupling by itself to determine the energy absorption as a function of the amplitude and frequency. With this data a calculation of the amplitudes of torsional vibration at the critical speeds can be carried out.

Melvin Price.[4] The author shows some vibration records taken from a crankshaft with a reduction gear with a spring coupling interposed between them. In his comment he applies the adjective "bad" somewhat, it seems, to the disparagement of the spring coupling. The use of the word "bad" in this connection seems a trifle harsh. The spring coupling is one of the devices often used to protect toothed gears from shock. This use seems a legitimate one. The investigations undertaken by the author are of particular value in the detection and avoidance of resonance. There is nothing in the records to show that resonance is produced by the spring coupling or, if initially present, that it is aggravated by the coupling.

In other records a relatively large torsional deflection is attributed to the Farman reduction gear used in one of the tests. The large decrease in the critical speed as a consequence of the insertion of this gear is one of the surprising results of his investigation with the torsiometer. While not as "springy" as the spring coupling the effect is much the same in making all speeds more or less "equally bad." Yet in such theoretical treatments the geared systems are treated as though the gears were rigid. The Farman assembly shows the possibility of considerable elastic distortion on account of the variety of shape and the arrangement of the members connecting the power with the work.

The experimental determination of torsional deflection in toothed gearing brings to mind the other machine elements in a drive-shaft assembly and makes one reflect on how best to estimate the torsional behavior of such things as bolted, flanged hubs—whether in ring-gear assemblies or flange couplings—and also of other types of shaft couplings, especially where hollow shafts are used, and these are common in aircraft. This at once raises the question of the efficacy of the "equivalent-length" method of locating the node points in the determination of critical speeds. It would seem to revert to the old question of what is easiest to judge or to estimate in the furtherance of a solution of a vibration problem. Professor F. M. Lewis prefers to use, instead of "equivalent lengths," a "rigidity factor" for each element interposed between the power and the work. It seems that this simplifies the solution somewhat, and makes it possible to apply the judgment to the rigidity factor with somewhat greater accuracy than is possible in the "equivalent length" analysis.

W. Klemperer.[5] Have experiments with the torsiograph also revealed any two-noded vibrations, that is to say, with one node in the crankshaft in addition to the one node next to the hub, or are the more-noded modes in heavier-than-aircraft engines of too high a frequency to be causing any trouble? This question is, of course, mainly of interest in connection with large 12-cylinder V-type engines.

Roland Chilton.[6] The author is to be congratulated on the development of a satisfactory torsional-vibration recorder for aircraft engines and for the excellent presentation of the results obtained. A glance at the curves shows that the resonance occurs in most engines at two, three or four different speeds due to different harmonics from the explosion impulses, and it appears to the writer that a very serious limitation in engine usefulness is involved in the attempt to keep the operating-speed range of the engine away from all of the possible zones of resonance.

Certainly, in specific cases, the limitation to increased engine speed is definitely that a period of resonance will be encountered and the redesign of the crankshaft to lift the period by only 200 r.p.m. necessitates enlarging the journals, including the connecting rod, which in turn may require a larger crankcase. The result will often be that the next lower period will be sufficiently raised to be encountered at cruising speed.

One of the most interesting points brought out in the paper is illustrated in Fig. 7 comparing cards *a* to *l*, with a rigidly attached propeller, with cards *m* to *x* taken with a thermoid coupling driving to a dynamometer. The writer has previously suggested that a coupling with appreciable hysteresis will suppress torsional vibrations, and it is hard to see why its usefulness would be limited to a case of a geared engine.

The automobile industry several years ago realized that many of its troubles were due to torsional vibrations and for a time most manufacturers tried to cure this condition by making the crankshafts heavier and heavier. Finally they were forced to the use of a damper which eliminates all periods at all speeds instead of merely locating the several synchronous points at somewhat higher speeds. Aircraft-engine builders have probably suffered more from failed crankshafts and crankshaft gears than any others, but they seem slow to take advantage of the experience already gained in other lines. The automobile designers took precisely the same attitude several years ago, but have now practically all been forced to some form of damper, of which there are now several different types giving satisfactory results.

When it is seen that a typical engine has vibration periods at 1550, 2060, and 2600 r.p.m., it is not obvious how the crankshaft stiffness can be juggled to bring all the new periods outside of the operating range. If the engine is already in production, the changes would be of an extremely far-reaching nature compared to which the installation of one of the several types of dampers already developed for automobiles would be relatively simple.

In face of the fact that the mere inclusion of the simple thermoid disc suppressed high-amplitude vibrations at all speeds, it seems that some such device should be investigated immediately in an aircraft-engine installation. It is not thought that the cou-

[4] Research Engineer, Allison Engineering Company, Indianapolis, Ind. Mem. A.S.M.E.

[5] Manager, Research Department, Goodyear-Zeppelin Corp., Akron, Ohio.

[6] Consulting Engineer, Wright Aeronautical Corp., Paterson, N. J.

pling will have to be larger than the spinners now in use. It should be noted that the Packard Diesel engine is already using rubber blocks to drive the propeller.

Author's Closure

Mr. DenHartog brought up the question of the firing order of line engines. All the line engines with 6 or 12 cylinders have only two firing orders which are practicable from the standpoint of distribution and smoothness. These are 1, 5, 3, 6, 2, 4 and 1, 4, 2, 6, 3, 5. There are other possible firing orders, but with the manifold system arranged with cylinders 1, 2, 3 and 4, 5, 6 of each bank fed by separate venturii, the other arrangements do not give uniform intervals between suction strokes on each carburetor, thus interfering with distribution. It is possible that another firing order could be used, which would eliminate one of the critical periods, but in aircraft work, odd-firing intervals, and balance masses would be objectionable from the standpoint of magneto ignition, which requires uniform firing intervals, and from the standpoint of added weight, were balance weights required.

It should be possible to calculate the "damping factor" for several engines, from torsional vibration records. If the damping were known, the amplitude of vibration could be calculated from the torque curve of the engine.

Mr. DenHartog called attention to the possibility that card *f*, Fig. 11, could be explained by the fact that the frequency was roughly 10 times the fundamental of the instrument. However, the same objection should apply to card *p* taken at the same speed, but with better fuel. There was no difference in the two runs other than the use of highly volatile fuel. Hence, it would seem hard to explain the difference, except on the basis of distribution.

It is realized that the approximate solution of grouping all the crank inertia masses at the center of symmetry of the engine is not exact, but the complete solution by the method of determinants complicates the calculations considerably. The close agreement between the constrained case (Timoshenko) and the observed frequencies, shown in Table 1, is gratifying, to say the least. Calculations show that the stiffer the crankpin and the shorter and stiffer the crank cheeks, the nearer are the constrained and unconstrained cases to each other. This may partially explain why, in Diesel work, little evidence is observed of constraint by main bearings. In most aircraft engines, the crankpins are not as stiff as the main journals, and the assumption of constraint is probably not far from the truth.

Professor Price objected to calling the records of Fig. 13(*a*) to (*m*) "bad." Let us call them equally good, but the fact remains that cards *n* to *z* are quite definitely better. It would appear that spring couplings need damping and freedom from backlash to make them entirely satisfactory. It has been found by deflection tests that some of the flexibility, in reduction gears, is in the gear case, although there is, of course, actual deflection of the gears themselves. Where drive shafts are used, it should be possible closely to approximate the torque per radian for the entire drive from propeller to center of the engine. The fundamental frequency of the crankshaft will then not enter into the calculations, since the entire drive will have a lower frequency than that of the crankshaft itself. In complicated drives, it would be necessary to deflect the system, and thus obtain the stiffness for calculation of frequency. The Matériel Division has not experimented with any drives other than propeller reduction gears with and without spring couplings, but in automobile drives the period of the propeller shaft bears no relation to that of the crankshaft.

Doctor Klemperer inquired as to evidence of higher frequencies than the primary. No evidence has yet been found of vibration with two nodes. It may be that such vibration will be encountered as engine speeds are increased, but to date the Matériel Division has no engines capable of operating at speeds high enough to make possible such higher frequencies. The Matériel Division's torsiometer has not been applied to the study of vibration in long shafts. Some form of belt drive would have to be resorted to, and such drive must be viewed with suspicion. Doctor Thoma's electrical device should be well adapted to such use.

Mr. Chilton has brought up a very interesting field for investigation. There is, however, some question as to the effectiveness of a damper, and there is the question of possible failure of the damping device to function. Is it not true also that a damper, while suppressing torsional vibration at one frequency, introduces another frequency at which vibration may occur? A case in point is a recently developed fluid friction damper of European make. Curves were shown, representing the amplitude of vibration versus speed in r.p.m., and were made on two 6-cylinder line engines. The two critical ranges due to the twelfth and ninth harmonics appeared at 980 r.p.m. and 1290 r.p.m., but the third, and by far the most serious, would be that due to the sixth harmonic, occurring at 1960 r.p.m. The latter period was, however, out of the speed range since the normal speed of these engines was about 1500 r.p.m. This would mean that the engines would cruise at about 1300 r.p.m., or in the second critical range. By the use of the fluid-friction damper, the critical ranges were lowered to about 850 r.p.m. and 1170 r.p.m., the third occurring at 1700 r.p.m. This arrangement undoubtedly improved conditions in the cruising range, but brought the third critical r.p.m. uncomfortably close to the normal speed of the engine, as evidenced by increased amplitude of vibration at 1500 r.p.m. The test curve of the engine, with and without damper, is reproduced in Fig. 24. It would appear that a better solution in this case would have been to raise the frequency of the crankshaft by design, from 98 vibrations per sec. to 125 or 130 vibrations per sec. Critical ranges then would have been 1200 to 1300 r.p.m., 1600 to 2730 r.p.m., and 2400 to 2600 r.p.m. This would not introduce a crankshaft of unduly large dimensions since there are 12-cylinder Vee engines whose crankshafts have frequencies above 160 vibrations per sec. The larger and stiffer shaft greatly decreases the alternating stress due to vibration and also the stress due to torque. A damper then becomes unnecessary, since an engine can operate continuously while in critical vibration, provided the combined torsional and alternating stress is low enough. This is evidenced by a recent test on the radial engine whose torsiometer records are reproduced in Fig. 8 of the paper. One of these engines passed 100 hours at 1900 r.p.m. and another 150 hours at the same speed without any indication of crankshaft failure. The first 50 hours in each case was at full throttle. The author did not believe these engines would go 25 hours at full throttle.

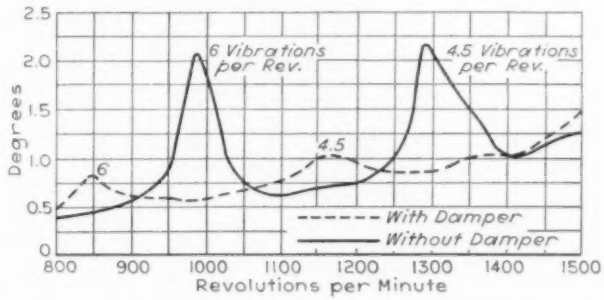


Fig. 24 Test of Engine With and Without Damper

It is, of course, true that a redesign of the connecting rods and crankcase follows an increase in crankpin diameter, but this has to be done only once, and the saving in service and increased reliability would appear to warrant such a change. In a new design, there would be no reason to incorporate a shaft whose vibration characteristics had not been investigated. It is worthy of note that a low-priced 6-cylinder automobile is said to have a crankshaft stiff enough to remove all torsional periods from the driving range, or up to a speed of about 60 m.p.h. This is without any form of damping device. Such a condition would, however, be impossible in an 8-cylinder, 9-bearing crankshaft, hence the damper becomes necessary. The use of a coupling with damping characteristics offers a field of great interest, as evidenced by the records of Figs. 7 and 16 of the paper.

AER-52-20

New Developments in Airport Lighting

By H. C. RITCHIE,[1] SCHENECTADY, N. Y.

The principles of airport lighting have been covered in many papers which have varied slightly and only in details, says the author, these principles now having been standardized through the medium of the Department of Commerce. These requirements are stated, and then the author points out some new developments which make it possible for airports to obtain maximum ratings for lighting equipment and also some of the new systems or devices brought out to supplement these requirements.

THE subject of airport lighting has been covered by many articles in the trade journals and in engineering magazines and also has been the title of many papers before engineering and commercial societies. The principles of airport lighting as propounded in these articles and in these papers have been practically the same and have varied only slightly in details.

These principles have been standardized through the medium of the Department of Commerce, Aeronautic Branch, by the publication of Airport Rating Regulations in Aeronautics Bulletin No. 16. These regulations, particularly referring to lighting, have been prepared by the Department after a comprehensive study of the lighting requirements and the recommendations of manufacturers of lighting equipment. The bulletin, published over a year ago, gives in detail the minimum requirements for lighting equipment for the different classes of airports.

It is intended to give briefly these requirements, and then to point out some of the new developments which make it possible for airports to get maximum ratings for lighting equipment, and also to state some of the new systems or devices which have been brought out to supplement these requirements.

Before a rating will be given to any airport there are certain minimum basic requirements which must be met. These are:

(*a*) A suitable landing area, described as a firm, smooth, well-drained plot of ground substantially level, without obstructions which would be hazards in taking off or landing, and with suitable approaches.

(*b*) Freedom from obstructions within the gliding angle to the field—namely, 7 to 1.

(*c*) The airport shall be situated on a good road leading to the nearest city or town.

(*d*) The airport shall be equipped with an approved type of wind-direction indicator.

(*e*) The landing area shall be marked, not only with a 100-ft. circle on the field, but the name of the city or town should be placed near the marker circle or on a building roof in letters visible from an altitude of at least 2000 ft. The outline of the landing area shall be marked with either sheet-metal cones or 4-ft. white circles placed not more than 300 ft. apart.

(*f*) If runways are used, they shall be constructed of a suitable material.

(*g*) The landing area shall be adequately drained so that the entire field is available for landing and taking off.

(*h*) The airport shall be provided with facilities for supplying aircraft with fuel, oil, and water, also drinking water shall be available. There must be dependable communication with the nearest city or town, as well as transportation facilities. Personnel shall be in attendance throughout the day or readily reached by telephone.

[1] Aeronautical Specialist, Lighting Section, General Electric Company.

Presented at the Joint Meeting of the A.S.M.E. Aeronautic Division with the Engineers Society of Western Pennsylvania, Pittsburgh, Pa., March 12, 1930.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The complete rating symbol of an airport consists of a letter, a numeral, and a letter. The first letter signifies the general equipment and facilities which are available; the numeral indicates the size of the field; and the second letter the night-lighting equipment.

To receive the highest, or "A," rating on night-lighting equipment, the airport shall have the following:

(*a*) An airport beacon, the type of which is described in full in Airport Rating Regulations.

(*b*) An illuminated wind-direction indicator.

(*c*) Boundary lights placed around the border of the usable landing area to indicate the extent of the field.

(*d*) Obstruction lights placed on all obstructions within gliding angle of the field.



FIG. 1 GENERAL ELECTRIC NOVALUX SERIES AIRPORT LIGHT (This is mounted on a Curtiss boundary cone; at Schenectady, N. Y., airport.)

(*e*) Hanger floodlights and roof marking must be such as to give the pilots a true picture of the airport buildings at night.

(*f*) A ceiling projector for determining the height of the ceiling at the airport must be provided.

(*g*) Landing-area floodlight system must be such that there shall be a minimum intensity of 0.15 foot-candle vertical or normal to the beam at all points of the landing area.

(*h*) The airport beacon, wind-direction indicator lights, boundary lights, obstruction lights, and roof-marking lights shall be kept burning all night and every night, and there must be sufficient personnel in attendance throughout the night for the proper operation of the other lighting equipment.

The lower ratings on night-lighting equipments require less equipment, but all except the lowest rating require all-night operation of the lighting equipment.

It will be seen that these requirements enable the pilot to find the port by means of the beacon, to see the extent of the landing area and the obstructions around the port, and the direction of the wind, and that they provide for illumination of the landing area so that he can make a safe landing.

During the last year several new types of beacons have been put on the market, and in addition improvements have been made in the Department of Commerce airway beacons.



FIG. 2 A 24-IN. ROTATING BEACON FOR AIRPORTS AND AIRWAYS

All probably are familiar with the design of 24-in. revolving airway beacons (Fig. 2) being installed by the Aeronautic Branch of the Department of Commerce on the National Airways. This consists of a 24-in. parabolic mirror, at the focal point of which is placed a 1000-watt, 115-volt incandescent lamp. This lamp is installed in a lamp changer, which, upon failure of the first lamp, automatically throws a second lamp into position. The front door of the beacon is fitted with a 25-deg. upward-reflecting cover glass which directs 10 per cent of the beam in an upward direction as an additional aid to the pilot when he is close to the beacon. The drum is also fitted with four zenith panels so that he can locate the beacon when almost directly over it.

On-course lights (Fig. 3) are provided at the beacon, one pointing each way along the airway. These are flashed by means of a cam and contact-making mechanism in the base of the beacon.

Improvements which are incorporated in the beacons now being installed by the Aeronautic Branch are the substitution of a spherical mirror in front of the lamp for the louvers or stray light shields formerly used. The louvers merely shielded the light so that the immediate area surrounding the beacon was not illuminated. The principal reason for this was to avoid annoying those living near the beacon location. The spherical mirror, however, not only cuts out this light, but it redirects a portion of the light back into the mirror and adds to the strength of the beam. This is not strictly a spherical mirror, but is really two halves of a hemisphere between which is placed a cylindrical section. This has the effect of lengthening the secondary image of the filament, and without detracting from the strength of the main beam gives additional light from the beacon to a wider angle. Adjustment is provided on this mirror to enable it to be focused with the lamp.

The flashing of the on-course lights is now accomplished through mercury-tube connectors rather than through air-break contactors. The signals are determined by means of a cam which is attached to the central revolving shaft of the beacon.

No change has been made on the design of the 18-in. course lights installed with every beacon, but the use of amber at intermediate fields has been abandoned and green substituted. The system now is to use red on-course lights at isolated beacons and green at intermediate fields and airports.

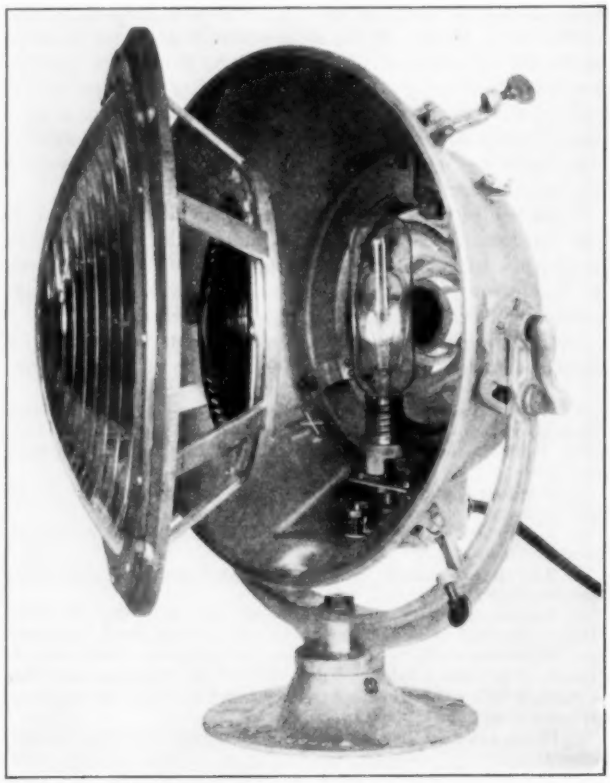


FIG. 3 AN 18-IN. ON-COURSE PROJECTOR
(The doublet lenses swing outward.)

At several places along some of the airways it has been found necessary to have small wattage beacons installed at closer intervals than is usual with the 24-ft. revolving beacons.

For this purpose there was designed the electric code beacon (Fig. 5), which consists of Fresnel-lens lanterns with two lamps, each of 200 watts. The lamps are flashed by the flasher with a

definite code signal, and while they are of much lower candlepower than the 24-in. revolving beacons, this code beacon has been found of considerable benefit in certain locations. This beacon also has other uses, and the equipment is now available for commercial application. It may be fitted with a green color shade and with 500-watt lamps instead of 200-watt lamps and used as an auxiliary beacon at an airport, being flashed by the revolving beacon, with the code letter giving the airport designation. This would mean that the approaching pilot would not only know that there was an airport close at hand, but would be able definitely to identify the airport. It has been recommended that the auxiliary beacons be installed directly above the revolving beacon.

This lantern may also be used and is recommended for the marking of towers of transmission lines crossing navigable rivers.

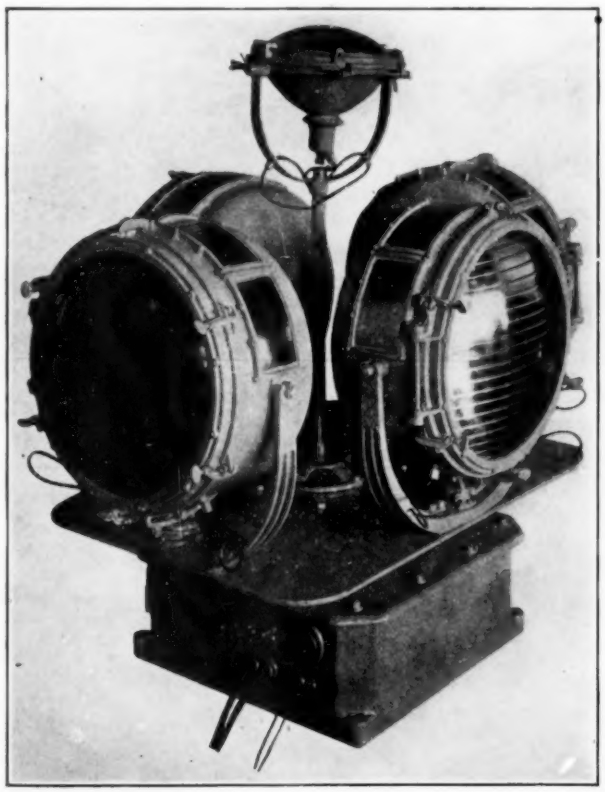


FIG. 4 QUADRUPLE AIRPORT BEACON

In this case the lantern is fitted with a red color screen and is not flashed.

Another type of airport beacon lately put on the market is enclosed with a large glass dome which covers the revolving mirror and lamp. This beacon has the advantage of giving full zenith indication, with the necessary eclipse period. Another great advantage is that there are no revolving portions exposed to the weather or to the wind. It has been found on the 24-in. revolving beacons that there is an uneven torsion on the center shaft and on the worm gear and worms caused by the non-symmetrical shape of the beacon drum. It has also been found that the presence of snow and sleet on the base of the beacon retards the rotation and in some cases damages the mechanism.

The glass dome is mounted in a cast frame which is hinged at the bottom, allowing the dome to be lifted for inspection of the beacon and for lamp replacement. The large covers in the base



FIG. 5 AIRWAY ELECTRIC CODE BEACON

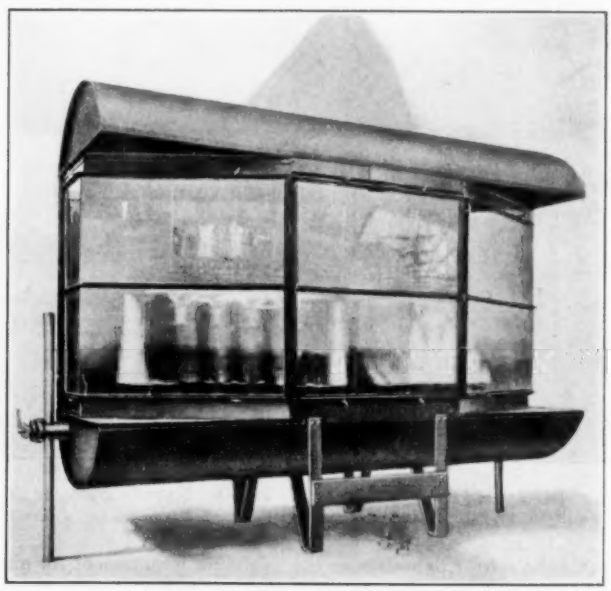


FIG. 6 A 24-KW. AIRPORT FLOODLIGHT

give easy access to all parts of the mechanism and allow easy replacement or adjustment of any parts.

The beam candlepower of this beacon, which uses a 20-in. deep parabolic mirror, is well over the minimum requirement for airport use and has the advantage also of giving a wider beam than the more shallow standard 24-in. parabolic mirror.

As a distinctive beacon for airports there has been designed the quadruple beacon (Fig. 4), which consists of four of the 24-in. Department of Commerce type drums fitted with front-door glasses of several color combinations, these four drums being surmounted by a floodlight with a diverging lens. One has been in use at Cleveland Airport for over a year and has been found

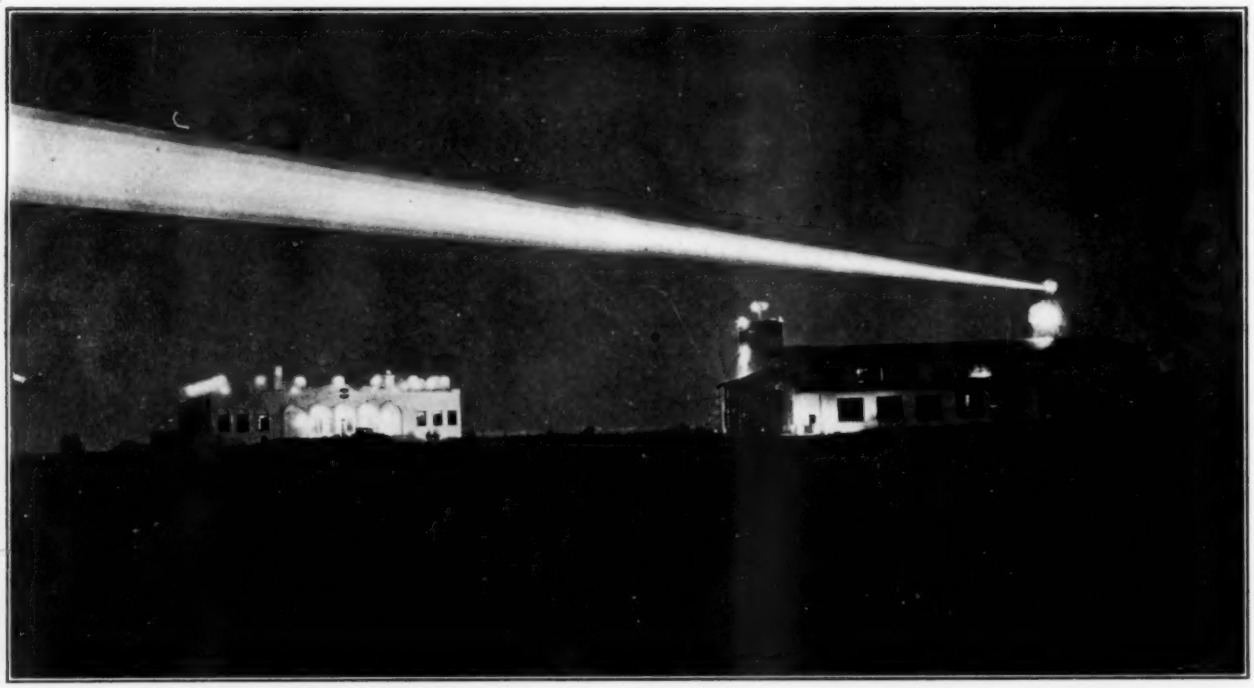


FIG. 7 NIGHT VIEW OF MURFREESBORO AIRPORT, NASHVILLE, TENN.
(Showing installation of 24-in. rotating beacon.)



FIG. 8 ILLUMINATION OF MURFREESBORO AIRPORT AT NIGHT
(One 24-kw. General Electric floodlight is used.)

of great value to approaching pilots. Its red and white flashes definitely distinguish Cleveland Airport from airway beacons or other lights in the vicinity, and its wide vertical spread enables it to be picked up at any distance within its range.

The airport rating regulations recommend the use of sheet-metal cones with boundary lights to serve principally as daylight markers. A cone (Fig. 1) has been developed which supports the boundary-light fixture, the base of the cone resting directly on the ground,over a disconnecting pot-head cutout. The fixture is connected by flexible cable to the plug portion of the cutout, and in case the boundary light is knocked over or dragged across the field by either a plane or an automobile, the plug is pulled from the receptacle and the series is completed by means of the spring clips in the receptacle.

The receptacle is provided with a spring cover which upon the pulling of the plug closes the opening against the entry of foreign matter. This cutout is not waterproof, but provision is usually made to so mount these receptacles that they will be above any water which may stand on the field.

While the airport rating regulations of the Department of Commerce limited the voltage of series circuits for boundary lights where insulating transformers were not used, they have permitted the use of any voltage on the boundary circuits where these disconnecting pot-head cutouts are used with each boundary light. Replacement of the plug in the receptacle is made easy by means of a large handhole and cover in the cone.

For floodlighting large landing areas one manufacturer has introduced an incandescent lamp unit which has a total capacity of 24 kw. (Fig. 6).

Differing from previous designs of large capacity floodlights, the wide horizontal angle of spread of the beam is obtained, not by lenses or front-door strips, but by the mirror arrangement. All are acquainted with the property of a parabolic reflector whereby a light source placed at the focal point of the reflector produces a beam with approximately parallel rays. A new type of mirror was designed for this floodlight which has a parabolic section in the vertical plane and is a straight line in the horizontal plane. A light source placed on the focal line of this mirror produces a beam in which the rays are parallel in the vertical plane and diverging in the horizontal plane. The best arrangement which has been found for this large floodlight has been of two banks each with three mirrors and four 3-kw., 32-volt lamps. The two banks are placed at an angle to each other to obtain a total horizontal spread of approximately 180 deg. The photometric curve shows a very even distribution and is especially arranged for very large landing areas. Smaller angles of horizontal spreads with corresponding increase in beam candlepower may be obtained by putting the two banks of mirrors in line. It is very seldom, however, that a spread of less than 180 deg. is considered advisable. Each one of the lamps in this floodlight is fitted with a spherical reflector which not only acts as a louver, to cut out stray light and to shield the light source from the approaching pilot, but redirects a considerable amount of light back into the parabolic mirror. The large mirrors are adjustable so that they may be perfectly lined up with each other and with the lamp.

One very important requirement of airport rating regulations is that all equipment shall be so operated and maintained at all



FIG. 9 NIGHT ILLUMINATION OF BOLLING FIELD AT WASHINGTON, D. C.

times as to meet at least the requirements of the regulations for the particular rating issued.

This applies particularly to lighting equipment which can be permitted to deteriorate in efficiency. One of the great problems at an airport has been the collection of dust on the lighting equipment, and in the case of floodlights which are ventilated, this dust is drawn into the floodlighting fixture and deposited on the reflecting surfaces and on the lamp. The large incandescent 24-kw. floodlight which has been described is enclosed as completely as possible, and no natural or forced ventilation is provided. As a substitute the volume and radiating surfaces of the floodlight are so proportioned that the 24 kw. of energy is readily dissipated by the external surface. This leaves only the problem of cleaning the covered front glass of the searchlight, which can readily be made a matter of daily routine.

As a supplement to the requirements of the Department of Commerce, a new system of airport lighting has been installed on several airports and gives promise of being of considerable assistance to the pilot when landing or when taking off at the airport.

By means of lighting units set flush with the surface of the ground and arranged in rows over the landing area, the pilot is given definite lanes on which to land. These lights are 100 ft. apart and mark quite definitely the surface of the field. Marker lights placed at the ends of each row indicate the direction of the prevailing wind, these lights being controlled automatically by a wind vane.

It is not proposed that this system should be a substitute for field floodlighting, but only as giving additional information to the pilot.

AER-52-21

Automotive Methods and Practices Involved in Aircraft Engines

By O. E. SZEKELY,[1] HOLLAND, MICH.

The author draws a striking comparison between the cost of standard parts in automobile manufacturing and the cost of parts in airplane manufacturing. Iron and steel and other metals that enter in automobile manufacturing can be secured in the market at apparently a very much lower price than when it is known they are intended for use in airplanes. Although airplane making is a new industry, most of the principles involved in aircraft-engine manufacturing have long been practiced in the internal-combustion engine industry.

VAGUE as this subject may seem, yet it is felt, from the point of view of general economics and from the public interest manifested, that it is a major factor.

At first the intention was to make it a purely technical paper in order that its discussion might bring out definite and distinct points of interest to members; however, in the end it seemed more timely and advisable to have this topic receive the cloak of generalities under which the various details may be discussed by those who wish to enter into this problem from a deeper and purely technical angle.

The aeronautic industry did not spring up uncalled for. Necessity and the demand of the present-day methods of doing things and arriving at results brought about ideas of inventors and activities of engineers, designers, and, finally, business men endeavoring to solve locomotion in a manner different from that available.

The ground has been exploited in a rather generous manner—first, the coolie carriage; then, animal-drawn vehicles such as wagons and stages; later on, steam and electrically propelled and rail-guided trains and cars; recently, the automobile, truck, passenger bus, and in the last few years elevated lines and subways.

While it is true that room is still available and that prairies cover territories infinitely greater than are covered by cities, yet because of geological, intercommunicating reasons and proximity to raw and finished materials for the sustenance of our daily lives, cities and towns are located within reach of one another.

Traffic in the last few years has grown out of all proportions to the space in which it performs; therefore, the conquering of the air started to materialize as a natural result. From the point of view of transportation, it is boundaryless and magnificently unrestricted as to speed, territory to be covered, distances in any direction, and the manner in which it is accomplished; from that of the general public, it is already a marvelous achievement; but from the point of view of the engineering profession, we know it is in its infancy.

For comparison between the automobile and aircraft, may not a distinction be made between the two, and this is that, with the advantages of the automobile running on terra firma on four wheels, it is only natural that one feels a certain comfort in knowing that by following the rules and regulation of common sense in the maintenance of speed and direction, no mishap will occur; while in flying, even under the most careful piloting of the craft, the element of danger is present.

Let us now discuss the motive power or power plant of an aircraft and make it the subject, both from the point of view of interest to engineers and that of the public in general. It must not be forgotten that without public acceptance of public demand, or without public absorption of the things that engineers develop or produce, there would be no incentive for further developments or further production.

The second move in this direction should be the possibilities of producing something at a rate and under conditions consistent with the present knowledge of proper design and production, coupled again and unforgettably with the economics of the world, in order that no failure shall come on part of the public in using up the energy so expended in the form of a product, thereby offering additional incentive for additional and better, more developed, and still further improved articles.

One might say, of course, that this is a vicious circle. And that is true and should be, because the moment that complete satisfaction is manifested in what we already have, then all ideas and progress will have ceased. Public demand interpreted in the sense of human endeavor will always try to produce a need for more.

This may be brought down to human effort in producing a mathematical formula. Let human experience, knowledge, and endeavor combined equal "effort" and call it Y, against which there is public consumption, public demand, and repetition of

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

public desire for more and better products and make them equal "usefulness," and, for the purpose of mathematics, call it X, and it will be found that the growth of one is relative to the growth of the other. If X is to grow in some proportion and Y is to follow, one can set up an equation in which can be found the resultant and the ratio offered by the two, thus:

If X is to grow at the rate of dx and Y is to grow proportionately at the rate of dy, it is only necessary for us to find the value of dy/dx.

For example, $Y = X^2$ (assuming that public demand is in such proportion to effort)

then: $Y + dy = (x + dx)^2$

$Y + dy = X^2 + (2X \times dx) + (dx)^2$ [$(dx)^2$ is so small, it is negligible]

$$dy = (2X \times dx)$$

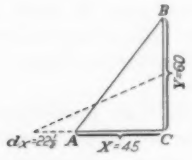
$$\frac{dy}{dx} = 2X$$

There is practically double the amount of public demand relative to engineering skill and available effort. This alone necessitates constant development and, mathematically solved, gives you a differential formula as to the manner in which effort must grow to the ratio of usefulness.

It behooves all to remember that an internal-combustion engine in any shape is still an internal-combustion engine, and while it is also true that on account of its peculiar characteristics it may be proportionately more expensive from the point of view of actual dollars expended in the manufacture and production of same, still there are limits set by public demand and by public absorption.

This, necessarily, brings about the question of cost. The circumstances must be mitigated by the fact that costs can only be reduced by mass production. Tools and operations assist only to the limit of their expediency; from which point, then, number of pieces per unit of time controls production costs. In the comparison, attention is called to the fact that a six-cylinder automobile engine produced at the rate of 60 per day involves a $45 net labor and material of approximately $40, or a total of $85.

Assuming that the material in an aircraft engine must have, say, 50 per cent higher quality of materials, therefore, a 50 per cent higher cost, a 50 per cent greater labor expenditure, this would relatively change the net production of aircraft engines to 33 per day. This is shown in the following formula:



$X = \$45$

$Y = 60$ engines

$dx = \$22.50$ or 50 per cent

$AB = \sqrt{(AC)^2 + (BC)^2}$

$AB = \sqrt{(45)^2 + (60)^2} = 75$

$(y - dy)^2 = (75)^2 - (67.5)^2$

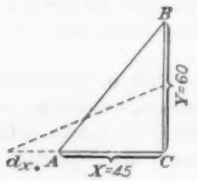
$= 5625 - 4556.25 = 1069$

or

$$y - dy = \sqrt{1069} = 32.6 \text{ or } 33 \text{ engines per day.}$$

Assuming now that, instead of 60 engines, there are only 5 engines per day produced—the cost of operations is increased relative to the 60 engines, and the labor cost on 5 engines per day to $74.83.

Reversing the formula to prove the cost per engine on the basis of 5 per day, it is as follows:



$AC = 45$

$BC = 60$

$AB = 75$

$Y = 60$

$X = 45$

$dy = 55\ (y - dy) = 5$

$dx = ?$

$(Y - dy)^2 = (AB)^2 - (X - dx)^2$

$(X - dx)^2 = (AB)^2 - (y - dy)^2$

$= (75)^2 - (5)^2$

$= 5625 - 25 = 5600$

$X + dx = \sqrt{5600} = \$74.83$

The cost of materials, as mentioned before, is 50 per cent greater or $42.50, or a total of the two of $119.83. Add to this the original cost of $85, or a grand total of $204.83, and taking advantage of any possible scrappage at the rate of 10 per cent, or $20, the total gross cost of manufacturing such an engine would still be around $225. At the rate the industry is at the present time operating and under the conditions in which the various parts are being manufactured, the price of a six-cylinder aircraft engine in comparison with that of an automobile engine is approximately five times the price of an automobile engine.

It is a well-known fact that a six-cylinder 75-hp. engine can be purchased in the automotive industry for approximately $250, yet a 75-hp. aircraft engine, even to manufacturers of aircraft, ranges around $900 and with a list price of from $1400 to $1600, and on a 100-hp. engine from $2000 to $2500 for the same design.

In order now to reconcile these statements with manufacturing methods and principles, let it be stated that all materials and all labor in an aircraft engine must be beyond impeachment. Nevertheless, the same machinery and equipment at present producing automobile parts are being used for the production of aircraft engine parts; therefore the primary investment did not change.

It is hardly believed necessary to propound the fact that the materials used in automobile engines are not in the least inferior in many respects to the materials used in aircraft engines, because automotive-type engines have been performing very well for a number of years, and if that is true, it is hardly reasonable to believe that the same material if properly handled and used in the proper places under the proper conditions, will not perform similarly in an aircraft engine.

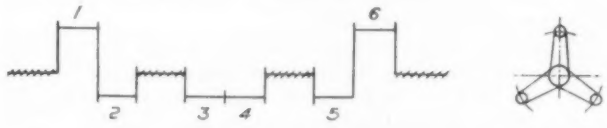
Nevertheless, an automobile crankshaft is manufactured from between 35 to 50 per cent carbon, heat-treated alloy steel. This crankshaft has six throws and not less than three main-bearing surfaces. Also, such a crankshaft has integrally forged and machined balance weight. These crankshafts are manufactured within a limit of 0.001 in. to the diameter of the crank and bearing pins. This is as close a limit as is absolutely necessary for the proper performance of a crankshaft, and it is a commercial tolerance under present manufacturing conditions. The manufacturing price of such a crankshaft is around $12. Compare this now with the same crankpin diameter single-throw crankshaft for aircraft purposes in which the material is a chrome alloy material properly heated-treated—however, without balance weight integrally forged. The purchase cost under the present production system in the aircraft industry for this single-throw crankshaft is $16, or, relative to the six-throw automotive type, eight times the manufacturing cost of the six-cylinder automotive six-throw crankshaft.

Assume a six-cylinder crankshaft, six crankpins $1^1/_2$ in. diameter, 2 in. long, and four main bearings $1^1/_2$ in. in diameter $2^1/_2$ in. long; therefore the total machined projected area would be 33 sq. in.; four set-ups are necessary. Compare the same with a single-throw crankshaft with three main-bearing contacts of 2 in. in diameter, 3 in. long, and one crankpin $1^3/_4$ in. in diameter, $2^1/_2$ in. long, or a total of approximately 22.37 sq. in. of machined surface, and only two set-ups are necessary for this purpose.

The ratio between the two set-ups will be $^1/_2$ or 50 per cent. The bearing-surface ratio will be 22.4/33.0, or approximately 0.68.

However, let it be assumed that the production ratio, on the same basis as the engines were figured, would be 60 automotive engine crankshafts against 5 crankshafts single-throw for aircraft purposes, or a ratio of 12 to 1. Also add to this the fact that the general machine work on the all-over finished aircraft crankshaft is 100 per cent more than on the automotive type crankshaft, or 2 to 1.

One can then set up the following formula:

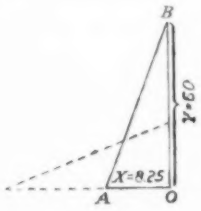


6 crankpins $1^1/_2$ in. × 2 in. = 1.5 × 2 × 6 = 18 sq. in. proj. area
4 main bearings $1^1/_2$ in. × $2^1/_2$ in. × 4 = 15 sq. in. proj. area 60 per diem production
Total = 33 sq. in. proj. area
33 sq. in. at $0.25 per sq. in., or $8.25 labor.
$8.25 × 60 = $495 and approximately 50 per cent of forging machine finished, however, produced on four different center set-ups.



Single-pin bearing $1^3/_4$ in. × $2^1/_2$ in. = 4.37 sq. in. proj. area
3 main bearings 2 in. × 3 in. × 3 = 18.00 sq. in. proj. area
Total = 22.37 sq. in. proj. area

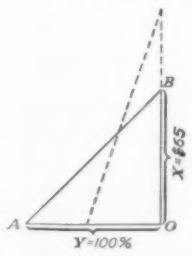
This shaft is machined only on two center set-ups, however, at the rate of only 5 per day; therefore the cost may be relative, thus:



$$AB = \sqrt{(OA)^2 - (OB)^2} = 65$$
$$Y = 60$$
$$X = \$8.25$$
$$(X + dx)^2 = (AB)^2 - (Y - dy)^2$$
$$= \sqrt{65^2 - 5^2} = \$64.80$$

However, for the sake of comparison, it may be admitted that the other finished surfaces will double the normal surface of an automotive crankshaft, therefore, relatively, raising the cost thus:

Finished surface aircraft 100 per cent = Y
Finished surface automotive 50 per cent = dy
Price (5 per day) automotive $64.80 = $65.



$$X = \$65$$
$$Y = 100 \text{ per cent}$$
$$dy = 50 \text{ per cent}$$
$$AB = \sqrt{OA^2 + OB^2} = \$119.26$$
$$(X + dx)^2 = (AB)^2 - (y - dy)^2$$
$$= \sqrt{119^2 - 50^2} = \$107$$

Another example is valves and valve materials. Silichrome or tungsten valves of $1^3/_4$ in. size for automotive purposes can be purchased between $0.40 and $0.80, depending upon the quantities purchased. The same valve of the same specification for aircraft purposes, manufactured under exactly the same methods of procedure, can be acquired at a price of between $2.50 and $4.50 each.

Piston pins do not break in an automotive-type engine. It is well known that the highest grade of automotive-type piston pins can be purchased at from $0.12 to $0.32, while the same size aircraft-engine piston pin is $1.25 to $3.50.

This disease has even reached the bearing manufacturers. From automotive practices, it is known that any make of S.A.E. No. 308 deep-grooved ball bearing can be purchased for around $2 each, but when an aircraft engine is manufactured the price is much higher.

Another interesting feature of the spark-plug picture is the fact that the same spark plugs which can be purchased in quantities by automotive-engine manufacturers for not over $0.10 a piece sell to the aircraft industry for from $0.50 to $1.50.

A carburetor for an automotive-type engine can be purchased by automotive-engine manufacturers at a price of between $5 and $7.50 and not over $10. Compare this same carburetor in manufacturing limits and tolerances, in economy, in durability, and in appearance and workmanship with an aircraft carburetor manufactured by the same people in which the limits and tolerances are not extremely far over those used in the automotive carburetor and in which the materials are not extremely better, yet a carburetor of the same capacity for the same horsepower is around $40 to $50.

Cast iron for cylinders in automotive-type engines has for a long time not been known to reach over $0.10 a lb., and such gray iron for internal-combustion engine purposes has been specifically defined by the service demanding 0.035 carbon, absolutely pure, less than 0.4 of 1 per cent sulphur, and very close grain, with a tensile strength upward of 30,000 to 35,000 lb. The same material for aircraft cylinders, allowing a certain margin for the quantities produced and for other necessary requirements, has not been known to be less than $0.35 to $0.45 per lb.

The aluminum industry is no exception when one considers that the aluminum alloy price per pound ranges quite a good deal higher than $1 a lb. for aircraft engines.

These conditions present problems which the public wishes to solve and puts it up to the engineering profession and, in turn, to the manufacturers, until the manufacturers themselves realize that the present-day methods are inadequate in the aircraft industry and that they must borrow knowledge from the automotive industries.

But little has been said about overhead in the aircraft line as against the overhead charges in the automotive line, but the crux of the situation at the present time can be laid, in addition to the mistaken available returns on produced pieces and parts, to that of the maintenance of overhead not tolerated nor known in the automotive industries.

The author would call attention to articles published in *Aviation* in the issues of February 15 and 22, through the efforts of Edward P. Warner, in which the majority of replies to a series of questions point toward a long-awaited realization and solution of the present problem. The exactness of the demand in the aeronautic industry has not been thoroughly answered—not because engines have not been able to withstand the punishment meted out by their use, but simply because no heed is taken of the already established principles, methods, and costs of operations readily available in the automotive industries.

The various prices and figures herein used are necessarily arbitrary and purposely outside of the limits of common knowledge and practice and are used purely for the comparison and

the reasoning established so that the results may show nothing but the proper ratio established.

As a matter of fact, so far no standards have been established, and each and every manufacturer of aircraft engines has an idea of his own as to the necessity of the parts being truly a special unit unlike any other on the market, while if, under the proper methods of production, parts were to be standardized or nearly so, the net labor cost and overhead figures in the manufacture of aircraft engines would drop to an acceptable level.

One must not forget, however, that this is a new industry, but it is felt that most of the principles involved in aircraft-engine manufacturing have long been practiced in the internal-combustion engine industry. The author, therefore, would summarize as follows:

(1) Consideration and application of automotive manufacturing principles and practices in the manufacturing of aircraft types of internal-combustion engines.

(2) The development of means and ways on an already established basis to enhance the ability of aircraft-engine manufacturing to reach mass production on a standardized basis.

(3) Let it be admitted that public demand and acceptance as well as absorption must be created; therefore the reasons for the industry's slow progress in sales could be classified as 60 per cent prohibitive cost and price and 30 per cent public knowledge of undefined character in the handling of matters, coupled with natural fear.

Truthfulness would call for taking an inventory of ourselves and answering the following questions:

(1) Are we in position to assist in stabilizing the industry through any means whatsoever?

(2) Do we realize that this industry is here to stay and that the mode of living for the future will be changed?

(3) Are we ready to accept the responsibility of facing the facts produced by the old adage of supply and demand?

(4) Are we willing to accept past experience as a teacher and use the knowledge gained to further the advantages?

(5) Are we applying our best efforts according to the responsibility placed upon us by the public in general?

(6) Knowing what we lack, are we doing our best to overcome this incompatibility?

AER-52-22

Metallizing the Airplane

By H. V. THADEN,[1] PITTSBURGH, PA.

At present there appears to be a gradual trend toward the use of metal in airplane construction. There are many design problems to be solved by research and experiments. Three basic materials or their alloys are now used in airplane construction. They are steel, aluminum, and magnesium. Extruded sections of duralumin can be formed readily and afford wide possibilities for moldings, trim, and primary structural members.

Internal cantilever wing bracing has developed to a point where it compares favorably in weight with externally braced structures. A smooth surface covering is most efficient, and rounding or filleting of corners and fairing the curves of the structure give high aerodynamic efficiency. The paper gives advantages and disadvantages of metal construction for airplanes.

PROBABLY one of the few opinions unanimously held by members of the aeronautical industry is that the airplane of the future will be all-metal. In fact, it is quite generally agreed that this is the ultimate. It is also interesting to note that prophecies of when this goal will be reached depend on the prophet's knowledge of the success of recent radical departures in metal structural design.

The advantages of metal airplanes generally conceded are (1) that their life is longer, (2) they can weather outside storage, (3) they are more fire resistant, (4) their structure is more homogeneous and crash resistant, and (5) they lend themselves more readily to mass-production methods.

There is a definite indication of a gradual trend toward the use of metal by most manufacturers. In some cases, one notes efforts being made merely to change the component wooden parts to metal. In other cases, component sub-assemblies are completely metallized, unit by unit. The latter method is decidedly better than the former, but in the author's opinion neither method is as satisfactory as going the full way and metallizing the entire structure, divorcing completely all wood or composite structural precedent. The problems involved in metallizing are distinctly different, and are most successfully solved by reverting to fundamentals rather than precedent.

The problems are not simple, and much research and experimentation are essential. The fact that so few really good metal planes exist today is proof of this. However, notwithstanding these difficulties, or possibly because of them, numerous minds are at present working on this interesting problem. In Germany, several schools of thought have reached quite successful solutions; more recently the French and English have been meeting with successful solutions; and here in America, there are a number of schools of thought just getting under way.

The attack on these problems is influenced by a number of factors, of which the business policy, the choice of material, the structural design, and the aerodynamic design are most important. Of these four, the first very largely influences the other three as it has to do with such fundamental economics as cost of production, volume, and profits.

In practically all production work, it is a well-established fact that the greater the number of similar articles produced, the cheaper they can be made because of the better production machinery which can be profitably utilized. However, production machinery, special jigs, and dies are expensive and the volume of the airplane market is comparatively small. Also, engineering obsolescence is, and will be for some time, a most serious factor. Under these circumstances, some compromise solution must first be made to determine how much money can be justifiably spent for special machinery, jigs, and dies in anticipation of a given volume of production. Even a most cursory analysis shows that airplane production is not today, nor likely to be for several years, in a class comparable to the automobile industry. In fact, a survey reveals that the aeronautical industry is comparative in size, and to some extent in market, to the present motorboat industry.

Considering all this, it behooves one to use conservative judgment for the next few years and to curb natural desires to install the most modern, or most specialized, production machinery because, obviously, the volume will not warrant such expenditures.

By this there is no implication that every effort should not be made to use existing production machinery, but merely that during the inevitable transition from small to large production, it might be profitable rather to accept higher charges for more hand labor than to lower special machine-made charges to which a disproportionately high burden must be added. The low prices of automobiles are due to the tremendous volume in which they are made and to the use of highly developed, but unfortunately costly, jigs and dies. Airplanes will never be on a comparable cost basis until they have a like volume and similar jigs and dies.

It might be well, then, so to modify present designs as to use the available production machinery, to simplify designs, and to standardize as much as possible in order to reduce the cost price. As the volume increases designs can be altered more profitably later to suit the automotive mass-production methods.

Choice of material is extremely important and must constantly be kept in mind. In these days of intensive research, a metallurgical development might well alter an entire production procedure. Lighter and stronger materials are constantly being developed as well as commercialized. There are today three basic materials, or their alloys, in use in commercial aircraft; namely, steel, aluminum, and magnesium. Of these, magnesium is still used only to a small extent, but its extreme lightness gives it interesting possibilities.

Magnesium sheet in thin gages presents certain unsolved problems of which fire resistance and corrosion are the most serious. Castings and forgings, however, have a useful field for low-stressed parts such as in secondary members. Steel has found extensive use in aircraft in various classes from the medium carbon to the high alloy. Seamless tubing in both the carbon and in the chrome-molybdenum alloy is used predominately in fuselage structures, and also to some extent in the spars of wing structures. Forgings, where a sufficient number of similar parts justify the expenditure for dies, form reliable members for the primary structure. Thin alloy-steel sheet in strip form is used in numerous places as it can be readily formed and welded. The sheet stock or the tubes, when welded and later heat-treated, develop very near their full physical prop-

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

erties, but in the event that length or size make heat treatment, after welding, impracticable, neither the physical qualities nor the fatigue resistance can be realized.

Aluminum is being used in increasing extent, both in the commercially pure state (98 per cent) and in various alloys such as duralumin and alclad. The latter material has particularly good corrosion-resisting qualities because of the coating of pure aluminum with which it is surfaced. Aluminum-alloy tubes, so far, have been commercially obtainable of duralumin only, but it is promised that they will also be made of alclad in the near future.

Of the aluminum alloys, probably the best known is that of duralumin which contains a small amount of copper. Under proper heat treatment this material finds very extensive use in the primary structure as well as in the covering of an airplane. The fact that it has considerable more bulk for a given weight than steel makes it more resistant to secondary failure such as the crinkling of the free edges.

Extruded sections of duralumin can be readily formed and offer wide possibilities for moldings and trim as well as for primary structural members. To date, the minimum practical thickness to which this material can be extruded is about $^1/_{16}$ in. Duralumin castings can be employed where stiffness or bearing area is required, but it seems somewhat hazardous to utilize them for tension members in a primary structure. Duralumin forgings have proved especially practicable and will unquestionably be used to a large extent. One of the advantages of aluminum alloy is the possibility of heat-treating the material, subsequently forming and shaping it to size, and still retaining the ultimate physical properties by virtue of its "aging" characteristics.

Generalizing, it would appear that these various materials could be advantageously used in some of the following ways. For pure tension, the best material from a structural and weight standpoint are the alloy steels. For pure compression, in short columns the magnesium and aluminum alloys offer possibilities except in very highly stressed members. For bending and compression, the limitations of available space for the structure determine the most practical material. In the event that steel is used, the thinness of the material would probably be the criterion; in this event, a bulkier material such as duralumin would work in more efficiently. In fact, the most serious limitation in the use of sheet metal of either duralumin or steel is the inability to use the material in the most economical sizes because of the stiffness factor.

The structural and aerodynamic design of airplanes are more or less interrelated, having their closest point of contact in such matters as weight and parasitic resistance. Some of the important factors influencing these problems are the relative merits of the monoplane and biplane types, the method of wing bracing, and the surface covering.

In the author's opinion, the monoplane makes the better commercial type of plane because of its simpler construction, the fewer number of parts, and the greater aerodynamic efficiency (some 10 per cent). On the other hand, greater maneuverability is obtained with a biplane structure due to its smaller moment of inertia. This, however, is of more importance to military than to commercial ships.

Internal cantilever wing bracing, which is of relatively recent origin, has developed to a degree where the weight involved is comparable to that in the externally braced biplane structure, or in the externally braced monoplane type. With the weights of the two types about on a par, the problem now revolves about the degree of profile resistance unavoidably present with thick-profile sections required in the cantilever type compared to the thin profile sections of the externally braced type with their extra strut braces. The trend seems to be toward the thicker cantilever wings, however, which have a higher lift permitting lower landing speeds and quicker take-offs.

In the matter of surface covering obviously the most efficient type is one with a smooth surface and conforming to a streamline shape. Rounding or filleting of the corners of structural intersections and generally fairing the curves of an airplane structure tends toward high aerodynamic efficiency. A rectangular round-cornered or elliptical fuselage is more efficient than a rectangular square-cornered type. In general, the fewer protuberances from the essential portions of an airplane, namely from the wings and fuselage, the less power is required to propel the airplane through the air. Full-scale wind-tunnel tests indicate smooth metal as having the least surface skin friction, with, however, painted metal and unpainted plywood having only slightly greater resistance. The influence of corrugating the metal, with corrugations similar to those used by Junkers and Ford, is a relatively small factor when studied from the standpoint of performance. The increase in surface area is approximately 15 to 20 per cent, and on a wing test alone, a decrease in efficiency of somewhere around the same amount is noted in the L over D between a smooth and a corrugated wing cover. However, when this loss is transformed into terms of performance of a complete airplane it is found that the horsepower required results in a loss of top speed of only 1 to 2 per cent. This loss is even more negligible at lower speeds and is practically zero at climbing speeds. This is better appreciated when it is remembered that at top speeds less than $^1/_3$ of the total horsepower is required to overcome the total wing drag. An interesting aerodynamic phenomena which is brought out by these same full-scale tests on corrugated sheet is that there is a general flattening out of the lift curve near the burble point. This has the effect of delaying the tendency of the wing to autorotate and thereby delay spinning.

Consideration of these various aerodynamic factors leads one to certain conclusions regarding the most adaptable type of structural design. It has been reasoned that if the outer cover is necessary for aerodynamic reasons, and if a structure is required to carry the imposed loads, then it might be well to design the covering to serve both purposes. This line of reasoning has brought about another principle of structural design.

At present there are two basic schools of structural design; one makes use of the conventional type of framed structure in which stresses are statically determined by well-known methods of analysis, and the other follows the "monolithic" (a term to designate single-piece construction which has recently come into use) type of structural design which in general is not amenable to conventional determinant analysis. This latter type has been generally taboo because of this fact, but notwithstanding the limitation of formal analytical analysis, it is gradually coming more and more into use. This type of structure must usually be physically tested to determine its structural strength, at least until sufficient data will have been acquired to correlate the theoretical hypothesis with the actual test structure.

Most airplanes today are designed on the basis of the first type, namely, the determinant framed structure type. Thus their design is based on fairly exact methods of calculation. The loads in each member being determinant, an economical primary structure usually results. Unfortunately, however, these structures, especially the fuselage, generally have aerodynamically inefficient cross-sectional forms and their square or rectangular, or even triangular frames have to be padded up and the corners must be rounded to give the desired streamline to the structures. This secondary structure is inefficient from a weight standpoint, besides being costly both in the initial manufacture and in field repair. On the other hand, the "mono-

lithic" or monocoque type of structure can be most easily made with rounded corners, rather than square, and the surface covering generally "streamlines" its contours naturally. As stated, this type of structure is not calculable and must be tested statistically to prove its strength. However, as the number of units of a given type increase with production, this disadvantage becomes negligible.

An important advantage of the "monolithic" type of structure is that small holes, or discontinuities in the surface, do not in general seriously detract from the strength. In a frame structure, the failure of one of the few essential structural members generally results in a total collapse. Thus from a military standpoint, shell holes would be decidedly less hazardous in the monocoque type.

One of the first structural components of the airplane in which this principle of construction was applied was the fuselage. The earliest types, and even some of today, use wood in a laminated form to make the outer contour of the fuselage. This type, however, has certain limitations because of the climatic variations to which an airplane is subject, the lack of resistance to fire, and splintering of the wood under crash loads. Likewise, this type of covering is not adaptable to mass production methods; therefore, attempts are made to metallize this type of structure. Two general types are being evolved. One uses a multiplicity of transverse bulkheads of the required aerodynamic cross-sectional shape, to the periphery of which are fastened longitudinals. The resultant "bird-cage" structure is covered with flat sheet metal. Securing the sheet to the bulk-

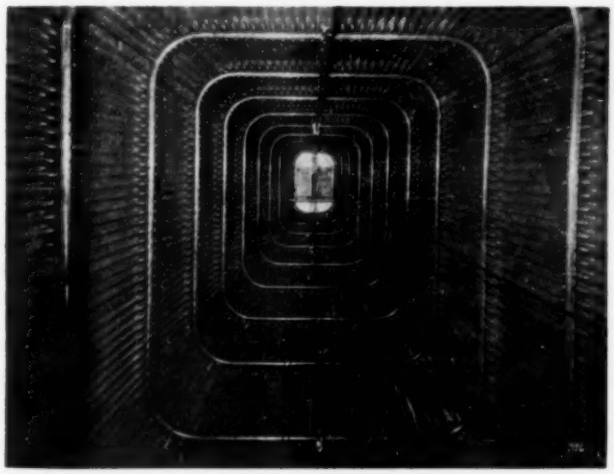


FIG. 1 INTERIOR OF MONOCOQUE FUSELAGE OF THADEN T-4 ALL-METAL AIRPLANE

heads and longitudinals has the effect of splitting the structure up into a series of small rectangular frames riveted to a flat plate skin, the flat skin acting as a shear web.

The second type of fuselage structure which is being evolved is one using a somewhat similar type of transverse bulkheads but omitting the longitudinals and replacing them with sheet metal corrugated longitudinally. This is, in effect, the equivalent of a multiplicity of very small longitudinal members fastened to the transverse bulkheads. In this case the skin takes all of the shear loads as in the flat-skin type, and also takes all of the chord loads due to the primary bending moments. The skin is truly stressed and, in fact, the bulkheads serve their chief purpose as separators constraining the sheet to its designed position. An example of this type of fuselage is shown in Fig. 1, in which the bulkheads are formed of aluminum-alloy tubes bent to the desirable aerodynamic cross-section, the longitudinally corrugated skin is riveted to the tubular bulkheads.

The wing structure is developing along much the same lines as the fuselage, but at a somewhat slower rate, chiefly because of the apparent efficiency of the conventional two-spar wooden and fabric construction. At least this is so with the externally braced monoplane and biplane types of structure. In the case of the cantilever monoplane, increasing spans and higher speeds demonstrated a serious limitation of the conventional type of wing structure. The resultant increase in torsional forces necessitated something stiffer than that obtained with the conventional wire drag bracing. Additional bracing was obtained by replacing the fabric cover with plywood. This materially increased the torsional resistance of the monoplane wing. A number of excellent examples of this type of wing are being



FIG. 2 REMOVABLE WING PANEL SHOWING INTERIOR OF MONOCOQUE CANTILEVER WING OF THADEN T-4 ALL-METAL AIRPLANE

made today, of which the Fokker is outstanding. However, wood in the wings, as in the fuselage, has serious handicaps, and hence efforts to metalize the entire structure, including the surface, is being made.

In the two-spar biplane and braced monoplane wing structures, the ribs were the first elements to be metalized. This has been done by stamped sheet-metal ribs in the small sizes and by built-up framed ribs in the large sizes. Spars were next in development, and to date, dozens of different metal types, including the welded-steel truss spars, the stamped duralumin box, extruded duralumin I-beams and high-strength alloy-steel strip spars, have been made. During the earlier stages of the development difficulties were encountered in equaling the strength-to-weight efficiency of an equivalent wooden spar. Recently, however, government experiments have met with success and duralumin box spars superior to those of wood have been constructed.

In the conventional framed type of spar-and-rib structure the cover, for aerodynamic reasons, is generally made of fabric and, therefore, contributes no strength. It is improbable that a low-cambered biplane wing can be efficiently covered with metal. In the author's opinion, a metal cover to be efficient must carry stress, and for this reason the thicker monoplane, and especially cantilever wings, are most admirably suited. Inasmuch as a stiff cover is necessary for torsional strength in the cantilever wings, the metal type can be efficiently used.

These factors are bringing about a change from the determinant type of framed-wing structure to the "monolithic" type

in which the cover is stressed. A further structural step was to increase the number of spars, giving rise to so-called multi-spar construction. In this case "I," box, or tubular beams are used. In the absence of ribs, or closely spaced bulkheads, corrugated sheet must be used to attain the required rigidity between supports. A very efficient wing of this type is the Junkers in which a multiplicity of tubes acting as chord members and inter-connected by diagonal web braces is covered with corrugated skin riveted to the tubes.

A still different wing structure, which, while not necessarily of a metal type, is being metalized, is the so-called "mono-spar" wing in which a single spar is located some 30 to 40 per cent of the chord. This spar is designed to take only the bending moments. Diagonal wires helically disposed around the spar, or an entire sheet-metal covering on profile ribs, is designed to take the torsional loads into the fuselage. The wing profile is generally one with a very small center of pressure movement. This is an interesting development, and the author only questions its practicability from a vulnerability standpoint.



FIG. 3 THADEN T-4 ALL-METAL CANTILEVER MONOPLANE

Another type of stressed-wing development is that one utilizing a large metal box or "caisson" construction to which light leading and trailing edges are fastened. The box is composed of flat metal for the shear webs and a laminated top and bottom sheet-metal chord conforming to the wing profile. Frequent transverse bulkheads are spaced along the interior.

A further development of this box type of spar, has resulted in a so-called "monocoque" wing in which the box structure extends entirely around the leading edge of the wing and an appreciable distance back along the chord of the wing. The spar chord members are tapered in plane and profile as well as in gage, making it possible to taper the strength of the spar in proportion to the air loads.

Either corrugated or flat sheets may be used for its covering, although there is a decided advantage in the use of the corrugated sheet as it greatly strengthens the chord members against buckling by secondary failures. An example of this wing construction is shown in Fig. 2.

One of the least understood and yet probably one of the most important factors in sheet-metal structural design is that of stiffness. Unless the structural member, such as a beam, the surface of the fuselage, or wing, can be constrained to remain in the position for which it was designed, little or no efficiency will be realized. This is exemplified in the tests conducted by the Army and Navy on the strength of spars as used in the conventional two-spar wing. The greatest deficiency of most of these spars was in their lateral instability. Relatively light loads would cause them to fail through lateral deflection and consequently secondary stresses. In a like manner sheet covering, unless constrained, has practically no strength. Corrugating the skin aids to a remarkable degree in strengthening the member. In fact, any curvature of the sheet about the axis of the load increases its load-carrying capacity. It is not generally appreciated by designers that corrugated sheet can also withstand quite heavy shear loads both normal as well as parallel to the corrugations.

One of the most serious limitations in the use of sheet metal, and in fact any material, is imposed by the joints. Properly made joints always result in an increase of weight and a loss of efficiency. Welding is probably the quickest method of joining metal parts, but unfortunately it cannot be entirely relied upon when done under the stress of production, since the quality of work, in production, decreases. Furthermore, to obtain full strength it is necessary to heat-treat after welding, and this becomes impractical because of limitations of size and because of warping. In a sheet-metal structure, particularly when made of aluminum alloys, riveting and bolting are the most practical methods of bonding, and in this case the chief limitation lies in obtaining sufficient bearing areas.

Another factor, having little to do with aerodynamics or structures but frequently entirely neglected by designers, is that of inspectability of the interior of the structure. This is of particular importance in the "monolithic" type of structure and has been one of the obvious deficiencies of numerous structures of this type. It will be noted that in both the fuselage structure and the wing structure, shown in Figs. 1 and 2, the entire interiors of both of these structures may be inspected by removing the panels on the lower surface of the wing. The lack of any and all bracing in the fuselage makes it easy to inspect this part of the structure. These features have two-fold advantages; first in the initial manufacture and second in maintenance and repair work in the field.

To obtain satisfactory results in the building of these monocoque structures, it is practically essential to prepare jigs in which to do the work. It will frequently be found that such preparatory work is more economical in the long run.

As a concluding comment it might be well to point out that there is no one simple type of metal structure which will satisfy all of the structural problems of a single airplane, much less the problems involved in various sizes of airplanes. The relative size of each part must be correlated not only to satisfy the requirements for the most efficient structural and aerodynamic design but also to insure a design practicable from a production standpoint. In the present state of the art it is probably true that the greatest asset of a designer of metal airplanes is a flexible and open mind.

Discussion

JOSEPH S. NEWELL.[2] For some time previous to the first of this year the writer concurred with the author that the ultimate airplane would be all-metal. He still believes that it may in the somewhat distant future, but he is of the opinion that the present trend toward all-metal airplanes will be followed by a swing back to the conventional type of the past few years, at least for the less-expensive sport and general-utility categories. After learning that one air-transport company has found it necessary to dismantle all-metal flying boats completely after each 400 flying hours in order to inspect the connections of the main hull structure and to replace the corroded skin, one begins to wonder about the longevity and the "man hours of maintenance per flying hour" of all-metal structures. After seeing an all-metal military transport with a record of some 900 flying hours during its four years of service having a skin so badly corroded that one could force the thumb through it almost

[2] Associate Professor of Aero-Structural Engineering, Massachusetts Institute of Technology.

anywhere, one wonders still more concerning the life span of present-day all-metal aircraft, and their ability to withstand outside storage. The writer concedes that these airplanes both had plain aluminum-alloy coverings, not alclad, but both had protective coatings, and as both were maintained by large and efficient organizations it is only fair to assume that the protection afforded conformed with the best practice current during the life of the airplane. A number of similar cases could be cited, but these two suffice to make one ponder the statement recently made by the chief engineer of one of our most progressive airplane companies. It was, "For durability and dependability I'll have my all-metal airplanes made of wood."

As for fire resistance, all-metal aircraft doubtless afford a smaller hazard for ground or hangar fires, but with the low melting point of the aluminum alloys it is doubtful whether or not they afford any appreciable protection during crash fires which practically always involve the intense heat of burning gasoline and oil. In such fires, at least if one is to judge from the writer's experience, the aluminum alloy structure which has been in the range of the gasoline flames has always been melted or completely consumed, and has given no more evidence of its original shape than would a spruce structure. The steel members have always been essentially intact, but one does not consider all-steel structures when speaking of all-metal airplanes. Not at the present time at least, when designers are laboring on stressed-skin or monocoque projects.

The mass production of aircraft is a thing of the future, probably 10 years at least, and its apparent advantages are to be closely scanned before being accepted as actual. For instance, it has long been assumed that wing ribs would afford an ideal subject for mass-production methods, and, with any reasonable number of airplanes involved, these methods could best be applied to stamped metal ribs. However, after ascertaining that one manufacturer of wood ribs stood ready to build and deliver sufficient ribs for more than 500 airplanes at a total cost approximately equaling the expense of making dies for stamping out an equal number of metal ribs, one wonders just where the advantages of mass production start. For the next few years, at least, it is probable that an output of 500 identical airplanes will be considered mass production for the aircraft industry regardless of size, type, or conditions of service. Analyses and comparisons based on mass-production methods in the automobile industry, where a production limited to 500 identical units a week is considered practically the equivalent of an admission that a manufacturer's existence is economically unjustifiable, must therefore be used with discretion. It appears at present to be better practice to utilize existing production machinery and design to reduce fabrication costs to a minimum rather than use special machines involving a capital investment which, in most cases, is disproportionately large compared with the "possible savings" involved.

For all-metal airplanes of the stressed-skin type it is probable that the aluminum alloys afford the best structural materials from the standpoints of strength to weight ratio and of workability. For such types, too, it is probable that the monoplane is superior to the biplane, largely because of its simpler construction and the reduction in number of constituent parts. It is doubted that an internally braced, all-metal wing can be built to have satisfactory strength and not weigh over $1^3/_4$ lb. per sq. ft. This is possible with an internally braced, spruce spar, fabric-covered type on airplanes of around 4000 lb. gross weight. The latter type may be built with no danger of flutter for speeds up to 300 m.p.h. if suitable precautions are taken, and it is interesting to note that practically the same precautions are necessary to eliminate this danger in the stressed-skin type, either plywood covered or all-metal. So it may be inferred that, for equal strengths, the all-metal wing will be heavier than the conventional two-spar, fabric-covered variety, at least for the smaller commercial types.

As for fuselages the writer has during the past two months discussed weight-saving prospects with some five or six competent engineers, and each has said he believed the all-metal fuselage could be built lighter than fabric-covered welded-steel frames, but where direct comparisons were possible each had so far found an actually greater weight in the all-metal construction. This is doubtless due to the fact that data on various types are so meager.

Several manufacturers have apparently built and tested aluminum-alloy fuselages and have back-figured the stresses occurring at failure using the standard beam formula, $f = Mc/I$, only to clothe their results with the greatest secrecy. In some half dozen cases the writer found that failure of the stiffened skin or semimonocoque fuselages occurred when the stress on the extreme fiber in compression was somewhere between 8500 and 9000 lb. per sq. in. at the critical section. The cross-sections were elliptical in shape, the moment of inertia being computed for the skin and longitudinal stiffeners combined. One or two values around 7500 lb. per sq. in. were noted, and there were a couple near 10,000 lb. per sq. in., but 8500 lb. per sq. in. appears to be a fair ultimate stress for preliminary designing at present. It is hoped that extensive research programs will soon provide definite values and will afford more rational methods of proportioning both stiffeners and skin. There are too few dependable data available at present to justify establishing any method of analysis or any set of allowable stress properties for flat sheet or corrugated covering. Such results as are available seem to indicate that analyses involving highly mathematical investigations or allowable stress properties based on the theories of applied elasticity should be confirmed by actual tests on full-size specimens in every case. For instance, theory indicates that a flat sheet 24 in. square should carry about one-sixth of the load which a 4 by 24 in. specimen would take assuming that the sheet is loaded in compression, that its column length is 24 in. in each case, and that the edges parallel to the direction of the load are stiffened to prevent localized buckling, but that either edge may rotate about an axis in the plane of the sheet and lying along the edge in question. Theory and practice appear to agree that the sheets will buckle into "squares," i.e., the 24 by 24 in. sheet will buckle to form a single square, the 4 by 24 in. specimen will buckle to form six, 4-in. squares. But as indicated by recent tests at the Bureau of Standards, the results of which are to be published shortly by the National Advisory Committee for Aeronautics, both sheets will carry the same load for stresses causing permanent set. Sufficient tests have been made by the bureau on different gages, widths, and materials to establish this rather serious discrepancy between elastic theory and practice. It appears necessary, therefore, to static-test all-metal structures for some time to come, and to coordinate and study the results with a view toward establishing workable design values.

The writer wishes to protest the introduction of "monolith" to describe all-metal airplane structures. It is true that monolith means single stone. But is it not probable that the average person will associate the characteristics of a stone with what might be described as the "falling tendency" of an airplane rather than with the integrity of its structure? It would seem that monocoque, meaning single shell, is the more logical and more satisfactory word for structures of the type in question.

In closing, the writer wishes to congratulate the author on his excellent paper. It affords a clear, rational, and well-arranged presentation of the problems of the manufacturer of all-metal airplanes, a presentation that is particularly valuable

at this time when all-metal types are being considered by a number of manufacturers as a possible cure-all for the present ills of the industry. That the writer has presented his comments from a different point of view is not to be construed as a disagreement with the author's excellent discussion, but rather as an interjection of the more somber aspects of the problem of metal aircraft construction and operation in the hope that the less experienced manufacturers may be constrained to consider the possibility that a remedy so simple and yet so powerful does not exist.

AER-52-23

Some Fundamental Economics of Aircraft Operation

By R. H. UPSON,[1] DETROIT, MICH.

The paper is intended as a sequel to the author's paper entitled, "The Airship for Long-Haul Heavy-Traffic Service," presented at the annual meeting of the American Society of Mechanical Engineers in 1922. In that paper the airship was considered mainly as an ocean-going vessel, the principal comparison being made with ocean steamships. Various design problems were also discussed and the desirability of certain improvements pointed out. The situation needs reviewing on account of the sensational bid for transoceanic service that has subsequently been made by the airplane and the fact that many of the improvements indicated as desirable in the previous paper have now been demonstrated in experimental form. Hence, this paper is devoted principally to the consideration of the relative economic fields of airships and airplanes, the attempt being made to deal primarily with fundamental characteristics rather than with current performance records.

IT SEEMS to be generally recognized that in comparison with the airplane the airship is best suited for operation over long distances. But any comparison to have much permanent value must be quantitative in character, and it must be based on characteristics more fundamental than mere types and performances of the day. In a paper given to the A.S.M.E. nearly seven years ago,[2] the author felt that it was most conservative to base comparisons on actual performances, there being insufficient aerodynamic knowledge for anything better. Now, however, the accumulation of numerous tests and their reliable interpretation by the methods of Prandtl, Glauert, and others make it possible to depend more on the real fundamentals.

Let it be assumed that the main purpose of either airship or airplane is to carry a certain load over a certain distance at a certain speed. Then the fundamental distinguishing feature of each type of craft is its lifting unit—namely, the wings for the airplane and the gas hull for the airship; and the fundamental items of performance for both are lift, drag, and speed. As the fuel consumed over a given distance is proportional to the drag, the distance pertaining to a given fuel load (as a proportion of lift) varies as lift divided by drag (L/D). Without going into detail in this paper (this having been fully covered in another paper[3]), the author will simply select the best-tapered internally braced airplane wing and the best airship hull, the latter with a drag practically equal to skin friction alone. (See Appendix.)

As a tentative approach, the airplane was assumed at constant lift-coefficient and varied in aspect and thickness ratio in such a way as to keep the weight of its main beam flanges increasing in the same proportion as those of the airship for increasing gross lifts.

As might be expected, in spite of the lower coefficient of skin friction applying to the larger sizes of airplanes, the structural requirements caused the actual L/D to drop slightly with increasing size. Therefore, the charts here shown have been drawn up on the basis of comparing a single airship hull of varying size with a multiplicity of airfoil elements at constant L/D. Assuming a minimum parasite drag of 33 per cent of the total (a figure not yet attainable except for gliders), it is equivalent to an L/D of 20 for the complete airplane.[3]

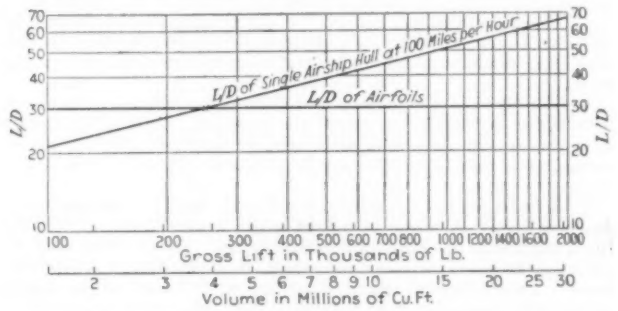


FIG. 1 COMPARISON OF L/D AT CONSTANT SPEED

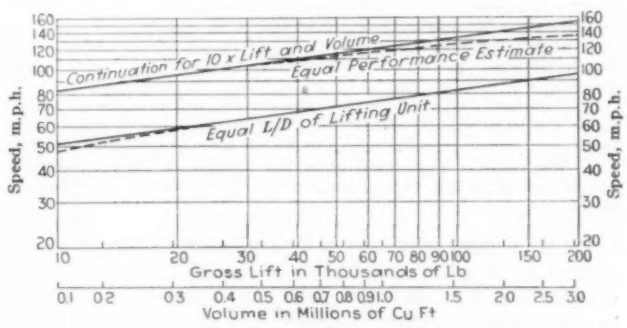


FIG. 2 SPEED OF EQUAL L/D

Fig. 1 shows the variation of max. L/D for the one speed of 100 m.p.h. This is the operating speed, the top speed being, say, 20 per cent higher for airplanes or 10 per cent for airships. Where these two curves intersect is the point where the lift, drag, and speed are the same for both airfoils and airship hull. If similar intersection points are calculated for other speeds,

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[2] "The Airship for Long-Haul Heavy-Traffic Service," *Mechanical Engineering*, April, 1923.

Contributed by the Aeronautic Division and presented at the Akron Meeting, October 21 to 23, 1929, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] "Wings—A Coordinated System of Basic Design," *Journal of the S.A.E.*, January, 1930, p. 15.

there is obtained the full line in Fig. 2 showing the speed of equal or constant L/D for different sizes of airship hull. This forms the most fundamental line of separation between the operating fields of airplanes and airships.

Of course, this theoretical curve cannot be taken as a complete answer to the question or as a substitute for good judgment. Its purpose is to make at least one important phase of the problem definite as a foundation on which to base that necessarily vague and uncertain thing called "judgment."

From a directly practical standpoint, the interest is not so much in the L/D of the lifting unit as in the ratio of useful load (exclusive of crew and ballast) to the total drag of the complete unit, which for a given speed and distance determines the pay-load capacity. The author's estimate of the curve of equal performance on the latter basis, for the best possible construction of both types, is shown by the dashed-line modification in Fig. 2. It is based on the following considerations of conditions along the equal L/D curve for both airships and airplanes:

1 The power plants are equal, due to the assumption of equal total drag and speed. This also balances out the item of distance except as it is reflected in the size and speed.

2 The stabilizing and control organs, including supporting structure, are somewhat larger in proportion for planes, but in each case their drag tends to remain a substantially constant fraction of the total.

3 The effect of size alone is to reduce the structural efficiency of both types as the size increases, the airplane a little more rapidly than the airship. But comparing the single airship to a group of planes, as is done here, the handicap of size applies to the airship alone.

4 A large part of the airship's structure, used for resisting aerodynamic forces, increases in weight as the square of the speed. On the other hand, the structural weight of the airplane is actually reduced as the speed increases, providing the lift coefficient is kept constant. Thus in larger sizes the structural efficiency of the airship suffers more markedly in proportion to the plane, not only because of the size, but because of the increasing speed.

5 The airship has an added weight of gas-holding material which is especially high in proportion for small sizes, but it is also serious for large sizes due to the apparent necessity for compartmenting.

6 Space requirements of airplanes necessarily involve more parasite resistance, especially in the smaller sizes and for the higher speeds.

7 A certain amount of water-recovery apparatus is almost necessary for a large airship, but on the other hand the use of some of the hydrogen (or fuel gas) for power tends to offset the weight involved.

8 The landing equipment of planes is more of an item in both weight and resistance than is the landing and mooring equipment of airships.

For the highest of the speeds here considered (134 m.p.h.) the assumption of a lift coefficient $C_L = 0.5$ for planes makes the wing loading 23 lb. per sq. ft., which would involve an excessive landing speed according to present American practice; but it could probably be brought within reason by a suitable use of flaps, slots, etc., which in turn must be allowed for in the weight and drag.

The whole subject of landing gear, including floats, for very large planes is a matter of considerable conjecture, and is likely to remain so, as there seems to be no fundamental factor to justify the building of sizes up to anywhere near the limit of this chart. As before noted, the "equal performance" curve represents the comparison of any large single airship with any group of medium-sized planes totaling the same carrying capacity.

The net efficiency of the airship will be considerably decreased if helium instead of hydrogen is used, but it will be increased if the aerodynamic lift can be utilized. That of the airplanes will be decreased if the wing loading is decreased. The airship, on the other hand, has certain special items of upkeep expense of which gas and gas-holding materials are the most important. The initial cost per unit carrying capacity, though much lower for the airplanes in small sizes, becomes more nearly equal in large sizes. Generally speaking, maintenance of route is more expensive for the planes, and terminals for the airship. For any specific use there are special factors to be taken into account.

The author's conclusions are:

(*a*) Airships must be used mainly for conditions below the *equal-performance line* if they are to be worth while. This applies to practically any use, military or commercial. For the latter use the *equal-economy line* would be a little further below, especially for small sizes.

(*b*) Even through a considerable range below the line, airships will be hard pressed to show an economic advantage if they are unduly burdened with extra "*parasite*" *expense* such as for helium, hangars, ground crews, perishable materials, unnecessary weight and resistance, etc. The reverse, of course, applies in so far as the airplane has to be held back by facilities such as for slow landing and seaworthiness. Some of these items are of course very good in themselves, but the extent to which they may be profitably used must be determined in every case by careful consideration of the economics.

(*c*) In view of the *meteorological conditions* prevailing over this country and the adjacent oceans, and the other means of transportation available, it is hard to believe that any aircraft with a top speed much under 100 m.p.h. can have more than a very limited use.

(*d*) From the foregoing considerations, an airship of somewhere between three and four million cubic feet seems to be about the *minimum profitable size* for commercial transport, and then only when gas and other important factors are economically favorable. (The *Graf Zeppelin*, certainly the best airship thoroughly tested to date, does not, the author believes, meet the requirements.)

(*e*) *The longer the trip, the bigger the ship.* There continues to be some gain in transport efficiency over transatlantic distances for airship sizes up to at least 30,000,000 cu. ft. But the law of diminishing returns is working so hard there that one would hardly advocate anything beyond 20,000,000 cu. ft. with the present available materials; and it would be foolish to try anything that large without further improvements in construction and more experience in operation.

(*f*) There is little change in basic airfoil efficiency for moderate changes of either *size or speed*. Thus the only object in making airplanes larger is to gain certain secondary advantages such as cutting down parasite resistance, distributing the load more equally, and gaining certain qualities of seaworthiness. Speed, however, is a great object in itself. For this reason, it is believed that airplanes will be increasingly used over nearly all routes that they can negotiate with a reasonable pay load (or military load) and at a speed probably well above the equal performance curve.

(*g*) Small airships seem to have a certain legitimate utility, for training and other special purposes, but not even that can be said for very large airplanes. It may be worth while to build planes up to the size where most useful load can be carried in the wings, or, say, 10,000 sq. ft. maximum wing area. This corresponds in lift to an airship of about 2,600,000 cu. ft. Thus the *main field of the airship* will probably begin beyond the size where the airplane, as an individual unit, leaves off.

(*h*) The dividing point in *actual distance* is harder to assign because it depends on so many other factors. For straight transportation the airplane seems generally to be more efficient than the airship up to at least 1000 miles, assuming the same relative perfection of engineering. (See Appendix.) On the other hand, no matter what the state of the art, there always will be a field for the airship beyond a certain distance, where by a suitable lowering of the speed, almost any desired L/D ratio may be had. For example, even without any further improvement there is a distinct field for the large airship in long-range naval scouting.

(*i*) At the present time, also, airships seem to be *safer* for flights at night, over rough country or long stretches of water, and in fog. Airplanes are probably safer in very rough air and near electrical disturbances (at least as compared to hydrogen-filled fabric ships).

(*j*) In respect to construction there are four main elements on which substantial progress has already been made, but which urgently need continuing attention to permit the fullest realization of aircraft possibilities, as follows:

		Airplane	Airship
1	*Form*	Highly tapered cantilever wings	Low fineness ratio (preferably not over 4.5)
2	*Material*	Substantially all-metal construction for both types, with special emphasis on economy of production	
3	*Strength*	Elimination of flutter	Greater strength and rigidity in rough air
4	*Control*	Elimination of spinning, and more positive control of landing	Positive stability, when flying and when moored

All progress has its price. Fortunately in the airplane field there have been in the past no lack of men willing to pay the price; but although development has been rapid there is still a long way to go—a fact not generally realized. Airship development, handicapped inherently by the large units involved, has been further restricted by the very limited number of organizations actively interested. The recently demonstrated Metalclad airship (built by the Detroit Aircraft Corporation for the U. S. Navy) is a significant step, but it is only a step, as it will have to be developed to much larger sizes to be commercially available.

With all the present interest in aircraft production and operation, it is to be hoped that more people will realize the dominating importance of continuing research and development along sound engineering lines, which may as well be called sound business lines, because it is the kind of business on which the entire future of the industry depends.

Appendix

Lift and Drag of Airship Hulls. Assume hydrogen gas with a mean unit lift of 0.065 lb. per cu. ft.; then the gross lift $L = 0.065$ vol. From miscellaneous tests in the N.A.C.A. variable density tunnel, the drag coefficient of the best airship hull may be closely approximated for varying size and speed by the formula:

$$C_s = 0.222R^{-0.15}$$

where R is the Reynolds number $= 9330\, v\, \text{vol.}^{1/3}$; v = speed in miles per hour; vol. is in cu. ft.

Then the actual drag

$$D = C_s\, q\, \text{vol.}^{2/3}$$

$$= 0.000144\, \text{vol.}^{0.617}\, v^{1.85}$$

Then $L/D = \dfrac{450\ \text{vol.}^{0.383}}{v^{1.85}}$

Lift and Drag of Airfoils. The general formula[3] for airfoil drag coefficient is:

$$C_D = (0.007 + 0.1t^2)\,(1 + 0.8C^2_L) + \frac{C^2_L(1 + \delta)}{\pi\, n}$$

where t is the thickness ratio; n is the aspect ratio; δ averages about 0.04.

The foregoing is for a Reynolds number of about 3,500,000.

For large changes in size, it is probable that the first coefficient (0.007), and perhaps the entire first term, should be taken to vary as $R^{-.15}$, the skin friction factor. Also, to keep on a comparative basis with airship hulls, either the wing loading W/S or the structural ratio n/t, or both, must be varied in such a way as to keep corresponding proportions of structural weight with increasing size.

If the beam-flange weight longitudinals w is taken as the criterion, it may be easily shown that for airship $w \propto W^{4/3}$, where W is the gross weight or lift; and for airplanes $w \propto W\,\dfrac{b^2}{a} \propto W\,\dfrac{n}{t}\sqrt{Sn}$, where b is the span and a is the effective wing depth.

Then if the airplane w is to vary as airship w, for equal gross weight, the condition is:

$$\frac{W}{S}\,\frac{n^{3/2}}{t}\,S^{3/2} = W^{4/3}$$

where the wing loading $\dfrac{W}{S} = \dfrac{v^2 C_L}{391}$ (v in miles per hour).

If C_L is assumed constant, this reduces to:

$$v^2\,\frac{n^{3/2}}{t}\,S^{3/2} \propto W^{4/3}$$

Even allowing for the decreased skin friction of larger sizes and a small allowance for decreased load factors and improved load distribution, the resulting L/D curve is still found to drop slightly with increasing size. Hence, it seems preferable to make the comparison on the basis of a fixed, constant L/D for airfoils. As shown,[3] the maximum L/D of "structural airfols," cantilever type, will run from about 25 to a maximum of probably 32, depending on the degree of taper and general structural efficiency.

Comparison of Airfoils and Airship Hulls. Taking for further study an airplane L/D of 30 as definitely attainable, the dividing point then between airplanes and airships will be for conditions which make the airship L/D equal to 30, or

$$v^{1.85} = 15\ \text{vol.}^{0.383}$$

from which

$$v(\text{miles per hour}) = 4.33\ \text{vol.}^{0.207}$$

This is plotted in Fig. 1.

Assume 10 per cent of the gross lift devoted to fuel = F (exclusive of reserve). Then $F/D = 3$.

Let distance = x (miles). As one hp-hr. = $33{,}000 \times 60 = 1{,}980{,}000$ ft-lb.

Fuel consumption $= \dfrac{0.5}{0.75 = 1{,}980{,}000}$ lb. per ft-lb.

assuming 0.5 lb. per hp-hr. and 0.75 propeller efficiency; then the total fuel $= F = 3D = \dfrac{5280Dx \times 0.5}{0.75 \times 1{,}980{,}000}$

or

$$x = \frac{3 \times 0.75 \times 1{,}980{,}000}{5280 \times 0.5} = 1690 \text{ miles, or } 1130 \text{ miles if } L/D = 20.$$

AER-52-24

Practical Airplane Performance Calculations

By DR. MICHAEL WATTER,[1] HARTFORD, CONN.

A design must meet the characteristics set forth in the specification if it is to warrant construction. Performance characteristics represent a most important requirement, and there are today many methods and theories of performance calculation.

The conclusion was reached to obtain simplicity even at the cost of scientific accuracy, so long as the calculations gave a correct answer. The method described is based on the evaluation of parasite resistance, wing or cell characteristics, and knowledge of engine-propeller behavior. It is intended for routine calculations, but may be extended for special cases.



TO FULFIL its purpose and to justify its construction, any given design must meet the characteristics set forth in the specification. This specification may represent the requirements of the customer or the objective which the designer must meet in order to compete successfully with other similar types in existence or in course of construction. The performance characteristics represent one of the most important requirements of an aircraft, and it is not surprising, therefore, that there are today numerous methods and theories of performance calculation. While accuracy is of paramount importance, the time factor cannot be ignored, because in a number of instances the designer is required to supply this information in comparatively short time and also to be able to revise the data for small changes in design which may be specified or found desirable. As there have been several excellent methods evolved, and numerous papers have been written on various problems of performance calculations, it therefore is with a certain hesitation that the author presents this paper.

The long-debated problems of slipstream effect, influence of the Reynolds number, coordination of wind-tunnel and free-flight data, as well as recent tests in the N.A.C.A. density and 20-ft. wind tunnels, were considered by the author. The conclusion was reached to obtain simplicity even at the cost of scientific accuracy, so long as the calculations gave a correct answer. In engineering investigations this is an age-long procedure, since all know that most of the structural problems are treated in exactly this way. The problem of strength of beams is solved by assuming that the relationship of

$$M = EI \frac{d^2y}{dx^2}$$

holds true past the elastic limit of the material, which while not being scientifically correct leads to a very easy and accurate way of proportioning the beam sizes by use of experimental values of allowable fiber stresses. Similarly, when confronted with the complexity of the aerodynamic phenomenon which takes place in flight, it is almost hopeless to attempt to evaluate all the mutual interactions and interferences when one considers how comparatively little is known of the individual behavior of wings, parts having parasite resistance, propeller and engine unit, etc.

Brief Summary of the Method

The method described is based on the evaluation of parasite resistance, wing or cell characteristics, and knowledge of engine-propeller behavior. It is intended for routine practical performance calculations, but can be extended for special cases and evaluation of influence of small changes in design.

Certain coefficients and laws of variation of resistance with change in angle of attack were obtained by the author from back-figuring the actual free-flight test of airplanes the characteristics and design of which were known. Other coefficients were obtained from different sources, and credit is given in the text.

In order to make a performance calculation, it is necessary to have the three-view drawing of the airplane, with sizes and shapes of the fuselage, hulls, floats, nacelles, engine, and radiators, and information in regard to struts, wires, wheels, armament, type of fittings, and as many of the details as it is practical to obtain—the brake-horsepower curve versus the revolutions per minute of the engine, the isolated wing characteristics, arrangement of the wing cell, and the following general data:

VL at which the isolated parts were tested
Cross-sectional areas of parts having parasite resistance
Areas of the wings
Gap
Stagger
Sweepback
Decalage
Spans
Chords
Angles of fixed incidence
Area of horizontal tail
Horizontal tail setting (can be assumed from available data)
Vertical tail area
Gross flying weight
Amount of fuel carried

The performance calculation usually consists in obtaining the following data:

Maximum speeds at sea level and specified altitudes
Stalling speed at sea level and specified altitudes
Service ceiling
Rate of climb at sea level and specified altitudes
Climb in 10 minutes
Time to climb to various altitudes
Best speeds of climb
Endurance at full throttle
Range at full throttle
Endurance at maximum range
Range, maximum

Assumptions

Before proceeding with the detailed description of the method, it will not be amiss to state and to justify certain omissions and simplifications used.

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Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

It was a standard practice in the past to take into account the slipstream effect, and certain well-known and authoritative books on aerodynamics give various methods of evaluating the influence of slipstream. The author, however, always has believed that the inclusion of the slipstream effect in performance calculations leads to an unnecessary complication without adding to the accuracy of results, as it, if anything, increases the chances of possible errors. In an article entitled "Slipstream Effect" which appeared in the February, 1928, issue of *The Aero Digest*, the author reviewed the suggested corrections for slipstream effect from a few representative articles. The following statements and conclusions are taken from that article:

The opinions of the experts on the subject are far from unanimous; the inclusion of slipstream effect in performance analysis is considered by some as indispensable for any degree of accuracy, by others as desirable for greater accuracy, while still others think it superfluous because of other uncertainties and assumptions used in theoretical calculations.

There are a few points on which all seem to agree:

(1) The slipstream effect introduces an error in level-flight analysis of such small magnitude as to be negligible.

(2) With the usual engine-propeller group, the maximum speeds can be predicted with the desired degree of accuracy neglecting the slipstream effect.

(3) The slipstream effect causes comparatively greater error in climb. The magnitude of this error, however, is difficult to ascertain, and there are other factors which may influence the analysis and yet be confused with the slipstream effect.

The slipstream effect can be considered as mutual interaction of the objects in the slipstream and the propeller proper. This mutual interaction must be considered as a play of internal forces resulting in changed condition of flow and propeller behavior and can be taken as consisting of two effects:

(1) Propeller-slip effect due to increased velocity of the air after it passes the propeller disk. The result of this is an increase in forces on the objects which are in the propeller's slipstream. This effect, however, cannot be evaluated by the customary law of V^2 for the reason given below.

(2) Obstruction effect. Since the propeller is not working in an unobstructed air, the objects in its slipstream have an effect equivalent to slowing the air down.

It must be remembered, however, that since the slipstream effect is an internal phenomenon, the final effect should be judged by combining (1) and (2).

Conclusions. There is a great deal more of experimental data and theoretical investigation published on the subject, but it is hoped that the material reproduced will bring forth some of the more important questions; namely: (1) the complexity of the problem, (2) absence of a satisfactory theoretical solution, (3) variation in experimental results. These points alone would not justify to disregard the slipstream, but the importance of the problem be rather weighed on the basis of quantitative considerations. Since it seems to be agreed by every one that the slipstream effect has little importance for level-flight condition, the problem is limited to the climbing speeds alone. Part of the resistance usually affected by the slipstream (fuselage, engine, radiators, landing gear, struts, wires, etc.) for small angles of attack constitutes 35 to 55 per cent of the total drag, while at climbing angles the drag of the wings becomes predominating, and that part of the parasite resistance seldom exceeds 35 per cent of the total drag of the machine. This explains why even a comparatively large error which the experiments do not seem to reveal would have considerably smaller importance when one considers complete drag of the airplane. Taking into consideration other numerous uncertainties, such as mutual interferences, assigned resistance coefficients, VL effect, propeller characteristics, brake-horsepower of the engine, etc., it seems that undue importance is stressed for the slipstream correction. It would seem that the same end, greater accuracy of results, can be achieved in case of standard designs by using proper values of parasite resistance and by assuming such a law of variation of the parasite resistance with change in angle of attack as to compensate for the probable error due to neglected slipstream effect.

Incidentally, this method reduces the difficulties to assigning proper resistance coefficients and a certain law of variation of parasite resistance, while an attempt to correct for slipstream involves further assumptions as to what expression to use for R/Ro and what is the magnitude of the obstruction effect.

It may be added that in climb the slipstream effect is further offset by the presence of the upward component of thrust, which at times, for airplanes capable of steep climbing angles, is somewhat appreciable. In view of the foregoing, this is also disregarded in the present method of performance calculations.

Characteristic Curves, Airplane Polar

Evaluation of airplane characteristics constitutes the most important part of the performance estimate, requiring not only knowledge but judgment as well. At this place it may be stated that the method described in this paper is fundamentally a wind-tunnel method. Because of the author's wind-tunnel experience, combined with performance calculations based on atmospheric wind-tunnel airplane characteristics, it was thought best to attempt to obtain a close agreement between calculated and tested polars. In addition to being a convenient result of author's experience, this method of approach was considered desirable also because it led to a unified method of performance calculation, whether based on theoretical calculations or wind-tunnel tests, and also the accuracy of the preliminary performance of a new design could be judged by how well the calculated polar checked the one obtained from a wind-tunnel model in later stages of the design.

The calculation of characteristics of an airplane consists in the following:

(1) Cell or wing characteristics from wind-tunnel data on isolated airfoil of standard aspect ratio.

(2) Estimate of parasite resistance.

(3) Variation of parasite resistance with change in angle of attack.

(4) Resistance due to horizontal tail area—may be included in (3).

(5) Lift, drag, and L/D on full-size model at 1 m.p.h.

Wing or Cell Characteristics

For the best results, it seems desirable to use characteristics of an isolated airfoil tested at or reduced to a $VL = 25$ to 30 (where V is expressed in feet per second and L in feet). This is equivalent to the standard size of airfoils (5 in. by 30 in. or 6 in. by 36 in.) tested at 40 m.p.h. For 7-ft. or larger wind tunnels the wall effect can be disregarded. Should the model be tested at a different VL, the experimental data must be corrected, and for this purpose one may use the following formulas obtained by Diehl:

Drag Coefficient Correction:

$$C_{D_2} = C_{D_1} - C_{D_m}\left[1 - \left(\frac{V_2L_2}{V_1L_1}\right)^{-0.18}\right]$$

where C_{D_1} = the drag coefficient at a given angle of attack and V_1L_1

C_{D_m} = the minimum drag coefficient at V_1L_1

C_{D_2} = the drag coefficient at the same angle of attack and V_2L_2

Lift Coefficient Correction:

$$C_{L_2} = C_{L_1} + 0.114 \log_{10}\left(\frac{V_2L_2}{V_1L_1}\right)$$

where C_{L_1} = the lift coefficient at V_1L_1 for the particular angle of attack under consideration.

The foregoing formula refers to absolute coefficients, and in engineering units it becomes

$$K_{y_2} = K_{y_1} + 0.000292 \log_{10}\left(\frac{V_2L_2}{V_1L_1}\right)$$

The use of these formulas is limited to a comparatively small range of VL.

Having reduced the characteristics of isolated airfoils to the same VL and which VL lies within the range of 25 to 30 suggested, one may correct the wing characteristics for aspect ratio, interferences, and arrangement. Corrections for interferences, while of the utmost importance, can be estimated only approximately on the basis of tests on similar arrangements, while those due to the aspect ratio and biplane arrangement can be made with considerable accuracy by any of the known methods.

Aspect Ratio Correction. Monoplane correction is made for change in induced drag and induced angle of attack for a given lift coefficient.

Aspect ratio $= S/b^2$, where S is the area and b the span

$C_{L_1} = C_{L_2}$

$$C_{D2} = C_{D1} + \frac{C_L^2}{\pi}\left[\frac{S_2}{b_2^2} - \frac{S_1}{b_1^2}\right]$$

$$\alpha_2 = \alpha_1 + \frac{57.3C_L}{\pi}\left[\frac{S_2}{b_2^2} - \frac{S_1}{b_1^2}\right]$$

where subscripts (1) and (2) denote the two aspect ratios.

Biplane (Diehl). In case of a biplane, the aspect ratio can be expressed as

$$\frac{S}{(Kb_1)^2}$$

where K is a certain coefficient variable with gap-to-span ratio, the ratio of spans, and distribution of wing areas.

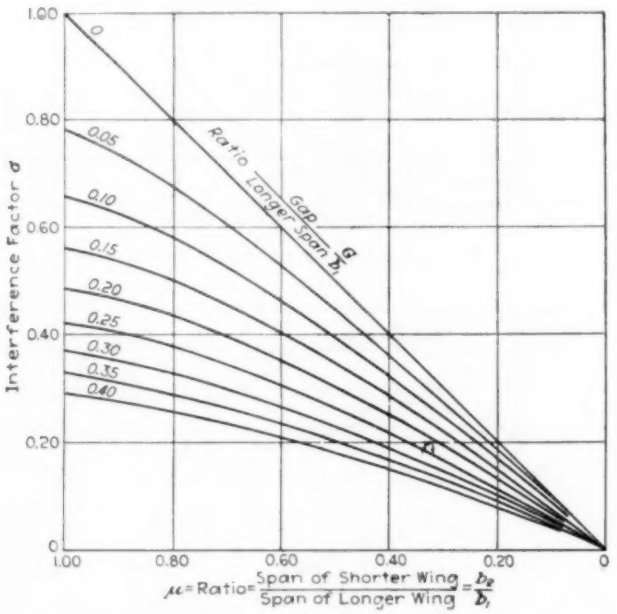


FIG. 1 PRANDTL'S DRAG INTERFERENCE FACTOR FOR BIPLANES IN TERMS OF MAXIMUM SPAN

Denoting:

b_1 = span of longer wing
b_2 = span of shorter wing
μ = b_2/b_1 = ratio of shorter span to longer
$L_1 \cong S_1$ = lift or area of longer wing
$L_2 \cong S_2$ = lift or area of shorter wing
L = $L_1 + L_2 = S_1 + S_2$ = total lift or area
G = interference factor

Then K may be expressed as

$$K = \left[\frac{\mu^2}{r^2(\mu^2 - 2\mu\sigma + 1) + 2r(\mu\sigma - 1) + 1}\right]^{\frac{1}{2}}$$

where $r = S_1/(S_1 + S_2)$ and σ can be obtained from Fig. 1. A more detailed explanation of the induced drag of multiplanes and calculations of equivalent monoplane aspect ratio can be found in Diehl's book, "Engineering Aerodynamics." Appendix 1 gives Diehl's curves, which facilitate finding of K.

Biplane Correction. In case of a biplane cell it is necessary to take into account the mutual interference of airfoils, and there are several excellent methods available. For a detailed discussion the reference is made to the National Advisory Committee Technical Report No. 151, "General Biplane Theory," by Max M. Munk, and "Engineering Aerodynamics," by Walter S. Diehl.

Munk's Method. Dr. Munk's method consists in evaluating the characteristics of a biplane cell by correcting the drag of an isolated wing because of additional induced drag and changing

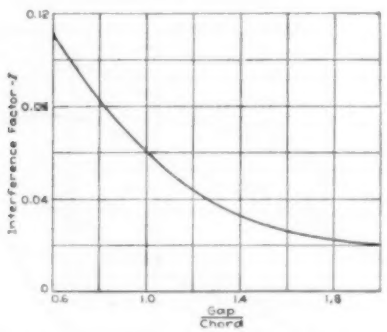


FIG. 2 MUNK'S INTERFERENCE FACTOR FOR INDUCED ANGLE OF ATTACK

the angle of attack by the amount of the induced angle of attack and the additional interference angle of attack.

$$C_{LB} = C_{Lm}$$

$$C_{DB} = C_{Dm} + \frac{C_L^2}{\pi}\left[\frac{S_B}{(Kb)^2} - \frac{S_m}{b^2_m}\right]$$

$$\alpha_B = \alpha_m + \frac{57.3C_L}{\pi}\left[\left(\frac{S_B}{K^2b^2} + I\right) - \frac{S_m}{b^2_m}\right]$$

where I is the interference factor (see Fig. 8).

In the foregoing formulas S/K^2b^2 is the equivalent monoplane aspect ratio for a given biplane. The coefficient K is obtained as explained.

BIPLANE AND TRIPLANE CORRECTION

The literature on triplane and multiplane correction is scarce, and although there does not appear to be any particular need for it, it seems advisable to present briefly a method of biplane and triplane correction evolved by H. Blasius and H. Hamburger which appeared in *Technische Berichte*, Vol. 2, part 2, 1918.

In the author's experience, the Blasius and Hamburger method proved quite helpful for cases of cells with decalage, and on two occasions when this method was used the calculated characteristics of the airplane were confirmed with surprising accuracy by wind-tunnel tests.

The fact that the numerical coefficients are based on profiles tested back in 1917–1918, which are somewhat different from the profiles used today, does not detract from the value of the method, because quantitative discrepancy is not great.

Biplane Formulas:

C_{LU}, C_{LL}, C_{DU}, C_{DL} = lift and drag coefficients, the second subscript indicating upper or lower wing
ΔC_{LU}, ΔC_{LL}, ΔC_{DU}, ΔC_{DL} = the change in coefficients resulting from the mutual interference

C_L, C_D = lift and drag coefficients of the cell
S_U, S_L = wing areas
b_U, b_L = spans
t_U, t_L = chords
G = gap
α_U, α_L = angles of attack (referred to undisturbed air)
β = stagger in degrees

Formulas:

$$\Delta C_{LU} = \frac{C_{LL} l_L}{b_U} \frac{(b_U + b_L)}{2G} \varphi_{LU}$$

$$C_{LL} = \frac{C_{LU} l_U}{b_L} \frac{(b_U + b_L)}{2G} \varphi_{LL}$$

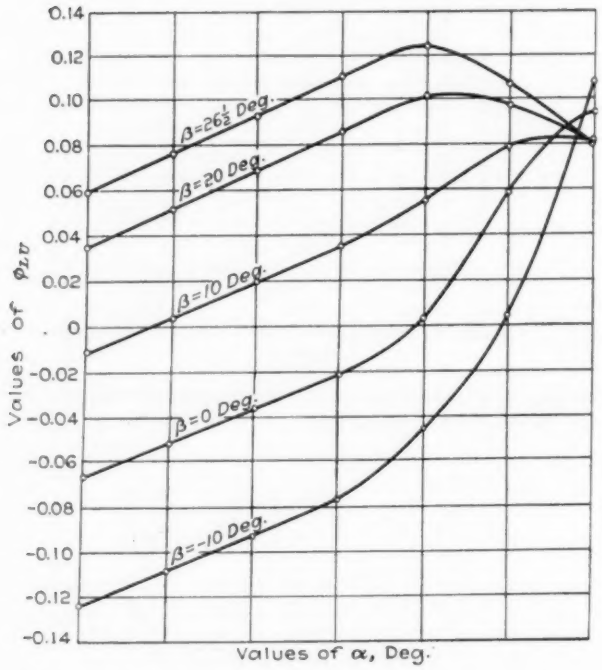


FIG. 3 VALUES OF φ_L FOR BIPLANES

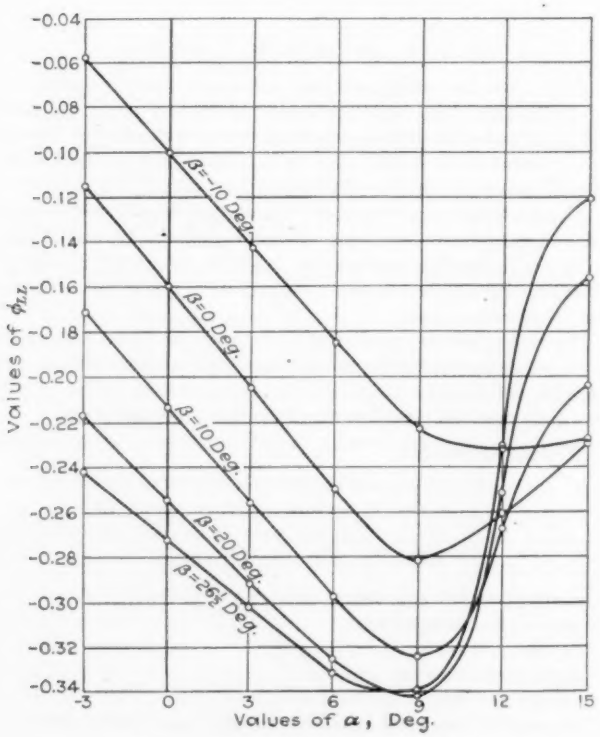


FIG. 4 VALUES OF φ_L FOR BIPLANES

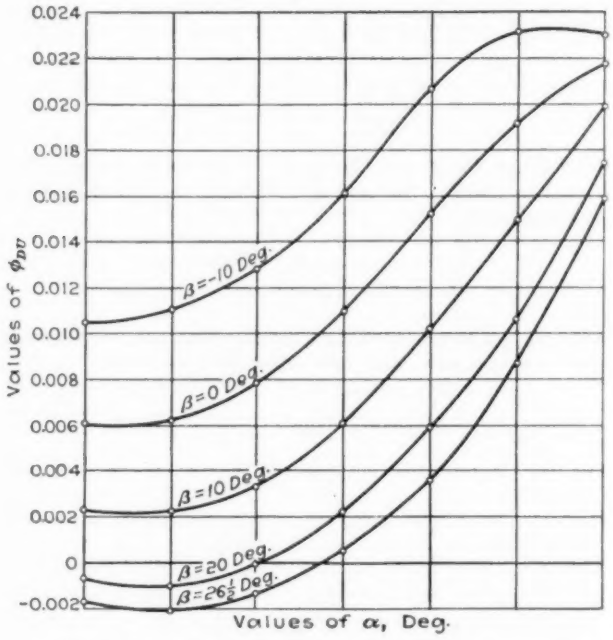


FIG. 5 VALUES OF φ_D FOR BIPLANES

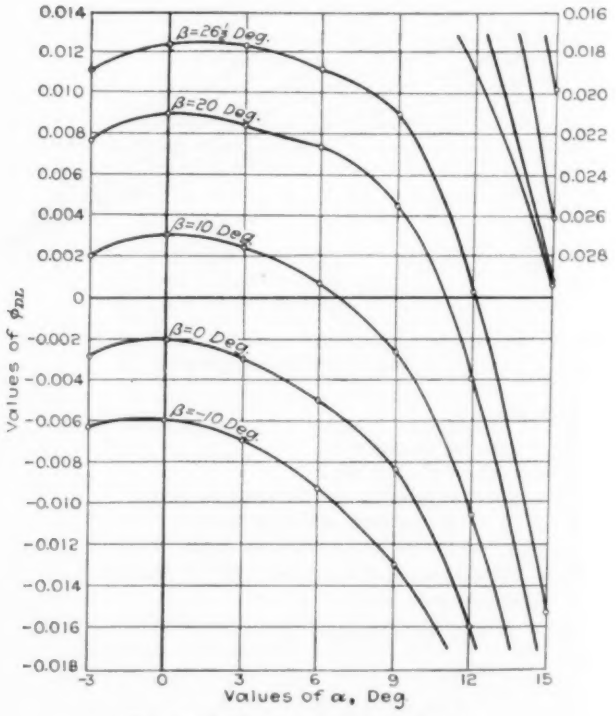


FIG. 6 VALUES OF φ_D FOR BIPLANES

$$\Delta C_{DU} = \frac{C_{LL} l_L}{b_U} \frac{(b_U + b_L)}{2G} \varphi_{DU}$$

$$\Delta C_{DL} = \frac{C_{LU} l_U}{b_L} \frac{(b_U + b_L)}{2G} \varphi_{DL}$$

The coefficients φ are functions of the cell arrangement and the angle of attack of the wing under consideration (see Figs. 3, 4, 5, and 6).

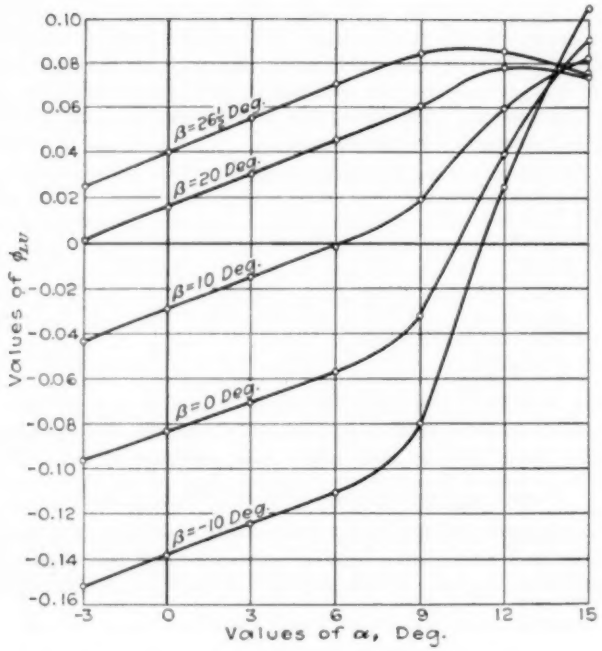


FIG. 7 VALUES OF φ_L FOR OUTER WINGS OF A TRIPLANE

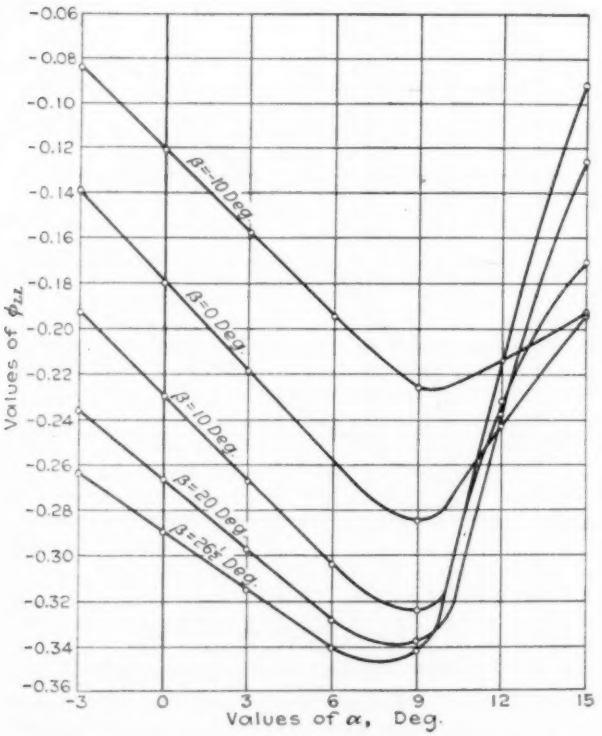


FIG. 8 VALUES OF φ_L FOR OUTER WINGS OF A TRIPLANE

$$C_L = \frac{S_U(C_{LU} + \Delta C_{LU}) + S_L(C_{LL} + \Delta C_{LL})}{S_U + S_L}$$

$$C_D = \frac{S_U(C_{DU} + \Delta C_{DU}) + S_L(C_{DL} + \Delta C_{DL})}{S_U + S_L}$$

Triplane. In case of a triplane the subscripts U, M, L refer to upper, middle, and lower wings, respectively. The interference of the upper and middle wing and middle and lower wing can be accounted for as in the case of biplane, but in addition one must take into account the interference due to the presence of the third wing. For example, in order to calculate the change in the characteristics of the middle wing, one must use the diagrams for the biplane and take into account the change due to the presence of the upper and lower wings, while in order to calculate the change in characteristics of the lower wing, one finds the change due to the middle wing from the biplane charts and the change due to the presence of the upper wing from the chart for the "outer wings of a triplane." Re-

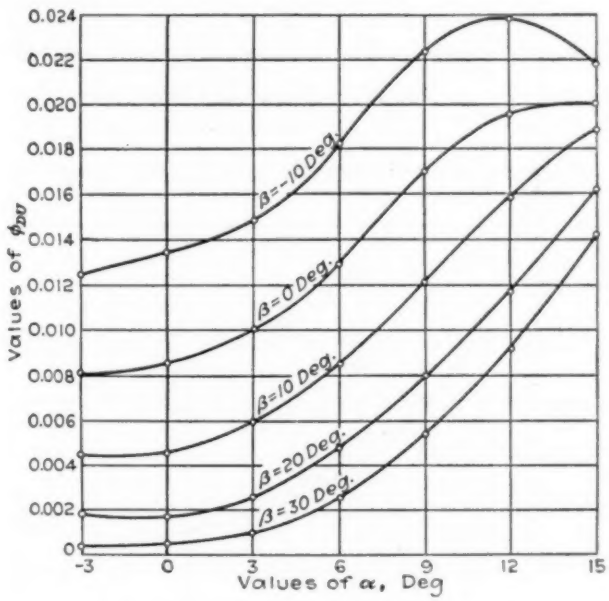


FIG. 9 VALUES OF φ_D FOR OUTER WINGS OF A TRIPLANE

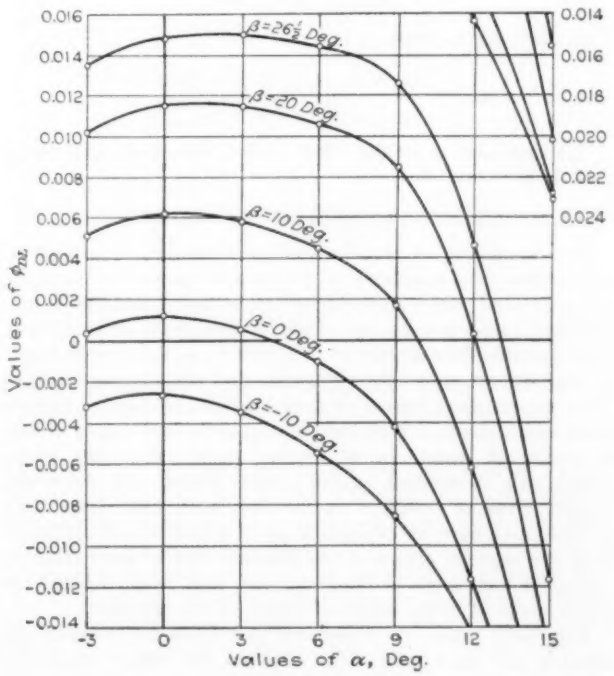


FIG. 10 VALUES OF φ_D FOR OUTER WINGS OF A TRIPLANE

taining the notations, as given, and using the subscripts to indicate the wing, one would have in this case

$$\Delta C_{LM} = \frac{C_{LU} t_U (b_U + b_M)}{b_M \quad 2G_{UM}} \varphi_{LL} \text{ (for } \alpha_M \text{ and } \beta_{UM})$$

$$+ \frac{C_{LL} t_L (b_M + b_L)}{b_M \quad 2G_{ML}} \varphi_{LU} \text{ (for } \alpha_M \text{ and } \beta_{ML})$$

$$\Delta C_{DL} = \frac{C_{LU} t_U (b_U + b_L)}{b_L \quad 2G_{UL}} \varphi_{DL} \text{ (for } \beta_L \text{ and } \beta_{UL})$$

$$+ \frac{C_{LM} t_M (b_M + b_L)}{b_L \quad 2G_{ML}} \varphi_{DL} \text{ (for } \alpha_L \text{ and } \beta_{ML})$$

The total change in the lift coefficient of a triplane would be

$$\Delta C_L = \frac{S_U \Delta C_{LU} + S_M \Delta C_{LM} + S_L \Delta C_{LL}}{S_U + S_M + S_L}$$

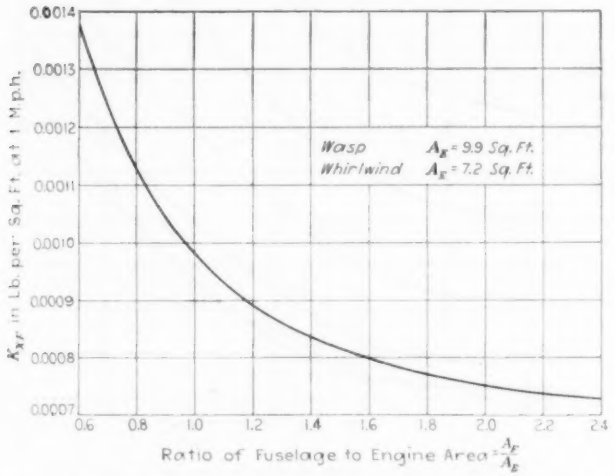


FIG. 11 DRAG COEFFICIENT OF FUSELAGE AND NACELLES OF AIR-COOLED ENGINES

Similarly for the drag

$$\Delta C_D = \frac{S_U \Delta C_{DU} + S_M \Delta_{DM} + S_L \Delta C_{DL}}{S_U + S_M + S_L}$$

The coefficient φ for the effect of the outer wings is given in Figs. 7, 8, 9, and 10.

CALCULATION OF PARASITE RESISTANCE

The calculation of parasite resistance consists in evaluating and adding the resistance of component parts. It must be borne in mind that a mere mechanical addition would be hardly sufficient, and a due account of interference and mutual shielding must be taken into account. It is hard to give any precise rule in that respect, and only experience and wind-tunnel observations may suggest means of taking it into account. Another point which should not be lost sight of is the importance of flow deviation on resistance of certain parts. Radiators under bodies, axle streamlines, engine nacelles placed near the wings, certain types of fuselages in the downwash from large wing surfaces may have an effective angle of attack entirely different from the angle of attack of the airplane, and in some cases no change in their angle to the actual flow may result through the whole range of flying angles of attack of the airplane.

This point must be considered in calculating parasite resistance, and since no rules can be given, the designer must rely on his judgment.

A word of caution is necessary also in regard to so-called streamline struts: some struts, but very few, conform to the best type of streamline strut known, and one must bear in mind the type actually used in assigning the resistance coefficients.

The following list contains parts usually included by the author in calculating the parasite resistance:

Fuselage and engine (when a unit)
Engine nacelles
Radiators
Floats
Hulls
Struts
Wires
Wheels
Tail skid or tail wheel
Armament
Tail surfaces
Interference and miscellaneous

At times other parts must be included such as landing lights wing flares, flotation gear, bomb racks, and bombs, etc.

FUSELAGE; OPEN COCKPIT; WATER-COOLED ENGINES

Fuselages with well-cowled water-cooled engines, well streamlined, and no radiator have a resistance coefficient of about, 0.00045 lb. per sq. ft., but ordinary types would be closer to

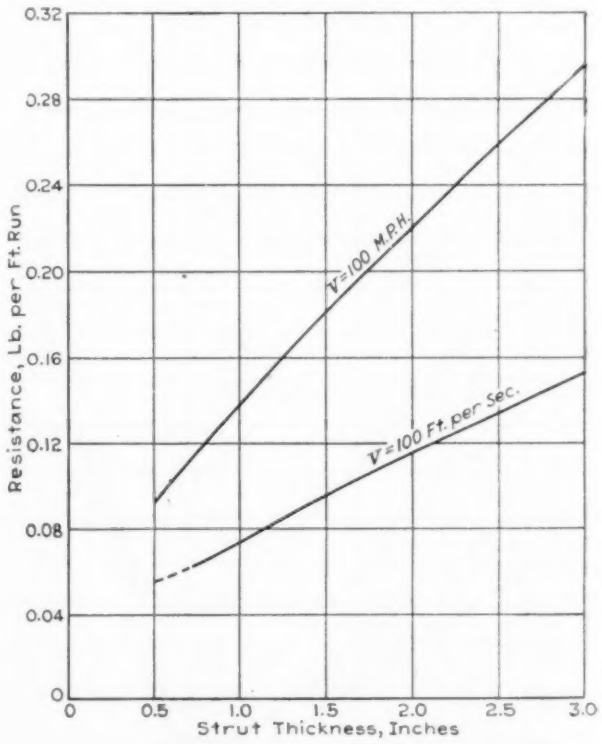


FIG. 12 DRAG PER FOOT AT 100 MILES PER HOUR

0.00060 to 0.0007. Military two-seater types with comparatively clean lines would have a coefficient of about 0.00055 to 0.00060. If no spinner is present, the lines of necessity are not as clean, and about 5 per cent must be added for the loss of proper streamlining. In case of a nose radiator the resistance of the body must be increased about 30 to 35 per cent.

CABIN; WATER-COOLED ENGINES

The resistance of comparatively well-streamlined cabin airplanes without radiator can be taken as 0.0004 for the lower limit and as an average figure of 0.0005 lb. per sq. ft.

Engine Nacelles; Water-Cooled Engines

In general the drag coefficient of an engine nacelle is very close to that of a fuselage, and the increase due to the presence of a nose radiator is from 30 to 40 per cent. When, however, the engine nacelle is located above the wing, the drag coefficient must be increased about 50 per cent to account for the interference. In case of the engine being located under the wing, this correction is less and may be taken as about 20 per cent.

It must be understood that the foregoing figures are only approximate and are not based on any definite test and that in individual cases the discrepancy may be considerable. On an average, however, they will be found sufficiently accurate.

Fuselages and Nacelles; Air-Cooled Engines

In order to obtain the data on resistance of fuselages and nacelles with air-cooled engines, the author analyzed several airplanes the performance and design characteristics of which were known to him, and using the present method of performance calculations obtained the resistance due to the engine-body and engine-nacelle units. The resistance coefficient is plotted against the ratio of maximum cross-sectional area of the body to the total area of the engine and may be used for nacelles, open cockpit fuselages, and cabins. If no spinner is present, a 5 per cent correction may be applied.

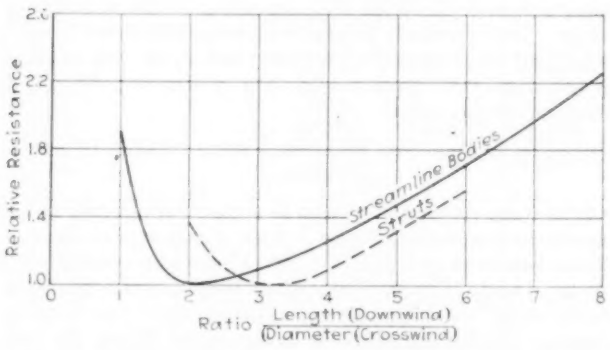


Fig. 13 Variation of Drag Coefficient With Fineness Ratio for Struts and Streamline Bodies

These data apply to the comparatively well-streamlined bodies, and in case of irregular shapes the coefficient would have to be revised, taking into account the fact that the resistance of an air-cooled engine can be considered as resistance proper plus interference. For poor-streamlined shapes the interference resistance would be decreased, but the drag of the body proper would be increased.

Floats. The average resistance of a well-streamlined but not a special racing float would be in the neighborhood of 0.00045 to 0.0005, while an average float and a deep hollow V-bottom float would have a resistance of 0.00055 to 0.0006 lb. per sq. ft. Wing-tip floats well-streamlined would be 0.00035. Wing-tip floats average 0.00055 and as high as 0.0007.

Hulls. The hull resistance coefficient varies from 0.0004 to 0.00065, but for an average hull one may use 0.00055 lb. per sq. ft.

Radiators. Free air radiator would be 0.0018 to 0.0022. A good average figure is 0.0020. In estimating the resistance of a free-air radiator one must give consideration to the possible flow deviation, interference between the radiator and the member of the chassis, etc.

Struts (Diehl). The lowest recorded drag coefficient was obtained on Navy strut No. 1, the offsets for which are given in Table 1 and the resistance in Figs. 18 and 19. For commercial shapes the resistance must be increased about 20 per cent.

Tail Skid or Tail Wheel. The average resistance is 0.0004 to 0.0005. In general the resistance can be approximately calculated when the design is known.

Armament. No definite data are available with exception of a flexible machine gun with a scarf mount, lined fore and aft. Diehl gives the resistance as 0.002 V^2; broadside the coefficient is increased about 50 per cent.

Tail Surfaces. In order to obtain the resistance of tail surfaces one may calculate separately the resistance of vertical tail surfaces, and having calculated the effective angle of attack of the horizontal tail to calculate its resistance. This refinement is, however, unnecessary because of the average error involved in assumed variation of parasite drag with change in angle of attack. It is best to include the resistance of the total tail area in the parasite drag calculation and assume a variation in parasite which would take care of this inclusion. The following, however, are various downwash formulas which seem to give a good agreement with downwash actually observed in the wind tunnel:

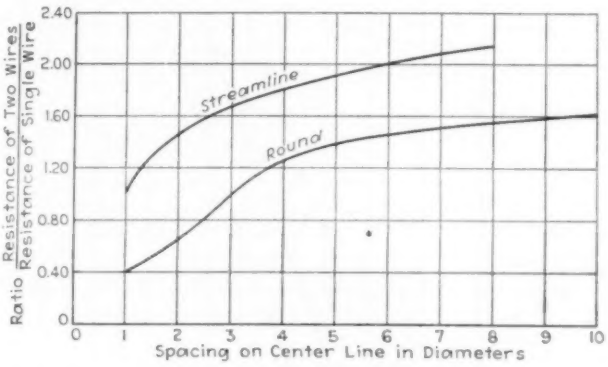


Fig. 14 Effect of Fore-and-Aft Spacing on the Drag of a Pair of Wires

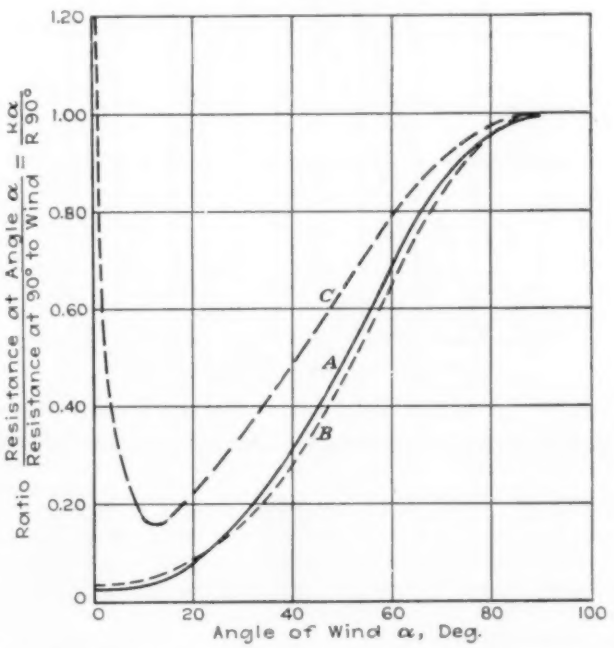


Fig. 15 Effect of Inclination on the Drag of Wire and Cable. Curves *A* and *B* Based on Unit Length, Curve *C* on Projected Length

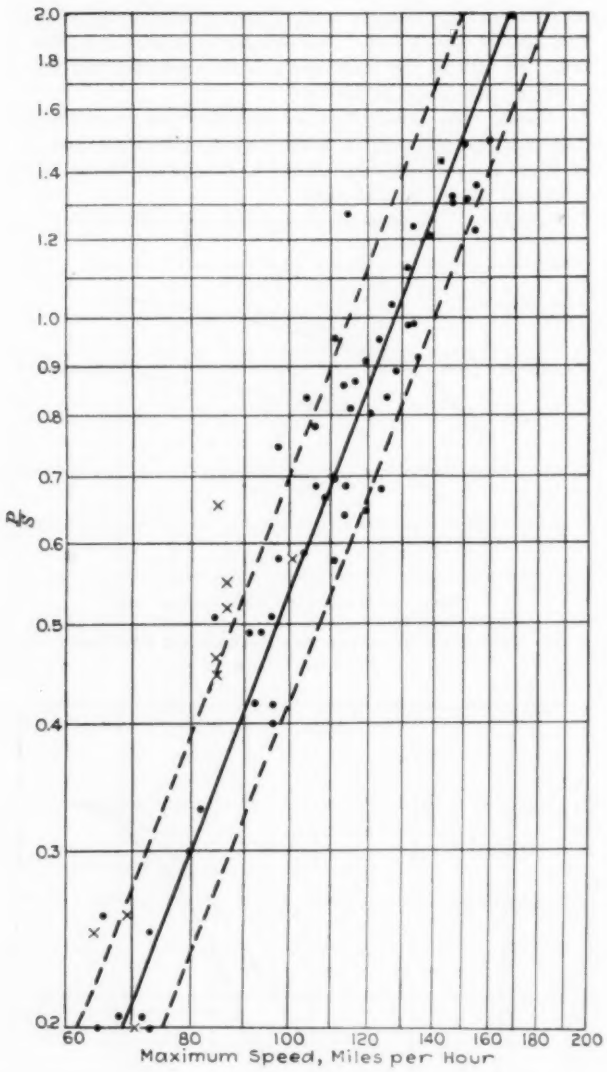


FIG. 16 DERIVATION AND TRIAL OF MAXIMUM-SPEED FORMULA

AVERAGE TAIL CHARACTERISTICS FOR AN ASPECT RATIO OF 3.

$\alpha°$	K_y	K_x
0	0	0.0000400
2	0.00032	0.0000412
4	0.00064	0.0000447
6	0.00096	0.0000505
8	0.00126	0.0000595
10	0.00160	0.0000722
12	0.00185	0.0000895
14	0.00210	0.0001120

Downwash (Diehl).

$$\epsilon = \frac{85C_L}{(K^2b^2/S)}\,(X+1)^{-0.38}(y+1)^{-0.23}$$

where x is the distance behind the trailing edge, y the distance above or below the chord measured in chord length.

Downwash (Warner). For the average tail location on the basis of the foregoing formula Warner suggests the following average expression:

$$\epsilon = \frac{36\,C_L}{(K^2b^2/S)}$$

Downwash (Föppl). The formula which from the author's experience seems to give a very good result is the one given by Föppl with slightly modified coefficients:

$$\epsilon° = \frac{C_{LU}}{2\pi}\,57.3\,\frac{t_U}{b_U}\left[1+\sqrt{1+\left(\frac{b_U}{2x_U}\right)^2}\right] + \frac{C_{LL}}{2\pi}\,57.3\,\frac{t_L}{b_L}\left[1+\sqrt{1+\left(\frac{b_L}{2x_L}\right)^2}\right]$$

where x is the horizontal distance between the trailing edge of the respective wing and the leading edge of the tail.

Downwash (Vedroff). The formula recommended by Vedroff and based on an analysis of wind tunnel experiments is:

$$\epsilon° = \frac{2.9C_L}{\pi(k^2b^2/S)}\,57.3\left[1+\left(\frac{b}{4x}\right)^2\right]$$

where x is the distance from $^1/_3$ of the chord to the point in question.

Interference and Miscellaneous

Inasmuch as it is practically impossible to account for the drag of minor parts and interference between different parts, their effect can be taken into account by increasing the total parasite from 10 to 15 per cent, depending on the cleanness of design. For a carefully streamlined design the effect of small projections and irregularities is larger, and 15 per cent increase always gave very good results. For a standard design 10 per cent would suffice.

Variation of Parasite Resistance With Change in Angle of Attack

It has been customary at times to neglect the variation of the parasite resistance with change in angle of attack or to separate this resistance in two groups of which, one was variable with change of angle of attack while the second one, usually including struts, wires, air-cooled engines, radiators, etc., was taken as constant through the range of flying angles. While the first assumption is unquestionably erroneous, the second will be found somewhat complicated, since the increase in drag may not be due directly to increase of resistance of individual parts, but rather to an increase in mutual interaction and interferences.

The author believes that it is best to assume the whole parasite resistance as variable with angle of attack, this increase being

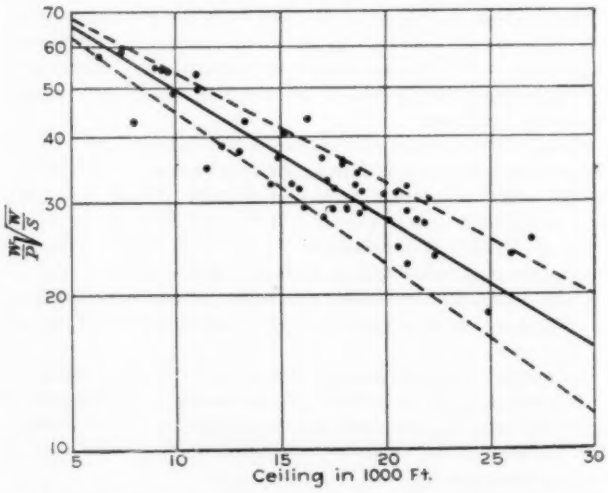


FIG. 17 VARIATION OF ABSOLUTE CEILING WITH WING LOADING AND POWER LOADING

TABLE 1 OFFSETS FOR NAVY STRUTS AND C-CLASS AIRSHIPS

% L	% D Navy strut	% D C-class airship	% L	% D Navy strut	% D C-class airship	% L	% D Navy strut	% D C-class airship
0	0	0	20.00	91.1	94.7	80.00	56.2	66.5
1.25	26.0	20.0	25.00	95.9	98.2	90.00	33.8	49.3
2.50	37.1	33.5	30.00	98.8	99.8	95.00	19.00	36.2
5.00	52.5	52.6	35.00	100.0	99.9	98.00	7.8	22.5
7.50	63.6	65.8	40.00	99.5	99.0	100.00	0	0
10.00	72.0	75.8	50.00	95.0	95.0			
12.50	78.5	83.5	60.00	86.1	88.5			
15.00	83.6	88.7	70.00	73.2	79.0			

TABLE 2 WIRES AND CABLES. DRAG COEFFICIENTS (ARMY HANDBOOK)

Round Wire B. & S. gage	Round Wire Diameter	Round Wire Drag coefficient	Cable Diameter	Cable Drag coefficient
14	0.0641	0.0000142	1/16	0.0000180
12	0.0808	0.0000185	1/8	0.0000380
10	0.1019	0.0000245	3/16	0.0000575
8	0.1285	0.0000318	1/4	0.0000770
7	0.1443	0.0000361	5/16	0.0000970
6	0.1620	0.0000410	3/8	0.0001165
5	0.1819	0.0000463	7/16	0.0001360
4	0.2043	0.0000525	1/2	0.0001555

TABLE 3 TURNBUCKLES. RESISTANCE AT 100 M.P.H. (DIEHL)

Cable diameter	Resistance at 100 M.p.h. Turnbuckles plus eyes complete	Turnbuckles plus one eye only
1/16	0.42	0.25
1/8	0.65	0.43
3/16	1.01	0.63
1/4	1.5	0.86
5/16	2.05	1.08
3/8	2.60	1.33

TABLE 4 STREAMLINE WIRES. DRAG COEFFICIENT AND RESISTANCE OF FITTINGS (ARMY HANDBOOK)

Size	Drag coefficient per foot	Resistance of two fittings
10–32	0.00000592	0.0000455
1/4–28	0.00000734	0.0000955
5/16–24	0.00000898	0.0001460
3/8–24	0.00001121	0.0001940
7/16–20	0.00001388	0.0002540
1/2–20	0.00001530	0.0003920

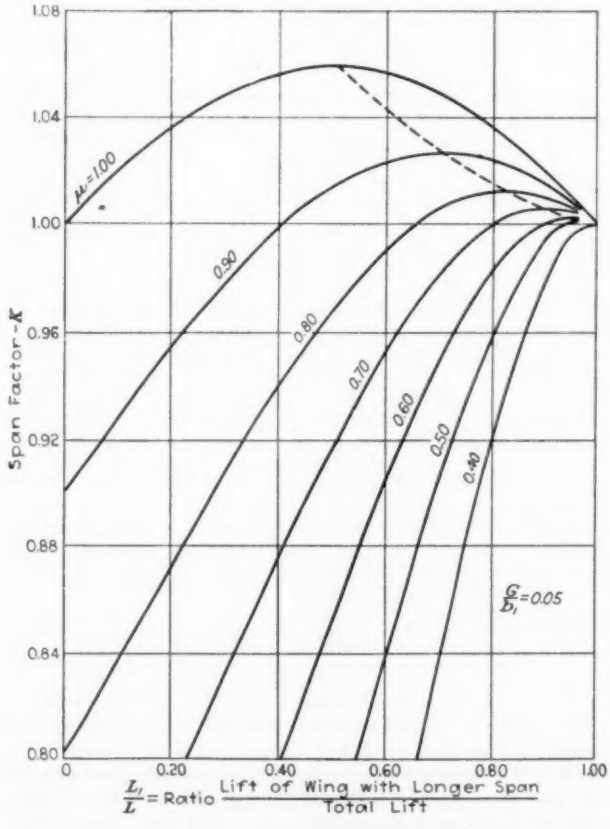


FIG. 18 MUNK'S SPAN FACTOR k FOR $\frac{G}{b_1} = 0.05$

TABLE 5 WHEELS. DRAG PER WHEEL IN POUNDS AT ONE MILE PER HOUR

Tire size	Bare wheel	Usual fairing	Full fairing
20 × 2	0.00061	0.00042	0.000203
26 × 3	0.001080	0.00072	0.000360
28 × 4	0.001410	0.00094	0.000470
30 × 5	0.001815	0.001210	0.000605
32 × 6	0.002285	0.001525	0.000761
36 × 8	0.003380	0.002250	0.001125
44 × 10	0.005120	0.003410	0.001705
54 × 12	0.007590	0.005060	0.002530

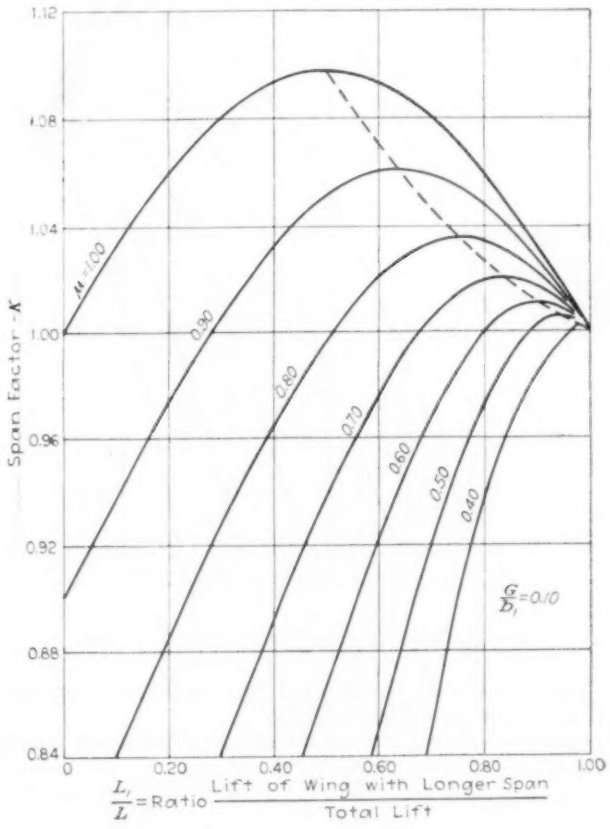


FIG. 19 MUNK'S SPAN FACTOR k FOR $\frac{G}{b_1} = 0.10$

a function of the angle of attack rather than a function of speed range. This increase is individual for different types of airplanes.

The angle of attack in this case is referred to the thrust line of the airplane and not to the wing chord. In view of a somewhat different behavior of floats and hulls, it seems that it is best to assume a different law of variation of parasite resistance of the latter, and consequently two curves of variation are given.

Parasite resistance versus angle of attack (excluding floats and hulls):

$\alpha°$	D/D min.
− 8	1.28
− 6	1.12
− 4	1.04
− 2	1.01
0	1.00
2	1.03
4	1.07
6	1.12
8	1.20
10	1.28
12	1.40
14	1.55
16	1.82
18	2.25
20	3.04

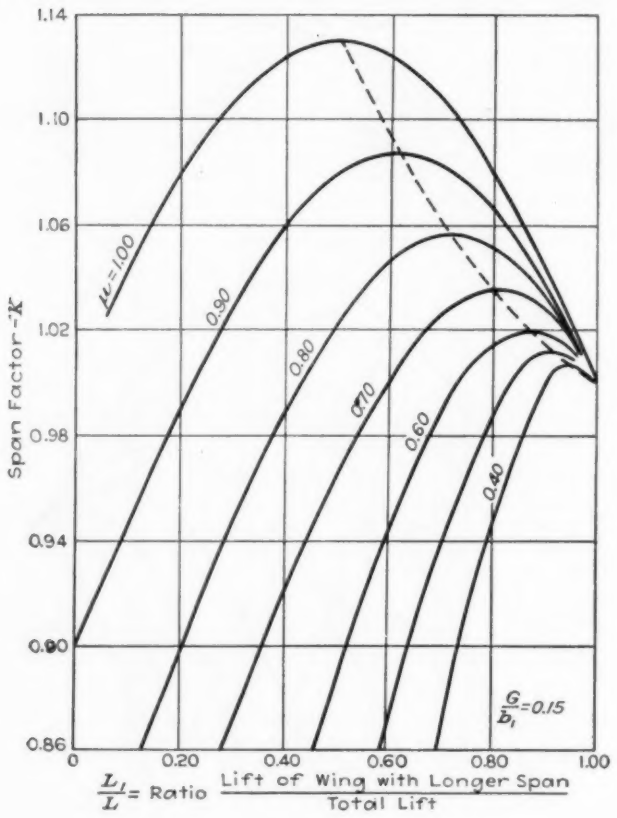


FIG. 20 MUNK'S SPAN FACTOR k FOR $\frac{G}{b_1} = 0.15$

This is an average variation and may vary about 15 per cent on either side for different designs.

Resistance of floats and hulls versus angle of attack (Diehl, N.A.C.A., T.R., No. 236):

$a°$	D/D min.
− 8	1.30
− 6	1.10
− 4	0.95
− 2	0.97
0	1.00
2	1.15
4	1.30
6	1.55
8	1.85
10	2.30
12	2.80
14	3.40
16	4.10
18	4.75
20	5.50

CHARACTERISTIC CURVES; LIFT, DRAG, AND L/D

Having the corrected characteristics of the wing or the cell and parasite resistance against angle of attack, it is possible now to obtain the lift drag and L/D of the full size machine at 1 mile per hour. The summation or resistance is done taking into account the fixed incidence of the wing, since the angle of attack of the parasite resistance is referred to the thrust line and not to the wing chord.

PERFORMANCE CALCULATION

Knowing the characteristic curves of the airplane, it is possible now to proceed with the calculation of the theoretical flight characteristics and general performance of the craft. It is necessary, first, to obtain the horsepower available and horsepower required, the knowledge of which will permit of calculating the maximum and minimum speeds, best speeds to climb, surplus horsepower, rates of climb and times to climb to any altitude, service and absolute ceilings. Furthermore, one can obtain conditions of throttled flight, range, and endurance and solve problems in connection with phases of operation such as take-off and landing runs, etc.

It is assumed that all are acquainted with the methods of calculation in use, but for convenience in Appendix 2 the author has collected such information as may be of benefit in routine performance calculation.

In calculating the horsepower available and the variation of horsepower with altitude, the author in the past has made use of the data collected by him, but since the discrepancies found with Mr. Diehl's curves are well within the experimental limits, it was thought best to adopt the latter information as the first step in an attempt of standardization of performance calculations. Appendix 2 also gives a brief explanation of a new use of the logarithmic polar diagram originated by the author. Although it has already appeared in the print, it is included here for the benefit of those who prefer grapho-analytical solutions and for the sake of completeness of references.

APPROXIMATE ESTIMATE OF HIGH SPEED AND CEILING

In his book, "Aerodynamics," Mr. Warner gives two diagrams which check very well the average characteristics met in practice.

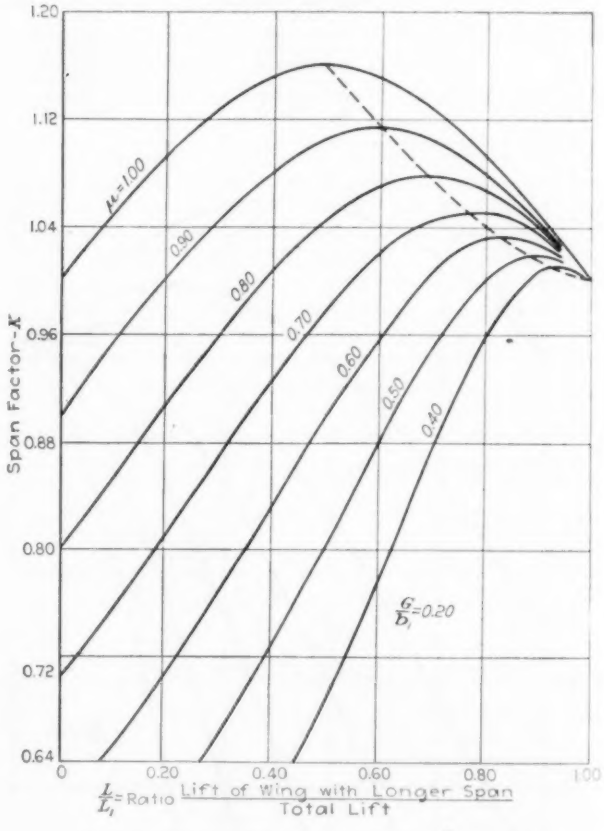


FIG. 21 MUNK'S SPAN FACTOR k FOR $\frac{G}{b_1} = 0.20$

These two diagrams present a further advantage, since one can plot lines for designs of different fineness and thus have a ready reference for an approximate estimate. (See Figs. 16 and 17.)

The average distribution of resistance at high speed and the percentage of total resistance are as follows:

	Well streamlined		
Type of plane	Monoplane	Sesquiplane	Biplane
Type of cooling	Prestone	Prestone	Air-cooled
Engine body	14.5	15.0	23.5
Radiator	6.2	6.0	0
Landing gear	10.5	10.5	8.5
Wing struts, wires, and interference	7.0	11.0	15.0
Tail surfaces	9.2	8.3	7.0
Wing drag	37.0	33.2	30.5
Propeller loss	15.6	16.0	18.5

The foregoing is an example of a typical distribution of parasite resistance in a two-seater military-type airplane. It is based on a very small number of estimates and cannot be generalized.

Conclusions

This paper was written with the intention of giving an engineer a ready reference to the methods and data necessary for practical performance calculations. Most engineers prefer their own methods and trust their own data, but a certain amount of standardization is desirable, and therefore an attempt was made to utilize what was thought the most reliable of available information.

It was impossible to give any proofs or more detailed explanations of the methods given, but it is believed that all are well

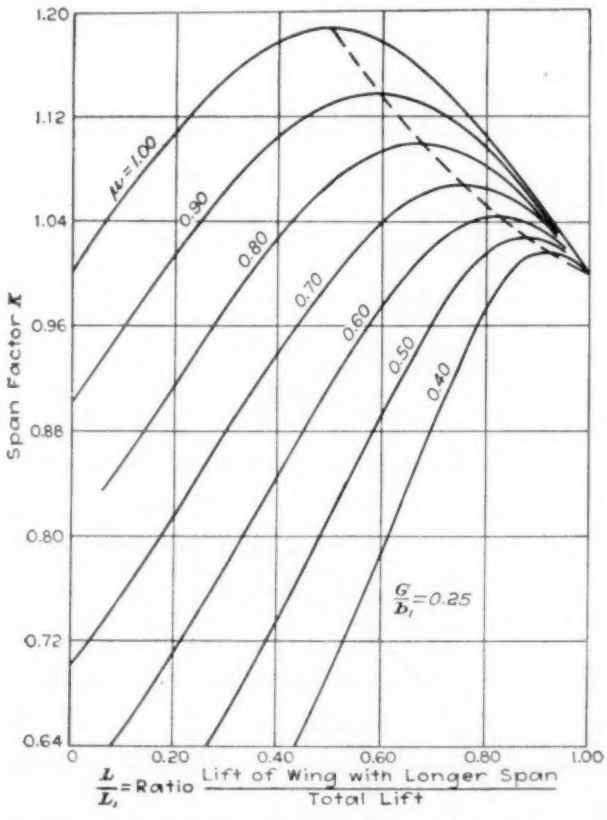


Fig. 22 Munk's Span Factor k for $\frac{G}{b_1} = 0.25$

acquainted with the subject of aerodynamics and will have no difficulty in applying the data.

It is hoped that this paper, in full or in part, will be found helpful by engineers in their everyday experience.

The accuracy of a performance calculation hinges greatly on the experience and judgment of the engineer because of the empirical nature of most of the coefficients used, and it is not surprising that in order to obtain good results, in addition to

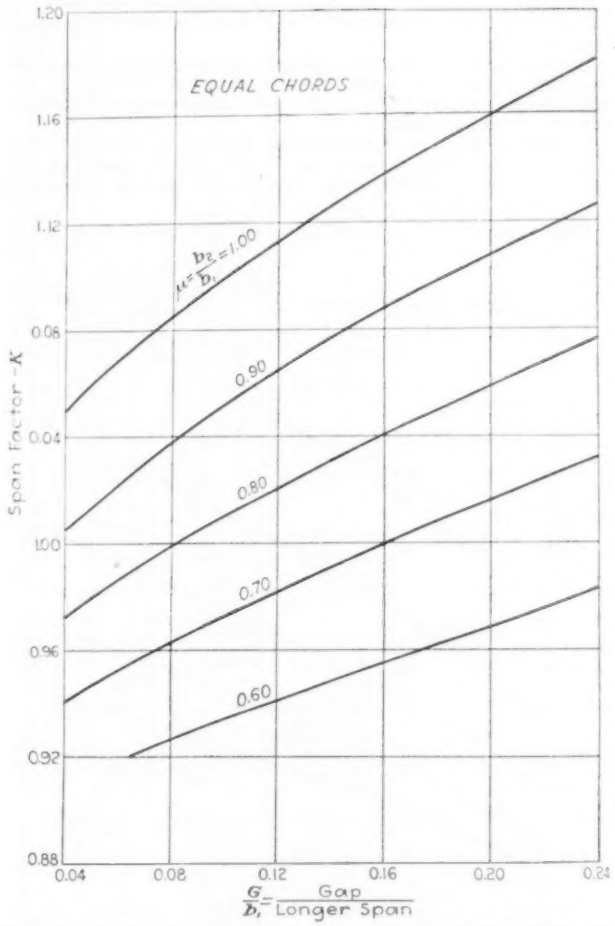


Fig. 23 Munk's Span Factor k for Biplanes With Wings of Equal Chords

experience one must possess what one may call the aerodynamic sense.

Knowledge of the theory and fundamentals is a necessary but not a sufficient condition to be an able performance calculator.

Appendix No. 1

Coefficient K. The magnitude of K may be found or interpolated from Figs. 18 to 24.

Appendix No. 2

Thrust Horsepower Available. Propeller-Engine Unit (Diehl). Propeller diameter: $D = K\sqrt[4]{\rho_0/\rho}\sqrt{\text{b.hp.}/(\text{r.p.m.}^2 . V)}$ in feet and miles per hour. The coefficient K is:

Wooden Propeller

Number of blades		2	3	4
K	Speed	285	255	240
	Climb	320	290	270
	Service	305	275	260

Metal Propeller

Number of blades		2	3
K	Speed	280	260
	Climb	315	300
	Service	300	290

General Efficiency Curve. Propeller efficiency for any value of V/ND ratio can be found knowing the maximum propeller efficiency and design $(V/ND)_0$. Fig. 25 gives values of η/η_0 plotted versus $\frac{(V/ND)}{(V/ND)_0}$.

Maximum Propeller Efficiency. The maximum efficiency for conventional two-bladed wooden and metal propeller is given in Fig. 26 (Diehl).

In order to take into account the mutual interaction of propeller body unit, a correction must be applied as given in Fig. 27. The body diameter should be taken at a point about one propeller diameter aft of the propeller.

In order to obtain the net maximum efficiency of a geared two- or four-bladed propeller, calculate the V/ND ratio for maximum efficiency of a two-bladed propeller and in Fig. 28 find the relative

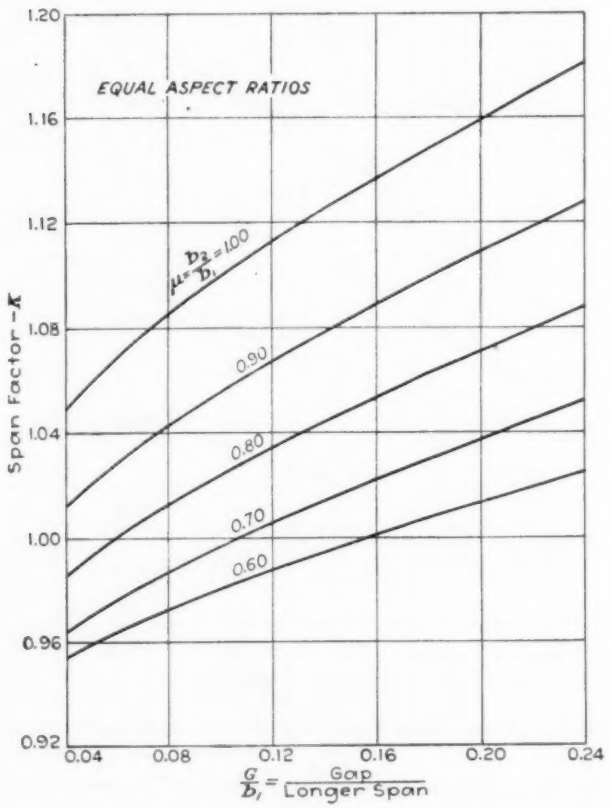


FIG. 24 MUNK'S SPAN FACTOR k FOR BIPLANES WITH WINGS OF EQUAL ASPECT RATIO

efficiency. The net efficiency is obtained by multiflying the maximum efficiency of a two-bladed direct-drive propeller times the relative efficiency (Diehl, N.A.C.A., T.R., No. 178).

AVAILABLE THRUST HORSEPOWER AT SEA LEVEL

In order to know the available thrust horsepower at sea level, it is necessary to know the variation of r.p.m. versus ratio of $V/V_{max.}$, variation of propeller efficiency versus V/ND, and also the variation of the brake horsepower of the engine versus r.p.m. Diehl in his book, "Engineering Aerodynamics," gives the derivation of average curves, and in view of uncertainties in propeller and engine behavior and other factors involved in performance calculation, it will be found sufficiently accurate for all practical purposes to use these average curves. (See Figs. 25, 29, 30, and 31.)

TABLE 6 VARIATION OF THP WITH $V/V_{max.}$ RATIO

$V/V_{max.}$	Ratio of $THP/THP_{max.}$ — R.p.m. max.		
	1800	2100	2400
1.2	1.035	1.015	0.995
1.1	1.025	1.015	1.005
1.0	1.00	1.00	1.00
0.9	0.96	0.97	0.975
0.8	0.908	0.925	0.94
0.7	0.85	0.865	0.885
0.6	0.78	0.80	0.82
0.5	0.70	0.72	0.74
0.4	0.60	0.617	0.635
0.3	0.49	0.495	0.515

THRUST HORSEPOWER, R.P.M., AND V IN THROTTLED FLIGHT (DIEHL)

Since some of the problems of performance calculations involve the condition of throttled flight, Figs. 32 and 33 give the solution of problems involving these factors.

TABLE 7 VARIATION OF THRUST HORSEPOWER WITH ALTITUDE (DIEHL)

Altitude	THP_a/THP_{SL}	P_{SL}/P_{ap}
0	1.0	1.0
2,000	0.925	1.03
4,000	0.854	1.061
5,000	0.820	1.077
6,000	0.787	1.094
8,000	0.723	1.128
10,000	0.663	1.164
12,000	0.608	1.201
14,000	0.556	1.240
15,000	0.532	1.261
16,000	0.508	1.282
18,000	0.465	1.325
20,000	0.425	1.370
22,000	0.387	1.418
24,000	0.352	1.468
25,000	0.336	1.521
26,000	0.320	1.577
28,000	0.291	1.635
30,000	0.261	1.697
32,000	0.234	1.763

VARIATION OF THRUST HORSEPOWER WITH ALTITUDE (DIEHL)

The variation as shown in Table 7 is given for a constant air speed which will be found very convenient in its application.

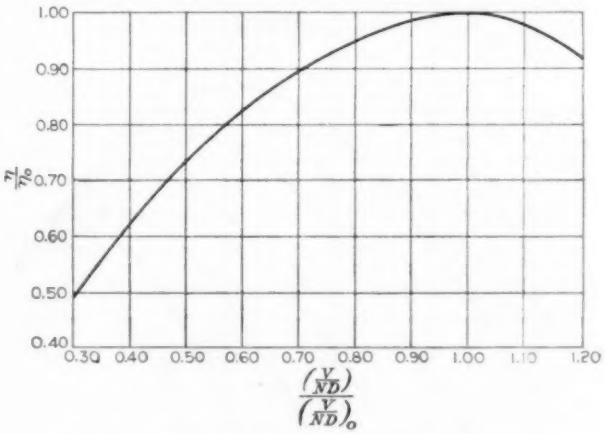


FIG. 25 GENERAL EFFICIENCY CURVE FOR PROPELLERS

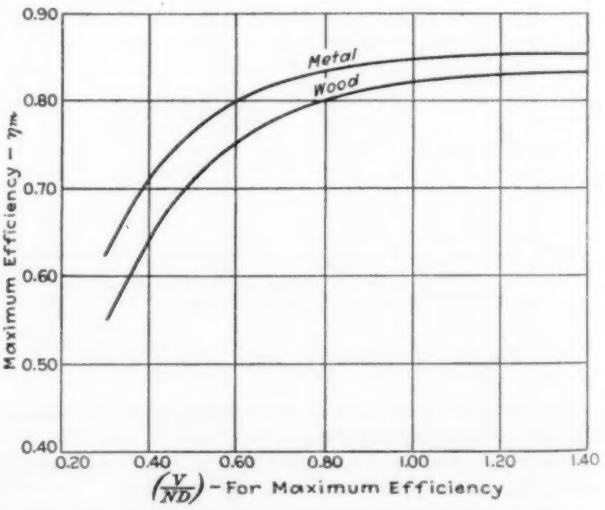


FIG. 26 VARIATION OF MAXIMUM PROPELLER EFFICENCY WITH $\frac{V}{nD}$

A New Application of the Logarithmic Polar Diagram

Knowledge of the characteristic curves of the plane and behavior of the engine-propeller unit permits of calculation of the theoretical performance of the airplane in question. The standard method of obtaining horsepower required and available and series of curves for sea level and altitude is assumed to be known to the reader. This method of calculation is somewhat laborious and requires a number of diagrams which are based on analytical calculations. In his practice the author prefers to use a grapho-analytical method based on an amplification of the logarithmic polar diagram. The method

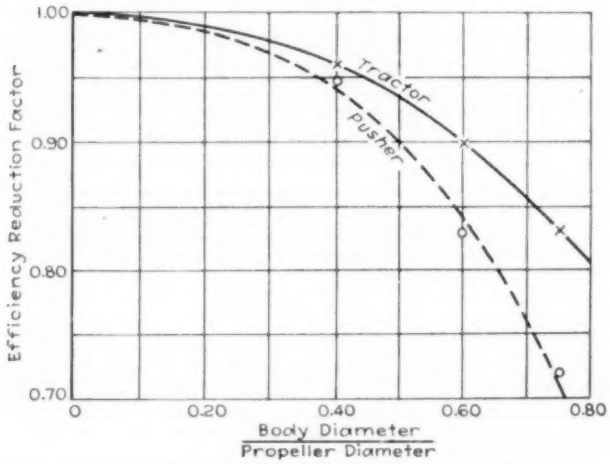


Fig. 27 Effect of Body Interference on Propeller Efficiency

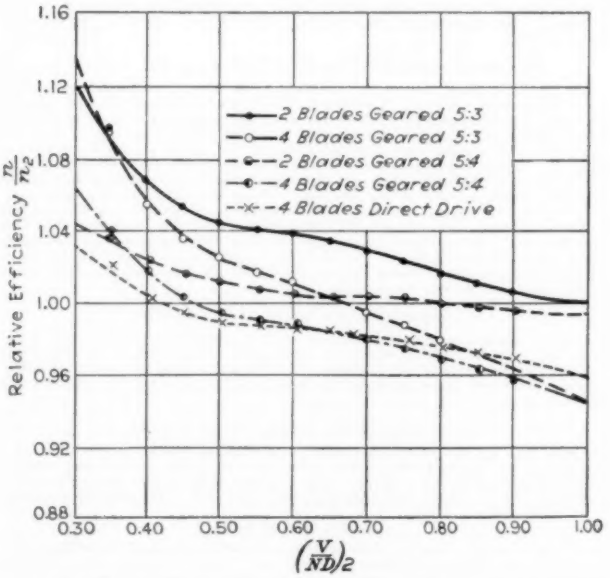


Fig. 28 $\frac{V}{nD}$ for Two-Bladed Propeller at Maximum Efficiency

was first described in the July, 1929, issue of *The Aero Digest*, in part as follows:

Lack of space prevents a detailed explanation of the theory of the logarithmic polar diagram and proofs of the new method, but the logarithmic polar principle is so simple that it will be best for the engineer to prove it for himself. The author suggests that the complicated analytical proofs be avoided since it is best to resort to simple graphical constructions and superpositions. After they are understood, numerous problems not mentioned in this article will suggest themselves and may be easily solved.

The original adaptation of the logarithmic polar diagram is credited to Mr. Rith of the Eiffel Laboratory and may be found in Eiffel's works "La Resistance de l'air et l'aviation" and "Nouvelles Recherches sur la Resistance de l'air et l'aviation." Disadvantages of the original Rith and later methods based on the use of the logarithmic polar diagram are in the approximate nature of the estimated performance obtained and the consequent inaccuracies and discrepancies as compared with other available methods.

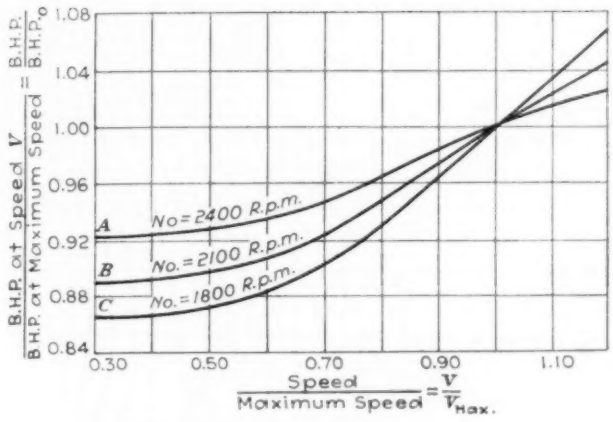


Fig. 29 General Curves for B.Hp. and V

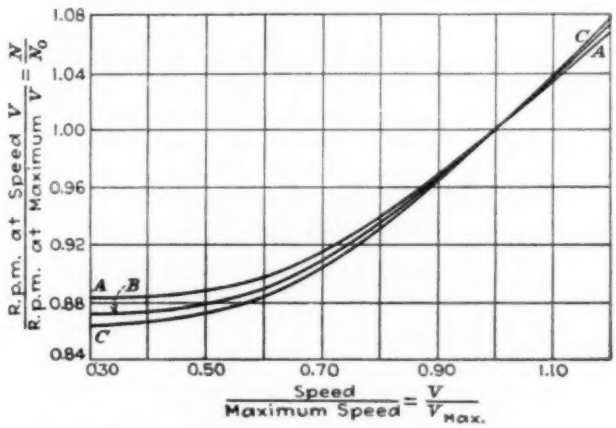


Fig. 30 General Curves for R.p.m. and V. N_0 = 2400 R.p.m. for Curve A, 2100 R.p.m. for Curve B, and 1800 R.p.m. for Curve C

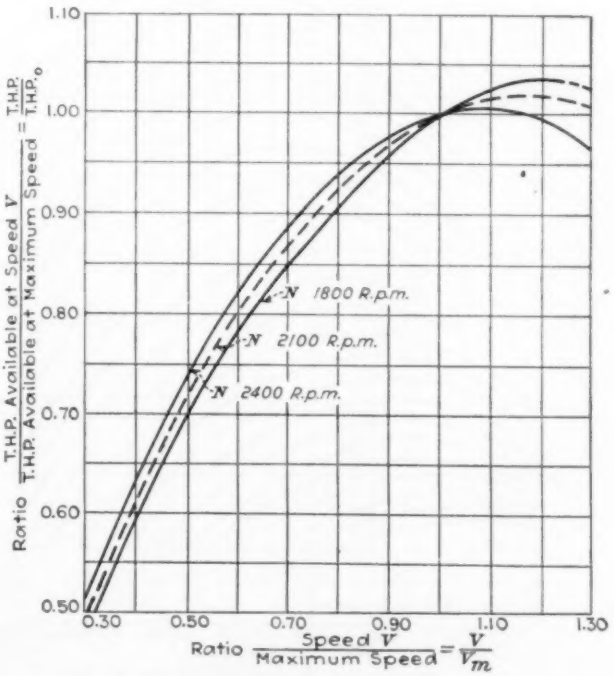


Fig. 31 General Curves for T.Hp. and V

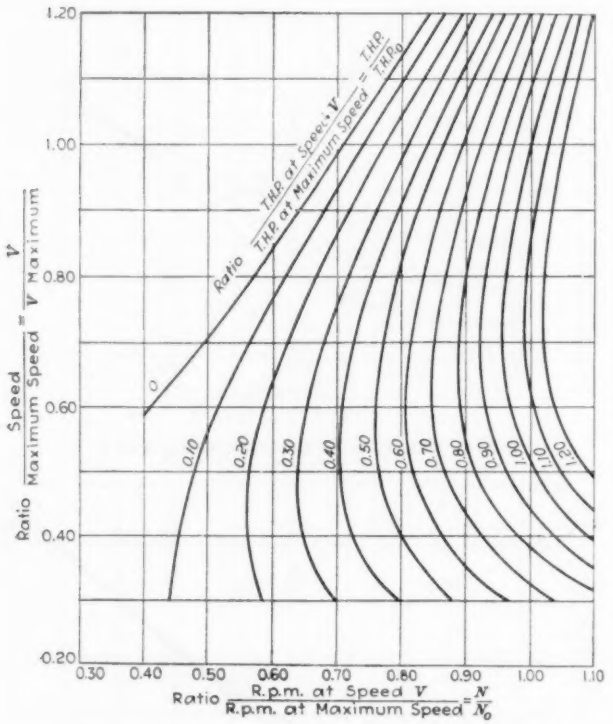


FIG. 32 GENERAL CURVES FOR T.HP., R.P.M., AND V

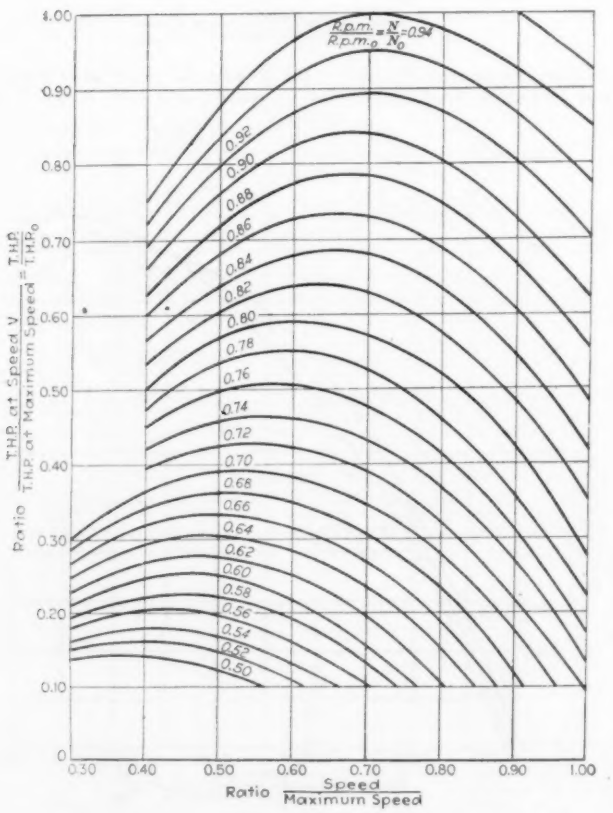


FIG. 33 GENERAL CURVES FOR T.HP., R.P.M., AND V

THEORY OF THE LOGARITHMIC POLAR DIAGRAM

Rith points out that if the characteristic curves of the airplane are plotted as a polar in logarithmic coordinates, the plot represents a functional dependence of all the main factors entering into the two principal equations of the flight characteristics of an airplane:

$$W = LV^2 \text{ and}$$
$$HP = DV^3$$

or expressed logarithmically:

$$\log W = \log L + 2 \log V \text{ and}$$
$$\log HP = \log D + 3 \log V$$

therefore:

$$\log L = \log W - 2 \log V \text{ and}$$
$$\log D = \log HP - 3 \log V$$

From the last two equations it is apparent that we can consider $\log L$ as the sum of $\log W$ and $-2 \log V$, while $\log D$ can be considered as the sum of $\log HP$ and $-3 \log V$. The values of $\log W$ and $\log HP$ can be plotted on the direction of the ordinate and abscissas directly, while $\log V$ can be plotted on a line inclined in such a way that the values projected on the direction of the ordinate will give $-2 \log V$ and the same point projected on the axis of abscissas will give a magnitude of $-3 \log V$. This new axis, called the speed axis, has the slope of 2/3 and the graduations have a modulus of $\sqrt{3^2 + 2^2} = 3.605$ times that of the modulus used on the other axes.

The diagram (Fig. 34) thus constructed gives the necessary functional dependence of the factors entering into the two equations.

In order to take into account the effect of the density of the air and so to obtain the characteristics of flight at altitude, we observe that in the two equations mentioned above we must incorporate the value of relative density. The new equations are written then as:

$$W = \sigma LV^2 \text{ and}$$
$$HP = \sigma DV^3 \text{ where}$$

σ is the ratio of the density at altitude to that at sea level. The necessary transformation can be obtained by assuming the origin of coordinates moved along a 45-degree line so that the magnitudes of L and D assume a new value equal to σL and σD. Consequently, in order to obtain the relationship of the factors involved at altitude, it is sufficient to incorporate a new reference axis (the altitude axis) which has a slope of 1/1, and to plot the logarithmic values of relative density with a modulus equal to $\sqrt{1 + 1} = 1.414$ of the modulus used for L and D axes.

Knowing the theory of the logarithmic polar chart, we can turn our attention to the practical application of it to performance estimate.

It is assumed that the characteristics of the airplane (that is, the lift and drag at one MPH on a full size machine) are determined either through calculations or from a wind tunnel test. Knowing the values of L and D and having a sheet of coordinates available, we plot the logarithmic polar curve.

DETERMINATION OF MAXIMUM SPEED AT SEA LEVEL

On the right-hand side of the diagram (Fig. 35), a vertical logarithmic scale is marked scale μ. This scale facilitates the determination of the available thrust horsepower when the propeller efficiency is known. Draw a horizontal line through the known propeller efficiency (line 1) until it intersects a 45-degree line passing through the point of BHP maximum (line 2). Draw a vertical line (3) through the point thus obtained until it intersects a horizontal line (4) passing through the ordinate indicating the gross flying weight of the airplane. Point A is the thrust horsepower available at the maximum speed. Drawing through point B line (5) parallel to the speed axis, the segment BC obtained between the intersection of this line with the polar (point C) and point B gives the magnitude of the maximum speed of the airplane. To find its value lay out from the origin of coordinates an equal segment on the speed axis and the reading at the point D will give the desired magnitude.

DETERMINATION OF STALLING SPEED

Draw a horizontal line (1) tangent to the uppermost point of the polar (Fig. 36). Draw line (2) through given gross flying weight. Line (2) is parallel to the speed axis. Segment AW (where A is the point of intersection of the two lines) gives the stalling speed of the airplane, which must be measured on the speed axis.

SPEED FOR FLIGHT AT MINIMUM POWER

Draw line (1) tangent to the polar and parallel to the V axis (Fig. 37). Draw line (2) through W corresponding to the given gross flying weight. The intersection of the two lines defines point A in such a way that AB is the speed corresponding to minimum power and AW is the minimum power.

MAXIMUM L/D

It can be easily proved that a vertical scale graduated logarithmically with the same modulus as used for the L and D axes, and located arbitrarily on the drawing will give us the ratio of L/D at the point of intersection of this scale with a line passing through the chosen point and inclined at an angle of 45 degrees. In graduating this scale, we must determine at least one reference point and graduate the scale in the same sense as the L axis. To obtain the value of maximum L/D, simply draw a line inclined at 45 degrees and tangent to the polar (Fig. 38). Point A gives the maximum L/D of the design.

The brief explanation of the theory and a few illustrations of the possible uses of the logarithmic polar diagram given above will suffice to explain the practical use of the new grapho-analytical method which consists of incorporating the curves of horsepower available and required drawn in logarithmic coordinates. This involves an incorporation of a few more reference curves as explained and illustrated in the following paragraphs.

HORSEPOWER REQUIRED AT SEA LEVEL

Knowing $V_{max.}$ calculate values of 0.9, 0.8, 0.7, 0.6, 0.5, 0.4, and 0.3 of $V_{max.}$ and mark them on the speed axis. This is not absolutely necessary for the purpose of obtaining the horsepower required, but later we will see that it will aid us in obtaining the horsepower available. The drawing of the curve of horsepower required will be illustrated by following the procedure for one point; for instance, point $V = 0.5V_{max.}$ (Fig. 39).

Draw line (1) through W and parallel to the speed axis. Project point

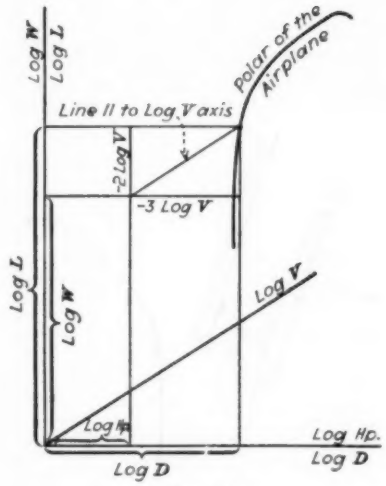


FIG. 34

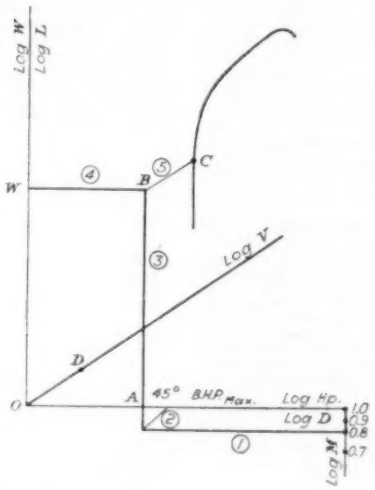


FIG. 35 DETERMINATION OF MAXIMUM SPEED AT SEA LEVEL

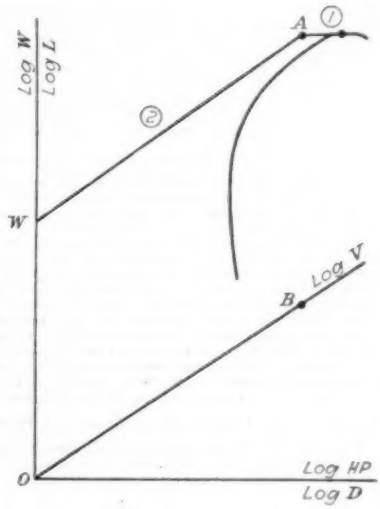


FIG. 36 DETERMINATION OF STALLING SPEED AT SEA LEVEL

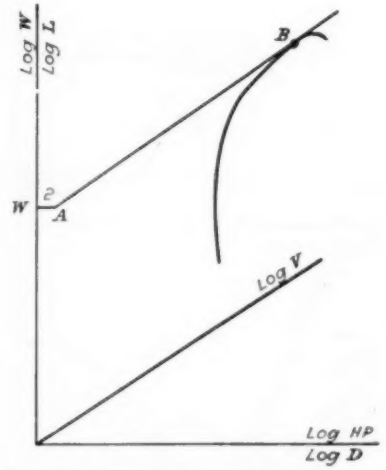


FIG. 37 DETERMINATION OF SPEED FOR FLIGHT AT MINIMUM POWER

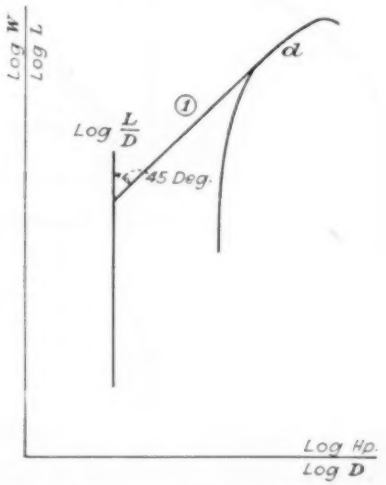


FIG. 38 DETERMINATION OF THE L/D RATIO

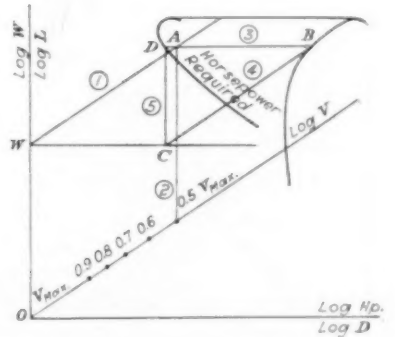


FIG. 39 DETERMINATION OF HORSEPOWER REQUIRED AT SEA LEVEL

$0.5V_{max.}$ on it. This gives point A. Draw a horizontal line through this point. Through the point B draw a line parallel to speed axis until it intersects the horizontal line passing through W. Through the point C thus obtained draw a vertical line until it intersects line AB. Since WC is the horsepower required to fly at a speed equal to $0.5V_{max.}$, point D represents a plot of it drawn at the proper point of the polar curve. Repeating this construction for several points and joining all these points by a smooth line we obtain the horsepower required plotted against the corresponding lift and, by virtue of construction, also the known speed.

HORSEPOWER REQUIRED AT ALTITUDE

In order to obtain the curve of horsepower required at any altitude, it is sufficient to shift the sea level curve to the right by the amount equal to the distance (1) which is measured between the speed axis and the ordinate axis at the height corresponding to the chosen altitude. Fig. 40 shows this construction for an altitude of 20,000 ft.

HORSEPOWER AVAILABLE AT SEA LEVEL

In order to construct the curves for horsepower available at sea level, we introduce three new constructional curves giving the variation of thrust horsepower against the ratio of $V/V_{max.}$

In Table 6 are found the three general curves giving the necessary relationship. In the lower left part of the diagram we incorporate these three curves.

The scale of propeller efficiencies is used now as the scale of THP/THP_0, while the scale of $V/V_{max.}$, is assumed to be plotted on the axis of altitudes (Fig. 41). In this construction we make use of the line (2) from Fig. 34. The power plant used determines which of the three curves shall be used.

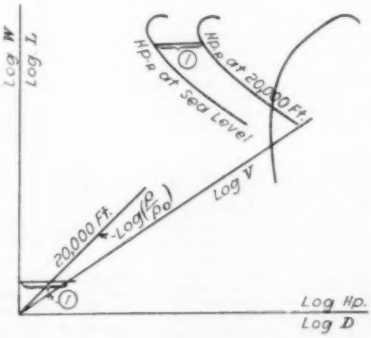


FIG. 40 DETERMINATION OF HORSEPOWER REQUIRED AT ALTITUDE

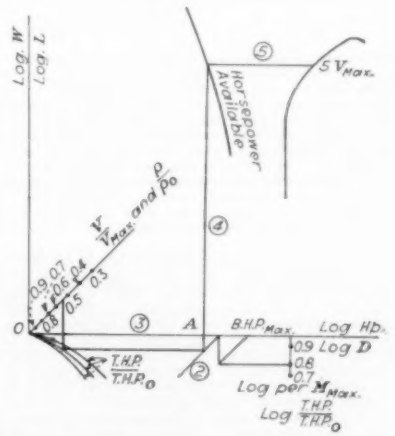


FIG. 41 DETERMINATION OF HORSEPOWER AVAILABLE AT SEA LEVEL

The construction will be illustrated for one point and must be similarly repeated for the rest of the points corresponding to fractions of $V/V_{max.}$ Fig. 38 shows the construction for point of V equal to $0.5V_{max.}$

Having chosen one of the THP/THP_0 curves we project the corresponding

point on the line (2). The point at intersection projected on the HP scale gives the horsepower available at this particular speed. Drawing line (4) until it intersects the horizontal through the point on the polar corresponding to the speed chosen, we obtain point B of the horsepower available curve at sea level. Repeating this construction for several more points and joining them with a smooth curve, we obtain the curve of horsepower available. The curves of horsepower available and horsepower required will intersect at point B of Fig. 2, corresponding to the maximum speed when the horsepower available is equal to the horsepower required.

Horsepower Available at Altitude, Unsupercharged

To obtain the horsepower at altitude, we will incorporate a curve giving the variation of the horsepower with altitude. Since our construction will be made assuming that the speed of the airplane remains constant, we can make use of the data given in Table 7.

Fig. 42 illustrates the construction of the point corresponding to an altitude of 10,000 ft. Inasmuch as the transformation of the polar for any altitude is made by an imaginary shift of the origin of the coordinates along a 45-degree line it is sufficient to locate the point B on the horizontal passing through the 10,000-ft. point on the axis of altitude so that a line drawn through it and inclined 45 deg. will cross the axis of horsepower giving the magnitude CO equal to 0.667 CA; that is, if the point A corresponds to 1000 HP, the value read at C must be 667 HP. Having the curve of variation of horsepower with altitude it is sufficient to slide the curve of horsepower available at sea level parallel to itself by the amount and along the direction (1) corresponding to the desired altitude as shown on Fig. 42.

Horsepower Available at Altitude With Supercharged Engine

Most of the present supercharged power plant installations are made so that the original sea level power is retained up to a certain fixed altitude,

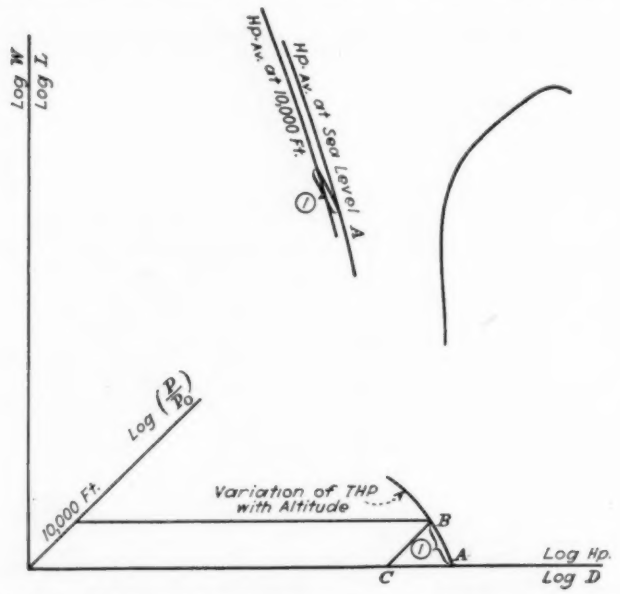


Fig. 42 Determination of Horsepower Available at Altitude, Unsupercharged

above which the power decreases at the normal rate. On our diagram this can be expressed by drawing instead of the previously drawn horsepower variation curve a new curve consisting of two branches: a vertical straight line from point A up to the horizontal, passing through the altitude up to which the sea level power is preserved and a curve from point C which is obtained by shifting horizontally the old curve until points B and C coincide. In the diagram the power was assumed to be retained up to an altitude of 10,000 ft. As the altitude performance on the polar diagram is obtained by an imaginary shift of the origin of coordinates along a 45-deg. line, to obtain the horsepower available at 10,000 ft., we must shift the curve of horsepower available at sea level to the right along a 45-deg. line by the amount equal to the segment (1).

Performance at Altitude

In order to obtain the speed and corresponding horsepower required and available at altitude, we must bear in mind that by virtue of construction that instead of a horizontal line passing through W (as it was done in Fig. 39) it is necessary to use another horizontal line such that if WB is parallel to the altitude axis, the segment WB must be equal to the segment OA where A is the desired altitude. Fig. 11 illustrates this for an angle of attack corresponding to the maximum speed at 10,000 ft. Since the point of intersection of horsepower required and available curves corresponds to the maximum speed, project point E on line BC and OF. Segment OF gives the magnitudes of horsepower required and available, since in this case they are equal. Drawing line CD parallel to speed axis, we obtain the magnitude of the maximum speed given by the segment CD, and point D permits us to interpolate the angle of attack corresponding to the maximum speed at 10,000 ft.

Complete Performance Prediction With the Aid of Logarithmic Polar Diagram

From the foregoing we are now able to obtain the complete data on theoretical performance of a new design. We can ascertain the maximum and stalling speeds for any altitude at which the plane is capable of flying. By joining the points of maximum speeds plotted on the corresponding horizontal lines and noting the intersection of this new curve with the line tangent to the polar and parallel to the speed axis (line used to obtain the minimum power required for flight), we can determine the absolute ceiling.

In order to obtain the best speeds for climbing and to determine the excess horsepower, we can make use of the following observation: From the analyses of a number of airplanes it can be found that the best speed at which to climb at sea level is usually equal to 0.58 $V_{max.}$ At the ceiling,

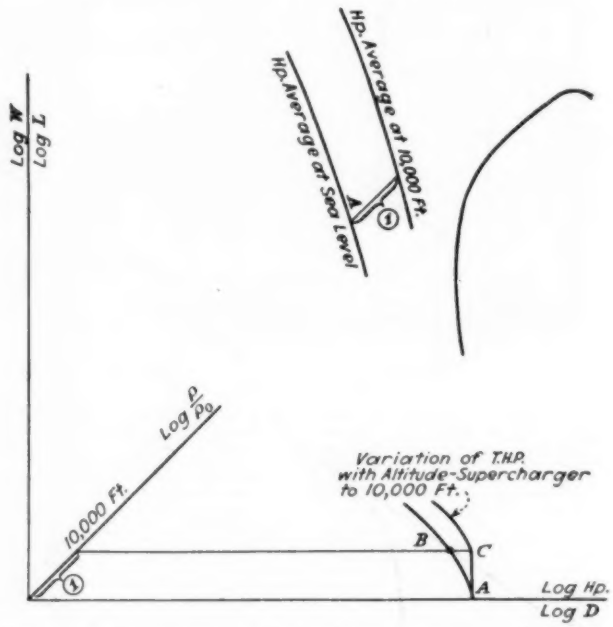


Fig. 43 Determination of Horsepower Available at Altitude, Supercharged

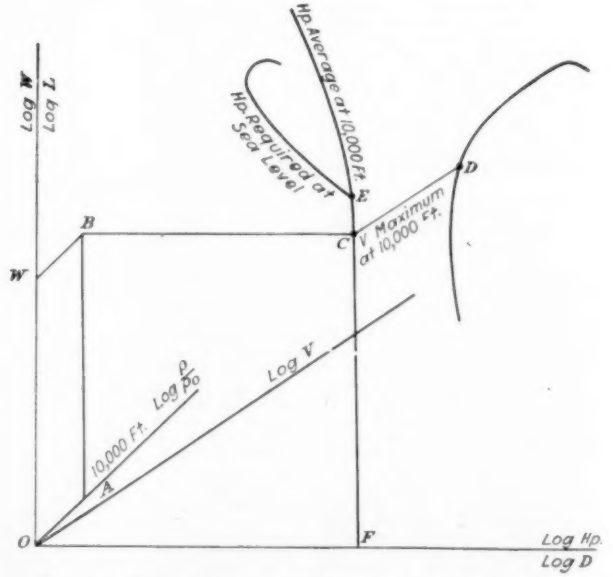


Fig. 44 Determination of Performance at Altitude

the planes would fly at a speed corresponding to the minimum power necessary to maintain flight. If we join by a straight line the point corresponding to the speed equal to 0.58 $V_{max.}$ on the horsepower-required curve at sea level with the point of minimum power necessary to fly at the ceiling, the points of intersection of this line with horsepower-required curves will define, within the accuracy of practical requirements, the best speeds for climbing. Knowing these speeds, we can obtain the excess horsepower available for climb.

We can resort now to standard analytical methods and obtain the curve of rate of climb and time to climb to different altitudes. Fig. 43 shows an example of the use of logarithmic polar as it is employed in routine practice.

The particular advantages afforded by the method described in the article, in addition to those already mentioned, consist of compactness of the record

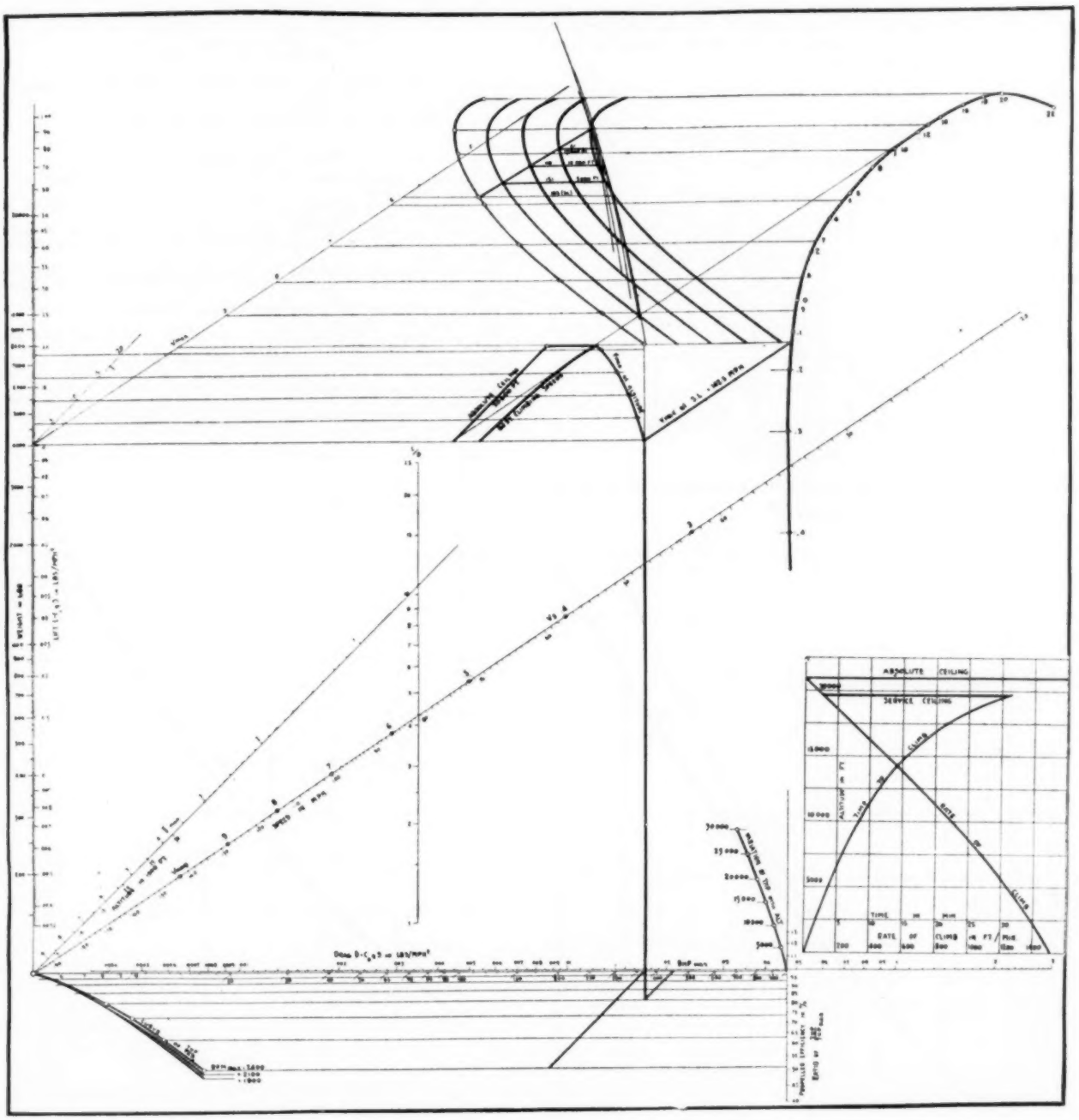


FIG. 45 AN EXAMPLE OF THE USE OF THE LOGARITHMIC POLAR AS EMPLOYED IN ROUTINE PRACTICE

and a ready means of estimating slight changes of design. In many instances lack of time does not allow an elaborate revision of a performance estimate and in such a case the logarithmic polar diagram furnishes an easy and rapid estimate of the effect of changes in weight, parasite resistance, etc. To the trained eye the graphical picture of the polar gives other valuable information such as the general efficiency of the design, the approach to inefficient gross flying weight condition, and such.

RANGE AND ENDURANCE

Breguet:

$$\text{Range} = 863.5(L/D)(\eta/C)\ \log_{10}\ (W_0/W_1)$$

$$\text{Endurance} = 750(V_{C_0}/\sqrt{W_0})(L/D)(\eta/C)\left[\frac{1}{\sqrt{W_1}} - \frac{1}{\sqrt{W_0}}\right]$$

where η is the average propeller efficiency, C the average specific fuel consumption, W_0 the initial gross weight, W_1 the final gross weight, and V_{44} the cruising speed at the gross weight W_2.

Diehl brought the Breguet formula into a simplified form and also obtained his own expressions for range and endurance where instead of average values of specific fuel consumption the initial specific fuel consumption is used.

$R = B_R(L/D)(\eta/C)$
$T = B_E(L/D)(\eta/C)(1/V_{C_0})$
$R = K_R(L/D)(\eta/C_1)$ (no mixture control)
$T = K_E(L/D)(\eta/C_1)(1/V_{C_0})$ (no mixture control)
$R = K'_R(L/D)(\eta/C_1)$ (with mixture control)
$T = K'_E(L/D)(\eta/C_1)(1/V_{C_0})$ (with mixture control)

Figs. 46, 47, and 48 give the factors appearing in the foregoing formulas.

It must be noted that in the case of the Breguet formula one must use the average specific fuel consumption C, which is obtained by calculating the average ratio of maximum to minimum speed and using Fig. 44.

In applying Diehl's formula for range and endurance, it is necessary to use the *initial* specific fuel consumption. In this case ratio of C/C_0 from Fig. 44 must be obtained for the initial value of the ratio of maximum to stalling speed.

Take-Off and Landing Run

Take Off Run (Diehl):
Neglecting the influence of the propeller

$$S_o = KV^2_s(W/\text{b.hp.})$$

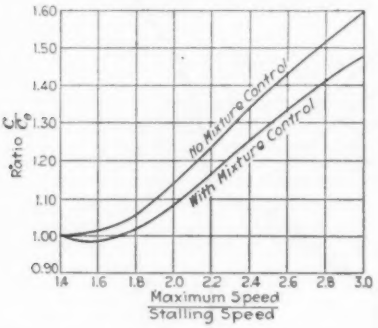


Fig. 46 Variation of Specific Fuel Consumption With Speed Ratio $\frac{V_M}{V_S}$

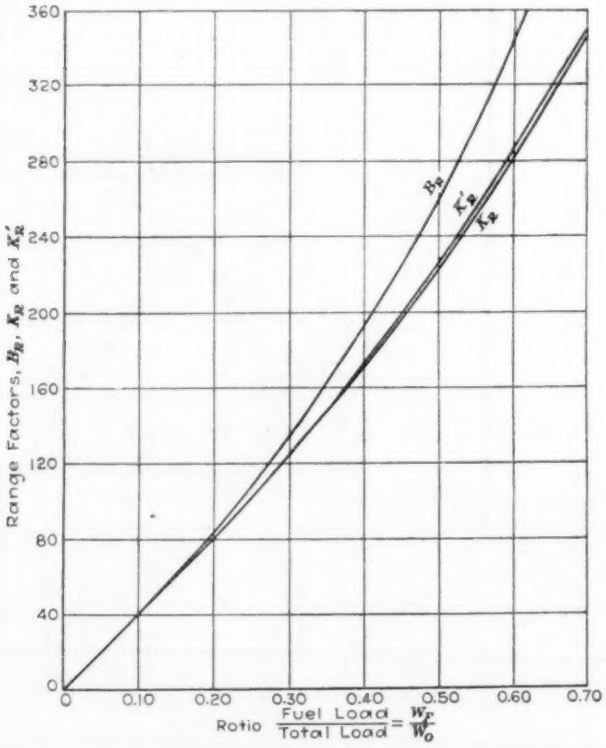


Fig. 47 Factors for Simplified Range Formulas

where S_0 is the take off run, V_s the stalling speed, W/b.hp. the power loading.

The average value of K is 0.012. More precisely, the foregoing equation can be written as follows:

$S_0 = 0.0215\ V_s^{2.25}W/T_0$, where the static thrust $T_0 = 6000[18.7 - 9.5(p/D)]$b.hp./(r.p.m.$D$).

Landing Run:

$$S_L = \frac{V_L^2}{2g(D/L - \mu)} \log_e \left(\frac{D}{L\mu}\right)$$

where S_L is the landing run, V_L the landing speed (feet per second), and μ the coefficient of friction.

The average value of μ is between 0.06 and 0.10.

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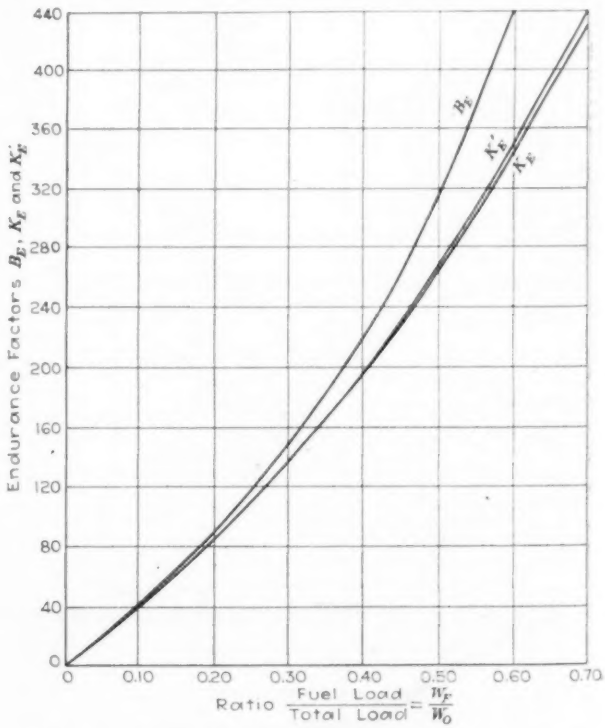


Fig. 48 Factors for Simplified Endurance Formulas

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AER-52-25

Reduction of Airplane Production Costs

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AS RAPID and uniform reproduction is the essence of efficient manufacturing, production economies will necessarily be limited in the airplane building industry until the yearly output reaches the five- and six-figure class. Few large commercial airplane factories today are operating at the capacity of which they are capable. The demand does not justify it. Thus many of the benefits of heavy production are lost with their attendant blessings of lower cost per unit, lower retail prices, and broader markets.

However, large economies may be effected in airplane production through the utilization of efficient factory planning, modern machinery, and all manner of labor-saving devices. As in the manufacture of any product, a short cut to efficient production lies in exact duplication of every part so that it is interchangeable with every similar part in every unit.

The public does not realize the amount of the engineering, high-grade material, and skilled labor which go into an airplane. Whereas the modern automobile's lines are designed chiefly for beauty, an airplane takes shape in accordance with aerodynamic principles, and the design of an airplane plays a much greater part in determining its performance than does the design of the chassis, engine hood, and top in the characteristics of an automobile. Fortunately for the airplane industry, the better structures and lines from an aerodynamic standpoint are usually the most beautiful.

The airplane engineer, in order to avoid compromising to a large extent on either strength or lightness, has developed scientific structural design to an extreme degree. As a result the factory craftsman is put to a great deal of painstaking detail in order to insure that the parts will not be overweight and yet will bear the designed stresses. Nevertheless, machine methods can be used in many operations with an immense resultant saving in production costs combined with the benefits of a more perfect and more interchangeable product.

In factories operating on a fairly heavy schedule, labor- and time-saving machinery will prove a quick turnover investment. A few instances of machine-made savings are given from those used in the factory with which the author is connected.

Before a molder was installed in the wood shop, the routing of spars required 16 hr. a day on the shaper. It was necessary to run the front spar through the shaper five times and the rear spar three times, while the rib cap strips required five such operations. With the installation of the power-driven molder the front and rear spars were shaped with a single operation each. The spars were molded into exact size, whereas under the hand-feeding process the spars were approximate in shape. The machine-molded spars thus could be quickly and easily placed into position in the jigged wing skeleton.

Instead of producing one cap strip in five operations as under the hand method, now five cap strips are turned out in one operation. The net result is a reduction to 4 hr. of work formerly entailing 16 man-hours of expensive labor.

In other departments as well as in the wood shop large factories will find machine methods a paying proposition from the twin standpoints of economy and quality of product. In the Eagle-rock plant an electric cutting machine in a single operation cuts sufficient cloth to cover $10^1/_2$ biplane wings. Cloth is sewed into shape with an electric sewing machine which double-stitches the fabric in one motion and is capable of 3300 stitches per minute. Women employees handle the sewing neatly and efficiently.

Every factory in time develops small time- and labor-saving innovations which are an important factor in the profit margin on a year's production. A device is used to facilitate the work of waxing thread with which the fabric is sewed and laced to the wing structure. This is accomplished by the simple procedure of reeling the raw thread from one spool to another by means of an electric motor. Between spools the thread passes through a pan of molten wax, and then a wiper regulates the proper amount of coating.

For wing covering, 100 layers of fabric are placed on the table and cut into proper lengths for sewing with a minimum of waste. Cutting is done with an electric cutter of the vibrating-blade type. The number of covers for fuselage and control surfaces which may be cut in one operation is far greater than is necessary to stock at one time.

Wing covers are sewed in an envelope, with the aileron recess and butt of the wing left open. Likewise covers for all other parts are sewed to be slipped over the framework as a one-piece case. Sewing the envelope in advance not only permits a better job of covering, but effects a tremendous man-hour saving. One machine operator consistently can turn out 32 wing covers and fold and put them away in the racks every 8-hr. day. This one process again illustrates the labor economies effected by use of modern machinery and proper arrangement. With a crew of 16 a properly equipped covering department can turn out fully covered parts for eight three-place open biplanes every 8-hr. day.

In the entire process there is no scrap which cannot be utilized. Practically all fabric for wings, ailerons, tail surfaces, center section, and fuselage can be cut from an efficiently designed pattern with an actual waste of not more than 5 yd. per 100 planes.

A standard covering job should be good for the life of the airplane although rough usage and long exposure to severe weather in time may necessitate replacements.

As true line production methods are adhered to as closely as possible, very good results are obtained with an overhead-track system connecting all departments in the factory, and enabling parts to be conveyed rapidly from the main plant to the lionoil and dope rooms which are located some distance away as a fire precaution.

In the old Denver plant two men used to carry a wing from one building to another and back. Now one man can pull 12 wings over the trolley system in less time, resulting in a man-hour saving of 23/24 of the time required under the former procedure. Because of the trolley system one man can handle wings in the dope room without help, cutting labor costs on the operation in two. The overhead transportation system also expedites installation of power plants previously tested as a unit in the fuselage while the ship is going through the assembly line. The 6000-ft. overhead-trolley system has paid material dividends.

A large saving was effected by the introduction of a lionoil wing-dipping vat. Formerly painters spent three hours hand painting a single wing skeleton and it was found that 85 per cent of the paint went into the air as waste. Now the wing is dipped in a deep vat filled with lionoil. Two men can process a dozen

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

wings in 1/2 hr. at the rate of five "man-minutes" per wing, whereas the former rate was three man-hours per wing.

A new company may determine its future profits or losses before it produces its first plane. No amount of selling can overcome the handicaps of a poorly designed and inefficiently operated factory.

Before the Alexander company began construction of its present factory in Colorado Springs, company heads, factory supervisors, and expert architects studied the planning over a five-months period. The factory was designed and redesigned 25 times before the present layout was decided on. Factory department heads stated concisely their needs and advised as to the most efficient location in relation to other departments.

Jigs are now common in large airplane factories, and result not only in economical production, but in exact duplication of each unit. The Alexander company was one of the first factories to use fuselage jigs so that the steel tubing could be welded and the fuselage turned out exactly interchangeable with its forerunners. Fuselage jigging is most efficient with a perfect interchangeability of duplicate structural members. The welder must be able to pick out the right member quickly and weld it into shape without necessity of filing and fitting it into the fuselage. Various tube members are cut to correct lengths with a power saw and then the ends are shaped so that a large quantity is available at all times, and so that the welder will lose none of his highly paid time finding and fitting the parts. He has only to pick out his part from the numbered racks in a rolling bin and snap it into place in the jigged fuselage. From 25 to 100 duplicate parts are in stock at all times, properly milled at the ends so as to fit smoothly into place. The bin is on castors so that it may be transported quickly between the machine shop and the adjacent welding department.

A jig should be more than a mere framework for the fitting of structures in proper shape. It should embody refinements known to every experienced aircraftsman which enable him to complete his job more quickly and efficiently. Fuselage jigs should permit easy accessibility to each welded joint. The tube should hold in place with minimum attention from the welder and the finished fuselage should be easily removable from the jig.

Where the design permits the use of a sufficient number of identical parts, it is economical to make dies for use on the punch press. Two or three die makers are employed and it is found practical to invest thousands of dollars in dies. Two thousand jigs, dies, and fixtures is a low estimate of the number required in manufacturing airplanes on a large-scale basis.

Until the factory is assured of selling the models in fair quantity, it is impractical to design a special motor mount for every new motor or freak motor which a prospective customer would like to install. If the new motor proves out in service and comes into wide demand, the factory can then afford and turn out the new installation. It is estimated that a new motor installation, and the tests and engineering necessary to win the complete model an approved type certificate, cost approximately $1000.

Factory workers should be encouraged to work out more efficient methods of performing their jobs and perfecting helpful devices. This activity can be stimulated by offering valuable cash prizes each month for practical ideas. A recent innovation which resulted from such a contest was a low couch on castors which enables members of the rigging crew to work more efficiently on the underside of ships.

To develop the now scarcely touched but potentially enormous general market, faster planes must be produced, inexpensive to buy and economical to operate. The initial cost to the buyer can be cut down only by a steady reduction in production costs, brought about by careful study of each operation.

AER-52-26

Airplane Design, Manufacture, and Sales, Commercial vs. Military

By REX B. BEISEL,[1] TULSA, OKLA.

The commercial industry differs materially from the military, and the paper discusses these differences from the general standpoint of design, manufacture, and sale. It would be a great deal more difficult for the commercial builder to enter the military field than for the maker of military airplanes to successfully enter the general commercial field.

NOT MANY years ago the military airplane dominated the field. When the airplane was mentioned, it was discussed purely in the military sense. Its commercial possibilities were realized only after a very slow process of development, handicapped to a great extent by the rapid progress made in the development of the military type. The early commercial airplanes were either adaptations of surplus war machines or were designed on lines closely approaching the familiar military models. However, due to a growing need for strictly commercial types, coupled with the foresight of a number of men, who now head some of the largest commercial companies in this country, the commercial airplane gradually came into its own and at the present time holds the field of importance both as to quantity produced and importance to the country as a whole. In a period of a very few years the commercial airplane has reached a stage of development comparable to the military machine. In a number of ways its development has been even more rapid. There are numerous reasons for this condition, but one which is important is the control of design and approval of types by the Aeronautics Branch of the Department of Commerce. This control has resulted in the commercial airplane being equally as safe to fly and possessing as good flying characteristics as the military plane. As far as comfort is concerned it is greatly superior.

There are, however, a number of points connected with the commercial industry which differ materially from the military and it is the purpose of this paper to discuss these points from the general standpoint of design, manufacture, and sale.

Design

In a strictly comparative sense the military plane is designed primarily to carry armament, whereas in the same broad sense the commercial airplane is designed primarily to carry freight or passengers. As a result, the requirements of each are very different. In a general way the paper will enumerate the requirements affecting the design of the military type.

High performance is the most essential requirement. Unless a pursuit plane, for example, has performance equal or better than its adversary, its value is lost. By performance is meant high speed, rate of climb, ceiling, and maneuverability. It must be easy to control, have excellent maneuverability, and yet be sufficiently stable to permit accuracy of gunfire, especially while diving. The best possible vision from the design basis could always be improved upon from the standpoint of the ideal requirements. The armament must be installed to function properly at all times and be readily accessible for loading or maintenance on the ground. In addition, provision must be made for the installation of bomb gear operated from the cockpit and for the installation of a small radio as additional equipment. It is necessary to install this equipment in a very limited space and still insure the proper function of all parts. The airplane structure must have factors of safety that will insure against its failure even though stressed to a point where the acceleration is as high as the pilot can physically endure and yet be designed with the lowest possible weight. The designer is confronted with the problem of whether to add more horsepower, resulting in more weight and a larger airplane with possible loss of maneuverability and poorer vision, or be satisfied with a smaller plane of less power. Fortunately he is not too greatly handicapped by cost, unless he happens to design an airplane wherein the cost is out of proportion to the improvement in other desirable qualities. The design of a modern fighting airplane requires more courage, ingenuity, and experience than does any other type of military craft. This is borne out by the fact that there are only two contractors in the United States who have successfully produced airplanes of this type.

The military-airplane designer must have, in addition to a thorough knowledge of general airplane design, an equal knowledge of armament and equipment installations, including fixed guns, either in the fuselage or wings, flexible guns, bomb gear, ranging from 25-lb. bombs up to bombs of 4000 lb., torpedo, radio, and camera. If he designs planes for naval use, he must be familiar with float or hull design, arresting gear, and flotation gear installation. If he is good, the application of an amphibian gear to an existing type is considered only a part of the day's work.

In order to properly understand and efficiently handle the design of military types with their numerous and varied installations, the designer should be reasonably familiar with military tactics and conditions encountered in actual wartime service. He should know that it is quite impossible to stand up and efficiently handle the flexible guns in the rear cockpit of a two-seater at a speed greater than 125 miles per hour. Knowing this, he should be able to devise ways and means of shielding the gunner against the terrific air blast or be able to design a special gun mount to overcome this difficulty. He should know when installing the fixed guns on a fighter that the airplane in diving vertically will assume a certain angular attitude with reference to the thrust line, and that unless the guns are set at this attitude they cannot be pointed at the target, and the pilot will report that the plane creeps and destroys his aim. He should know something of formation flying, especially in the case of multi-engined planes with the pilot located in the forward end of the fuselage, or he may turn out a job where the leader's vision is limited and he is unable to properly see the rest of the formation.

It is true that the services provide handbooks and drawing

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containing a great deal of useful information and instruction covering installation of military equipment, but after all the use of any handbook must be accompained by familiarity gained only by experience with the actual work itself.

From the foregoing it can be seen that the design of the military airplane differs inherently from that of the commercial airplane in the following respects:

(*a*) High cost of construction is justified if it results in speed climb or maneuverability higher than existing types.

(*b*) A thorough knowledge of armament installation as well as an intimate familiarity with military requirements is essential to a successful design.

(*c*) The horsepower is as high as can effectively be used.

In contrast to the military plane, where it is considered permissible to sacrifice everything except the proper installation and functioning of the military equipment in order to obtain high performance, the commercial airplane has several requirements, each of equal importance. The designer must give equal consideration to cost, performance, comfort, and load-carrying ability. It is true that certain special types permit of attaching greater importance to any one of these requirements, but in general they are equally important in laying out a new design. These requirements are very definitely related and affect each other so closely that they must be studied collectively rather than separately. Whereas the military designer designs around the highest horsepower and latest type engine adaptable to a particular type, the commercial designer must choose his power plant with greater care. High power means high performance, but also higher initial and upkeep costs. Higher power with increased fuel and oil load in a given airplane also means lower pay-load-carrying capacity unless the landing speed is permitted to exceed practical safe limits. Only recently has comfort received the attention it deserves. True, an increase in comfort means added weight and perhaps a decrease in performance due to a roomier compartment. Fortunately the flying public has reached the stage where a passenger is no longer paying for the thrill of having a gale of wind blown down his neck, but rather is interested in getting to a particular destination in a hurry, and he naturally and rightly demands that it be in comfort. The successful designer will in the future pay more and more attention to the comfort and appearance of the airplane. In this connection one can take a leaf from the book of the automobile designer. Automobiles at the present time are sold as much on comfort and appearance as on any other quality. It might be mentioned that the lack of appreciation of comfort on the part of the designer is directly traceable to the old military influence. The designer who gained his experience on military aircraft was so instilled with the requirements of light weight and high performance that he was very slow to accept the necessity for providing comfortable spring-cushion seats, cockpit heaters, non-breakable glass windows, adequate baggage compartments, and similar comforts and necessities because of the resultant additional weight or lower performance.

It is extremely essential, therefore, that the commercial designer not only appreciate the relative costs of different types of design, but must also be capable of designing a given type of construction such that it can be built in the cheapest manner possible. Essentially the commercial airplane must be of cheaper construction than the military airplane. Here again he must balance low cost against light weight and aerodynamic cleanness of design. In determining cost he is handicapped in not knowing accurately on what quantity production he can safely figure. He cannot count on ever being paid for the engineering and shop costs of the first experimental article as is so often true in the case of the military designer who can anticipate reasonable production if his design is a success.

The tendency at the present time as a result of market conditions, is to build cheaply almost regardless of quality. The argument which comes from the sales department is that the way to sell airplanes is to sell them cheaply. The easiest and simplest way is to sacrifice quality. This is a condition which presents a new problem to the designer. He should face it squarely and continue to insist on quality of design. If any structure or vehicle of transportation requires quality of design, none requires it to so great an extent as the airplane. Rather than sacrifice quality he should concentrate on evolving a simpler design and if necessary reduce cost through the use of lower horsepower. The problem, as the author sees it, is not to unload airplanes on the public, but rather to educate the public to the necessity, convenience, and safety of the airplane. This certainly cannot be accomplished at the present time by inferior quality.

The military designer must be a specialist on armament installation. The commercial designer also must be familiar with details other than the design of the structure of the airplane and its component parts. He not only has to contend with such problems as stability, controllability, especially at low speed, fire hazards, and easy-riding qualities while taxying, but must also be thoroughly versed on coach design, interior finish, heating and ventilating, and other similar requirements affecting the comfort of the passengers and crew. It is extremely necessary that a great deal of attention be given to these latter items. It is interesting to realize that the extensive use of brakes and even tail wheels on military airplanes was adopted only after being successfully developed and used on the commercial airplane.

A problem which has not yet received sufficient attention on the part of the commercial designer is the prevention of noise and vibration. Passing out cotton to be stuffed in the passengers' ears shows thoughtfulness on the part of the transportation officials, but is certainly unsatisfactory and only a half-hearted solution of the problem. Slower propeller-tip speeds, exhaust mufflers, and insulation of cockpits against noise and vibration must be studied and adopted by the designer if air transportation is to be a success.

Manufacture

At the present time the manufacturer of military airplanes on the whole requires and uses more extensive equipment and employs a higher grade of experienced personnel than is true of the commercial manufacturer. Basically, the military airplane is the last word in refinement of design and light weight and is the chief reason for the foregoing condition. Equipment such as heat-treat furnaces for both steel and dural, sand-blasting equipment, swaging machines, numerous lathes, milling machines, grinding equipment, several sizes of punch presses, hydraulic presses, shapers, tube-bending equipment, special woodworking equipment, and special equipment for fabricating parts from dural and sheet-metal parts are essential in the manufacture of military aircraft. The foregoing equipment is used in the manufacture of commercial airplanes, but for the most part only by manufacturers who are contractors to the U. S. Government. The tendency in commercial construction to build economically limits the use of special equipment to as small an extent as possible, especially in the case of equipment used primarily as a means of weight reduction in the airplane with a resulting increase in cost.

The following are a few examples of design and construction which, although permissible on military aircraft, are prohibitive on commercial aircraft if they are to be built at a figure low enough to insure sales:

(*a*) Beams fabricated of dural sheet employing either box or latticed construction assembled and often constructed almost entirely by hand.

(*b*) Comparable with this construction is the tubular steel beam with latticed tubular webs which is heat-treated after assembly.

(*c*) Fuselage of monocoque metal construction requiring a great amount of hand forming and hand riveting.

(*d*) Fittings and parts requiring extensive machining operations and heat treatment as a means of weight reduction.

A number of planes especially in the larger types or high-performance class embody a number of expensive manufacturing processes in their construction. Such planes, however, are being sold to operating companies, and the number that have been sold or that will be sold is limited. It should be recalled that, with the exception of airmail carriers, none of these companies is making money, due in part to the high cost of flying equipment. The class of people to whom airplanes must be sold in order to insure quantity production are the individual or small-firm owners. This class will not buy unless the price is low, which naturally means inexpensive construction. It is quite true, of course, that the present expensive construction could be very appreciably reduced in cost if produced in sufficient quantity. However, there are numerous arguments against large-quantity production. In the first place it is problematical whether there would be a sufficient market for the quantity production necessary to permit of an appreciable price reduction. The building up of a market must be a gradual process of education, familiarity, and understanding of the possibilities of the airplane on the part of the public, accompanied, as the market grows, by a similar increase in production and by improved methods of manufacture. There is ample precedent in other branches of industry for the truth of this statement. Another reason against extensive tooling and jigging in the manufacture of a certain model is that production over a relatively short period of time will inevitably fall off due to the airplane becoming obsolete as a result of the constant changes in the art.

Even though the production of both military and commercial aircraft is relatively small, the commercial manufacturer has one distinct advantage over the military manufacturer in production control. This is the matter of changes. The Army and Navy are realizing more and more the condition which results from introducing changes during the course of production. However, the military manufacturer still has this problem to contend with, and it still causes him many a sleepless night. In the majority of cases, the cost of changes in military airplanes is paid for, but the actual final cost to the manufacturer can hardly be calculated. Production is a process comparable to the workings of a machine where the assembly of each part of the airplane can be likened to a cog in a train of gears. To disturb the workings of this machinery it is only necessary to throw one of these cogs out of mesh and it will have its effect on nearly every other cog in the train. The author has seen a whole production line held up because of a slight change in mounting the fixed gun. Until this change was completed the entire fuselage installation, as well as the assembly of landing gears, wings, and tail was held up. Not only were production schedules disrupted, but other work had to be found for the idle units affected, resulting in still additional changes in the program. Inasmuch as a military airplane demands the utmost in perfection, changes in design and construction are often excusable, but it is always possible to keep these changes down to a minimum by careful testing of the experimental or previous production articles. It often happens that a change made without service testing proves less desirable than the original part or assembly. Changes invariably introduce weight because of the fact that they are not an inherent part of the original structure but rather something additional. Enough such changes may so influence the final weight of the airplane as to make it much less satisfactory than in the original state.

In the case of the commercial airplane the company may thoroughly test the experimental article to its own satisfaction. It is possible then to run a predetermined number of airplanes through production with no changes whatever. At least the company itself is master of the situation, deciding whether a change is absolutely necessary from the possible effects on sales and is not influenced by a definite decision from the customer. Previous to the release of a new production schedule all necessary changes can be incorporated as a part of that production at the very beginning of the schedule, thus preventing any holdups later.

Another factor which makes the cost of military airplanes higher than commercial airplanes is government inspection. On the whole, government inspection is desirable and often very necessary. However, it does affect the cost of the finished airplane in that it introduces another cog in the train of gears. It is impossible to prevent production holdups due to delay in either government or company inspection. With no control over the government inspection system the contractor cannot affectively speed it up by pressure or overtime work. It often happens that the subject of rejection is of such a nature that even after a protracted discussion it is difficult for the resident government inspector to make a final decision, and it is then necessary to hold up production awaiting a final decision from headquarters, which naturally takes time. If the part in question were for a commercial airplane it could be rejected at once. Even though slightly defective or not in accordance with the drawings, the company could use it if it did not too seriously affect the final quality of the product. Or the company could balance the cost of rejection against the final effect on present or future sales of the airplane. In comparison, if the government is paying for a certain specified article, it naturally expects to receive that for which it has contracted.

Materials used in airplane construction also affect the final cost. All military airplanes are built from materials meeting rigid government specifications. This material being of the highest quality, naturally commands a higher price. The commercial manufacturer can here again carefully compare the use of certain materials with any possible effect on sales and many find it desirable to use less expensive material without feeling that it reduces in any way the quality or value of the finished article. As often happens each of two articles, accessories for example, may be equally good from the standpoint of the manufacturer, but only the one approved by the government can be used on military airplanes. The A. & N. standards have been a great help in simplifying the procurement of material for government contracts. It is believed highly desirable for commercial manufacturers to adopt these standards wherever possible—first, because the material used on commercial airplanes should be, so far as strength and safety requirements are concerned, the best obtainable, and, secondly, standardization in material and parts simplifies procurement and reduces costs.

Sales

Perhaps the chief difference between military and commercial sales is that in the former the sales department deals with a large organization while in the latter it deals with an individual. Commercial airplanes are bought by organized transport companies, but even in this case only a few individuals connected with the company are concerned with the actual purchase of the airplane. Designing and selling airplanes to separate individuals, each with a different idea as to what he wishes to purchase, is quite different from selling to a large organization which desires a particular type for which there are definite specifications as to load carried, performance desired, and even the motor required.

In commercial sales it is necessary for the sales department to

determine upon one or more types of airplanes to be manufactured, without, in the great majority of cases, even consulting the customer as to his wishes or desires. It is necessary for the sales department to anticipate them and gage, as carefully as possible, the price at which a particular model will be salable as a production article. It is then necessary for the engineering department to design accordingly. Even though the airplane when built is not entirely up to expectations or is inferior to the product of a competitor, the sales department still has a reasonably good chance of making individual sales either through expert salesmanship or through the inability of the purchaser to pass expert judgment on the article he is buying. The purchaser of a commercial airplane must rely, to a very great extent, on the integrity of the salesman, except possibly with respect to flying qualities which, more often than not, are determined with a light load. It is impractical to put the airplane through a long series of tests to obtain performance and detail flying qualities since the time required in a demonstration test is usually a matter of minutes rather than days and possibly weeks. The average commercial customer is much more apt to be impressed by a good line from a clever salesman than by anything he can learn in a fifteen-minute flight, especially if the airplane is being demonstrated by the salesman who through experience can make a so-called "crate" handle like a pursuit ship.

The sales procedure which will bring the largest number of ultimate sales must be predicated on a high-quality airplane, having the best possible characteristics. This can be done by very careful design, testing, and changes. It will first of all sell the salesman, who will naturally be enthusiastic. If in addition he is armed with honest data as to performance, a reputation can be built up which cannot fail to insure ultimate sales volume.

Military sales on the other hand are conducted quite differently. As a general rule airplanes are purchased on a basis which permits the service to carefully consider one or more articles of a particular type. Unless the airplane to be constructed is built entirely on "his own" by the contractor, which happens only in the minority of cases, the design is carefully considered either by a board of officers or by various officials who are considered experts in their particular line. Specifications embodying, in considerable detail, exactly what the service demands or requires are drawn up for the contractor to follow. The contractor's engineers carefully discuss the preliminary plans submitted to the service with the board or corps of experts and a very definite understanding of the airplane to be constructed is obtained. The contractor and purchaser are then in a very close agreement as to the exact requirements and just what product is contemplated. Such selling is basically through the engineering rather than the sales department. Aggressive sales ability in the usual sense is of little avail unless the engineer himself can sell his job to the engineers of the service purely on the merits of his design and on his ability to produce. A language is spoken which is clearly understood on both sides and facts are the only arguments that count. A somewhat unsatisfactory condition exists in the calling of a board made up of a group of officers experienced with respect to the particular type being considered for approval, either in the preliminary design stage or as the first experimental article. Although the members of such boards invariably constitute the pick of the service in so far as judgment and ability are concerned, they are seldom permanent members. For example, a board may be called to approve the "mock-up" and preliminary designs of a newly proposed type. Later, when the first experimental article is tested, which may be after a period of a year or more, a new board is called which unfortunately often consists of an entirely different group of officers. They are very apt to be a bit prejudiced with reference to the findings and recommendations of the previous board, especially so if the findings and arguments of the latter are not carefully discussed or considered. Changes are recommended which could not meet the approval of the original board. This results in an unfavorable condition in so far as the contractor is concerned, especially if considerable design modification is necessary. If it were possible to pick at least 60 per cent of the members of each new board from members of the old board there might be a decided improvement in the foregoing condition.

Even though the contractor decides to build an airplane entirely on "his own," he invariably obtains unofficial approval of the design by determining either through contact with various competent individuals in the service or by actually submitting preliminary layouts and specifications for general approval and criticism.

The procedure of building on "his own" is very desirable from several viewpoints. In the first place, the contractor is able through his engineering organization, if the latter possesses the necessary experienced personnel, to obtain a very good picture of the service requirements. This can be done without being unduly handicapped, in his desire to turn out an acceptable article, by the individual opinions of a few official experts whose duty it is to make decisions. This statement is not meant to be derogatory in any way. It is quite evident that the acceptance of a particular type, in so far as quantity production is concerned, is dependent on the opinions of a greater number of persons who will use the airplanes under actual service conditions. It should be easier, therefore, for a competent engineer, entirely unprejudiced, to intelligently arrive at definite conclusions with reference to airplane requirements rather than to depend upon the judgment of a relatively few individuals. It may also be the case that the contractor has a particular "hot" idea, the worth of which can only be proved by actual construction of the airplane. He may consider it well worth while to gamble a certain amount of money on the strength of his belief.

Further, when building on "his own," the contractor is not affected by government inspection. The article will later be approved from this standpoint, especially in so far as it affects later production. But if he is experienced, neither the contractor nor the service itself need be unduly concerned with this point. Also, the contractor is freed from the necessity of meeting certain delivery dates. However carefully these dates may have been estimated, any error is naturally on the low side, and either a great deal of embarrassment results or the job is rushed through too quickly, with the usual result of overweight, poor design, and poor workmanship and finish.

Another advantage of building on "his own" is that the contractor retains all design rights at least until after the completion of a production contract. Inasmuch as he is then all tooled-up for production he should be able to put in a low bid on new production and still make money. An extremely unfortunate condition exists in the official laws controlling the letting of government contracts. They require under certain conditions that the contractor sell his design rights, with the first experimental article, which may contain a number of features actually classified as or bordering on inventions. In case his bid was low on the latter, as a result of keen competition and in order to be sure to obtain the contract, his production bid may have to be somewhat high in order to make up for experimental losses. In the event that a competitor bids lower on the production contract he gets the bid without having to stand the original engineering costs and can be reasonably sure of making money on the contract. Many offer the argument that the contractor uses poor business judgment in making his original experimental bid low simply to get government business, even at a loss. A recently published opinion by an official of one of the large aircraft corporations refers to this practice as "cut-throat competition." No doubt there is

some basis for such a remark, but on the other hand it is equally true that a contractor may find that his bid was in error due to his not being able to estimate accurately on certain new design features or types of construction. Rather than turn out an inferior article he accepts the loss and turns out a creditable but expensive airplane only to find out later that he has lost a much hoped-for production contract. In any case the situation, as it exists, is not conducive to developing the very best in military aircraft unless we agree that this can be accomplished by the contractor building on "his own." Even this procedure is apt to be a gamble.

In conclusion it is believed that it would be a great deal more difficult for a commercial manufacturer to successfully enter into the military field, assuming no previous military experience on the part of the personnel, than for the military manufacturer to enter into the commercial field even though inexperienced in this branch of the industry. This would be especially true in the latter case if production were limited to the higher-priced, higher-performance class of airplanes.

AER-52-27

The Construction and Operation of Gliders

By GILBERT G. BUDWIG,[1] WASHINGTON, D. C.

The construction, operation, and maintenance of gliders present problems for which there is little precedent. From the regulatory standpoint, the rules of the Department of Commerce will permit licensing of those built prior to October 1, 1930, by passing a satisfactory visual inspection and test. After that time, the construction of gliders will come under more severe rules for licensing.



THE glider situation is a rather comprehensive subject for discussion, and it is difficult to determine what phases of this rapidly growing activity should be reviewed here in order to serve the most constructive purpose. The construction, maintenance, and operation of gliders present problems for solution for which there is little, if any, precedent. However, in light of experience, certain conclusions suggest themselves and it is upon these points that this paper will be based. Early this year the general interest in gliders indicated the necessity for constructive regulation. The Department of Commerce undertook the problem, and as a result, certain amendments to the Air Commerce Regulations, including the Airworthiness Requirements, have been made. As all are probably aware, the policies that the Department has adopted are a direct result of innumerable recommendations and suggestions which were received from those interested. In addition, a number of conferences, both formal and informal, were held with a view of determining how the glider situation could be best handled from a regulatory standpoint. It was finally decided that an amendment to the Airworthiness Requirements, so far as gliders and light aircraft were concerned, was advisable. Therefore, suggested amendments were circulated upon which comment was invited, although a provision was made that the suggested amendments could be used in the interim as a basis for obtaining Approved Type Certificates for such types.

However, due to the substantial number of gliders which have already been built and are in operation, and the inability in most cases of the manufacturer or builder to furnish drawings and stress analyses to the Department for approval, gliders built prior to October 1, 1930, will be licensed on passing a satisfactory visual inspection and test. Gliders built after October 1, 1930, must, however, come under the following three general classes to be considered eligible for license:

1 Gliders manufactured under Approved Type Certificates. This corresponds to the present procedure involving the licensing of airplanes which are built under Approved Type Certificates.

2 Gliders constructed by any one provided they are built in accordance with specifications and designs previously approved by the Department of Commerce. This permits the construction of gliders by any one whosoever, provided they are built in accordance with an Approved Type Certificate and specifications.

3 Gliders not manufactured under Approved Type Certificates, but which meet the requirements for such a certificate. This covers gliders which meet the requirements, but for which an Approved Type Certificate is not desired, perhaps due to production being discontinued or the construction of a single glider only.

In view of the great interest shown by the general public in the use of gliders, the type of operation to which the glider will be subjected and the more favorable weather conditions under which a glider will be flown, the Department felt the qualifications and restrictions of glider-pilot licenses should be more lenient than those of any class of airplane pilot. The following are the requirements set forth in the Air Commerce Regulations to cover such glider pilots:

An applicant for a commercial glider-pilot's license must pass the present physical requirements for a private-airplane-pilot's license, and in addition, must satisfactorily pass a flight test consisting of normal take-offs and landings, a series of gentle and moderate banks, precision landing, and 360 deg. turns both to the left and to the right. The holder of this grade of license is entitled to fly commercially and to instruct students for hire in licensed gliders.

A non-commercial glider-pilot's license has been established for which there is no physical examination, and such licenses will be issued to those who are able to demonstrate their ability to pass the flight tests involved. This license will entitle the holder to fly licensed gliders for pleasure or sport, but not for any commercial purpose. They cannot, of course, instruct for hire. The flight test required for this grade of license will be the same as that for a glider-pilot's license with the exception that the 360 deg. turns will not be required.

The glider-student-pilot's permit issued by the Department will enable any person to take instruction in licensed gliders from a licensed glider pilot. These permits are issued by supervising inspectors or field inspectors of the Department to any person upon application.

There is no physical requirement for this permit nor for the non-commercial pilot's license other than having the applicant sign a statement to the effect that he or she has good vision in both eyes either with or without glasses, complete use of both legs, both arms, and does not suffer from any constitutional disease or has ever had convulsions or fits.

Design and Construction

As stated previously, in light of increasing experience, a revision of the Airworthiness Requirements was advisable. The design load factor specified in the proposed amendment—6 for the

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[1] Director of Air Regulation, Department of Commerce. Mr. Budwig was born in 1895 at Cleveland, Ohio. He has been connected with commercial aviation since 1916 and was employed during the World War as civilian instructor stationed at Chanute Field, Rantoul, Ill.; Kelly Field, San Antonio, Tex.; Rockwell Field, San Diego, Calif.; and March Field, Riverside, Calif. He flew the air mail immediately after the war, and then did two years of "barnstorming," and later an amount of free-lance test piloting. Mr. Budwig became affiliated with the Aeronautics Branch of the Department of Commerce in June, 1927, as aeronautical inspector, was appointed Chief of the Inspection Service in May, 1928, and was made Director of Air Regulation on October 4, 1929. He is a member of the Professional Pilots Association and the Early Birds.

Presented at a meeting of the Aeronautic Division with the Chicago Section, Chicago, Ill., August 25, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

high angle of attack condition, 4.25 for the low angle of attack condition, 2.5 for inverted flight, and 5 for landing conditions—was based upon aerodynamic data and a study of existing types. As a result of this study, it is believed that these factors are equally applicable to all types of gliders. As will be pointed out later, however, it is not always possible to design the structure and detail parts exactly to these factors. To accommodate handling on the ground and to provide for unusual landing conditions, it is generally necessary to make the fittings and their component parts considerably larger than would be necessary to meet the minimum requirements. It is felt, therefore, that these strength requirements should be considered as an absolute minimum, particularly for training types.

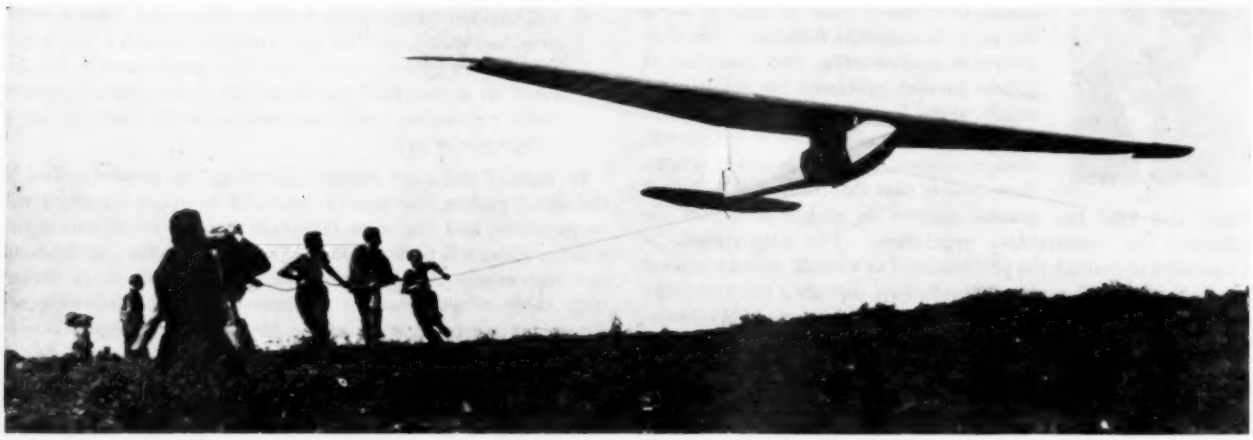
As far as structure and aerodynamics are concerned, the problems encountered in glider design are fully commensurate with those of the smaller airplanes. Aside from the stalling speeds involved, there is little or no difference between a glider flight and a "dead stick" landing with powered aircraft. It is essential, therefore, that the same degree of structural strength and aerodynamic control be provided for the former as for the latter. Although the structure of the glider is possibly somewhat simpler than that of the airplane, the problem of control, particularly lateral, is more difficult of solution.

In the case of the soaring glider, moreover, one may venture to state that its design is an art comparable with the best efforts in airplane design where weight, control performance, and aerodynamic finesse play such an important part.

Wings

In the case of primary and secondary training types, the conventional two-spar rectangular plan form of construction predominates. Due to the fact that these gliders are flown and as a general rule handled by beginners, it is essential that the wing structures have ample strength and rigidity. This is accomplished by incorporating in the design a double system of drag-wire bracing, liberal use of plywood at wing tip and wing butt, a substantial aileron false spar, adequate wing-tip skids, a well-reinforced leading edge, and a taut, securely attached covering. Wing panels incorporating all of the above-mentioned features have been constructed with a weight of less than 0.60 lb. per sq. ft. The wing covering should be provided with inspection doors or transparent panels at critical points so that damage occasioned by a hard landing or by dragging a wing tip will be discovered before the glider is flown again. The wing-tip skids should be deep enough to protect the aileron and should be so designed that they will fail before any damage is done to the spars. The wing-attachment fittings must be designed with an ample margin of strength to resist the drag loads imposed by wing-tip landings. In the case of the primary training type, these drag loads will materially exceed any load that would be imposed upon the fitting in flight. It is advisable to have the cabane fitting in the landing-wire system so designed that it will fail before the attachment fitting at the outer strut point pulls out. The reason for this is that it is a much simpler proposition to repair the fitting at the cabane than it is to remove the fabric and replace the fitting at the strut point. Although it is not a requirement of the Department, it is strongly recommended that when hard wire is used for the external bracing of the wings, double wires be used throughout.

The more advanced types, i.e., the soaring gliders, use single-spar construction or closely interbraced two-spar design. Due to the high aspect ratios and large spans involved, torsional rigidity becomes a very important matter. This is ordinarily obtained through the use of a reinforced leading edge in combination with the spar or spars which are usually of the box type



Launching a Glider

Fuselages

Most glider fuselages are constructed either of spruce and plywood, or of steel tubing in either a single central truss or along the lines of conventional airplane fuselages. Here again the requirements of rigidity and provision for handling will determine the design of a large portion of the structure of primary- and secondary-training type. In the case of soaring gliders, due to the greater proportion of tail-surface area to size and weight of glider, the design of the structure after the wing will ordinarily be determined by the required tail-surface loads. The proposed amendments to the requirements specify a loading of 12 lb. per sq. ft. for horizontal tail surfaces and 9 lb. per sq. ft. for vertical tail surfaces. It has been found that these loadings result in a fuselage structure that possesses sufficient rigidity and stands up well under service conditions. In the construction of the rear portion of elementary training-type gliders, the design should be stronger than is necessary to support the tail-surface loads. In the design of the forward portion of a glider fuselage, the most important element to be considered is the protection of the pilot. He must be protected from hard landings, side-slip landings, striking obstructions, and nosing over. In order to accomplish this, the structure around the pilot should have an ample margin of strength. The vertical members just behind the pilot must bear the brunt of hard landings. If a wooden support is used, it

should be well taped and doped to prevent splintering. The structure underneath the pilot, including the landing gear, should be capable of withstanding severe side loads. In order to protect the pilot's legs, the skid proper should be extended well forward. Experience to date has shown that a slight additional weight in the structure at this point is of considerable value from the standpoint of safety.

Control Surfaces

As gliders are operated at speeds considerably lower than those encountered in powered flight, it is obvious that, in order to provide adequate control, the ratio of control-surface areas to wing areas must be considerably greater. This is particularly true in the case of rudder and ailerons. This increase in size introduces the problem of rigidity. In the case of training types, the horizontal and vertical tail surfaces are usually externally braced, and it is essential that this bracing as well as the attachment to the fuselage structure be carefully designed. Because of the use of automobiles for towing, it is necessary that the tail surfaces be investigated for flutter. As towing speeds in the neighborhood of forty to fifty miles per hour are often reached, speeds in excess of these values should be used in testing for tendency to flutter.

When wing-tip ailerons are used, care should be taken to investigate the torsional rigidity of the actuating tube and particular care must be taken to protect the aileron from damage in handling or in striking the ground.

Control System

There should be no hesitancy on the part of a designer to provide ample margins of strength of cable, fittings, and horns in the construction of a glider-control system. The handling and disassembly that a glider undergoes out in the field would prevent a theoretically designed system from being reliable very long under service conditions. The same care to eliminate friction and wear that is observed in airplane practice should be applied to glider controls. The use of standard turnbuckles, shackles, pins, and bolts, is advisable. Although the control system may be greatly overstrength at different points, it will be found that the confidence it inspires in the pilot will amply repay the extra expense and material involved.

Equipment and Furnishings

The strength of the safety belt and of its attachment to the primary structure is one of the most important safety features. The Department's requirement of the strength of 850 lb. for the belt and its attachments is considered a minimum. It is desirable also that the seats provide some lateral support for the pilot in order not to throw eccentric loads on the safety-belt attachment.

Considerable care should be exercised in the design of the towing hook and its release mechanism. The release lever should be conveniently located, easily operated, and positive in its action.

Maintenance

The maintenance of gliders, with the exception of freedom from power-plant troubles, is similar to that encountered in the operation of airplanes. However, due to rougher handling and the operation of gliders over less favorable terrain by less experienced personnel, it is obvious that gliders are subject to a greater amount of minor damage. They should, therefore, be more frequently and adequately inspected. The Department has taken recognition of the repair problems involved to the extent of permitting minor repairs to be made under the supervision of a licensed airplane mechanic without the necessity for the usual departmental inspection; that is, provided a notation to this effect has been made in the log book over the signature of the licensed mechanic responsible.

If, however, the damage to any component, such as the fuselage or the wing, exceeds 50 per cent in the opinion of the inspector, the repairs must be approved by a Department inspector before the glider again may be operated. As in the case of minor damage, proper notation shall be made by the responsible mechanic, and in addition there must be an endorsement by the approving Department inspector.

This liberal policy with reference to the repair of gliders was purposely established so as to facilitate the repair of gliders by their operators. Among other things, the policy eliminates the submission of technical data for the approval of repairs as is now the case in the repair of aircraft. It is believed that more frequent and adequate repairs will result from this facilitation of approval.



Glider in the Air

Operation and Student Training

Too much importance cannot be placed on the necessity for intelligent glider operation and there is definite need for improvement throughout the country in the methods used. There is also a somewhat general disregard for fundamental safety requirements. Specifically, there are innumerable cases of unqualified pilots who are violently catapulted or towed by auto-

mobiles to hazardous altitudes. This practice results not only in bodily injury and occasionally with fatal results, but in considerable damage to gliders as well, all of which has a decidedly detrimental effect on the glider movement.

Experience dictates that a student, before being allowed to leave the ground in a glider, should be given a thorough course in "ground flying." This generally consists of placing the student in a primary-type glider faced into the wind, on level ground, thus permitting him to practice the use of aileron control. This can be done in an average primary glider with a wind of from ten to fifteen miles an hour. By this method, a student is able to learn the feel of the aileron control by moving the stick from side to side and observing the action obtained. Following considerable practice at this, he may then be shown the use of the rudder and the ailerons at the same time, the coordination of these controls under conditions analogous to that during actual flights. When it is assured that the student has a reasonable knowledge of the effect of these controls, together with a fundamental knowledge of the use of the ailerons, his training may proceed.

The method which has apparently been the most satisfactory in this country for training from this stage seems to be conservative, well-directed automobile towing. However, at this stage of instruction, great care should be exercised and too much importance cannot be placed on the skill of the person handling the automobile. The student should be given innumerable trips along the ground at speeds insufficient to permit him to rise, for the purpose of teaching him the coordination of the controls involved. It is only after a thorough training in the use of the controls that the car should be run at speeds sufficient to enable the glider to leave the ground for even a short distance. When, however, it is decided that the student has sufficient knowledge to permit short jumps, the automobile or a catapult may be used for the purpose. Very short flights should be made and at very low altitudes, probably not exceeding five or ten feet, until the student has learned to some degree the hazards involved in the incorrect operation of any of the controls. Catapult launchings should be skilfully handled and the altitudes to which the glider can reach should be gradually increased. Unusually forceful catapulting should be restricted to experienced pilots.

When the student has reached the stage where he can be towed or catapulted to altitudes of twenty-five feet or more, he may then be permitted to practice gentle turns. This will enable him to familiarize himself with the peculiarities of the controls and the difference in control at various speeds. Great care should be taken by the instructor in charge of these flights to impress on the student that speed is essential to flight and that where there is no speed there is no control. By this time, the student should begin to understand the principles of flight and the advisability of smooth flying. In view of the fact that practically all gliders built in this country have relatively fast elevator action and relatively slow rudder and aileron action, the judicious use of the elevator control should be stressed. A student's advance in his instruction from this point is largely a matter of experience during which time strict supervision should be maintained.

It has been our experience that glider clubs actually function but an average of a few days a month, the balance of the time being devoted to repairing the glider. This condition, for the most part, is due to inadequate instruction and to poor structural details of the gliders involved. It also should be borne in mind that the present necessity for constant maintenance is having a very decided detrimental effect on the glider movement. The responsibility in most cases lies directly with the manufacturer for not making necessary corrections in design or construction as may be dictated from operation experience. Defective material has played an important part in a number of disastrous crashes. In some cases the material used is of such poor quality that it seems unbelievable that any one familiar with aircraft construction could have permitted its use.

It seems likely that this condition will, to a great extent, be eliminated when the use of approved gliders is more general. A majority of the states require that aircraft operated in interstate commerce be federally licensed, which of course includes gliders. It appears likely that state enforcement, when applied to gliders, will eliminate unairworthy equipment.

It is apparent that consideration is being given to the manufacture and the distribution of small "kicker" engines for use with gliders, thus converting them to power equipment. There is considerable development already under way to manufacture a light engine of low horsepower for this purpose. It should be borne in mind that when this is done the glider becomes a low-powered airplane and is subject to the requirements for such craft, which are set forth in the proposed revisions to the Airworthiness Requirements.

The interest in glider flying has increased tremendously, but unless the situation is adequately and intelligently regulated, it seems likely that there will be a decrease in interest due to causes which are for the most part avoidable. It would be well and to the interest of all concerned if more attention were paid to flying characteristics and structural details of gliders and adequate instruction in their use. Also, many disastrous accidents would be avoided if the use of unairworthy gliders were discouraged.

In conclusion, it might be well to mention that the present excellent cooperation which is enjoyed between the Department of Commerce, the glider manufacturers, and various clubs and associations will go a long way toward finding a satisfactory solution for the adequate construction, operation, and maintenance of gliders.

Discussion

DONALD F. WALKER.[2] The splendid work of the Aeronautics Branch of the Department of Commerce has resulted in the establishment of sane regulations and will permit, at considerable sacrifice to the personnel of the department, of proper inspections and adequate supervision of the design, construction, and flying of motorless aircraft. The National Glider Association is pleased to avail itself of this opportunity to again thank Colonel Young and his staff for their cooperation and to ask that no opportunity be allowed to pass for urging on the representatives in the Congress of the United States the vital importance of providing the Aeronautics Branch with ever more adequate appropriations for their essential work.

In no other line of manufacturing can a slight defect in materials, a small error in design, or poor craftmanship result in so serious, indeed often fatal, accidents as in the manufacture of aircraft, including gliders. In no line of operation has the man at the controls so little latitude as regards human error as in flying. Therefore strict control must be exercised, while at the same time good judgment must be used, to see that the exercise of authority is not carried so far as to cripple or stop operations, inventive genius, and experimental research. In only one instance do the rules laid down by the department pass over the line, in the writer's judgment, and that is in the matter of requiring licensed airplane mechanics for certain repair work, and even here the department is showing such tact in administration as to remove the greatest part of the handicap.

It is true that there are few actual flying days per month for the average club. However, it must be remembered that only Saturday afternoons and Sundays and holidays are available

[2] Manager, The National Glider Association.

for flying unless the club members have flying facilities much nearer their homes than has the average group. On the other hand, minor repairs can be made at night in small workshops in the neighborhood. Yet there are too many repairs necessary.

So far, the emphasis in America has been laid on providing gliders at the least possible expense to the clubs and others interested. It is possible that the desire to lower the initial cost has actually increased the long-term cost. It is argued strongly by several clubs that the primary training gliders are too cheap; that they do not stand up well under any training methods; that they are dangerous for any but the simplest auto-towing, generally conceded as the most efficient method of training; and that they can be neither easily soared nor safely airplane-towed. While it is admitted that they are relatively easy to repair, they point out that continuous repairs afflict the owners and kill interest. It is generally conceded that a serious "crack-up" in a primary is very likely to mean the hospital or even death for the pilot. True, it can be argued that the primary is often abused and should be used for limited purposes, but when that is said it is admitted that a ship that can be used for all purposes including training and that gives higher performance and greater safety to the pilot is superior for initial purchase.

The result has been the development of the utility glider in America. The utility glider is one that can be bought as cheaply as possible consistent with qualities permitting every use by the owners.

These utility gliders have not yet reached perfection. They can always be further refined, but they are undoubtedly the best purchase today for the average club if it can raise the initial price, which is some $200 in excess of the average primary. These ships usually are on the technical border line between a primary and a secondary glider. In fact, as the utility glider increases in popularity and decreases in price, the very terms "primary" and "secondary," always arbitrary, will pass from use, and there will be stock utility gliders for everyday use and custom-built soarers for the wealthy and those of scientific mentality. Nor will the soarers in the ordinary "breaks" of weather, piloting, and luck win all the contests, although theoretically they should.

The utility gliders are now in production with wing areas not larger than that of a primary, some being tapered for higher performance and some being square-tipped for easier repair. Nearly all the manufacturers offer a choice of wings, some being larger and of higher performance than others. One can buy a small set for training. While training, they can save their money, and when they have completed their training, they can buy a bigger set for advanced work.

It is in the fuselage that the utility glider comes into its own. They are of steel tubing, enclosed, and equipped with one center wheel and sometimes with a skid. The wheel is equipped with a brake for spot-landing purposes. These fuselages are just as light as wood, and while theoretically no stronger, when they reach the point of failure, they bend and do not splinter and break. Serious accidents where such ships have spun and dived into the ground have resulted in no harm to the pilot. Only one death out of 16 accidents has occurred in this type of ship, and in that case a good pilot in satisfactory flight was seen to fall forward, hands outside the cockpit, and dive out, it obviously being a case of heart failure, fainting, or something of that kind. These ships are stressed for airplane towing and cloud and storm soaring, as strenuous a form of flying as many outside the services will ever engage in. They have been used in all these forms of flight and have held up. Captain Frank Hawks' "Eaglet" is one of them, and no glider ever put up with the punishment nor gave better performance than did that ship.

However, disregarding the "Eaglet" and its older sister, "Franklin 9491," as both were larger and more expensive ships than what is usually called a utility glider, let consideration be given only to what the little utility jobs have done. They have been used for months in auto-towing training, shock-cord training, and shock-cord licensing flights, where they secured licenses for pilots taking off from the same hills where they had tried for months to get licenses in primaries; they have been flown to the very limits of auto-towing advanced work; they have been airplane-towed successfully with the consent of the Department of Commerce; and, finally, Jack O'Meara, of the Baker-McMillen Company, hopped off at Elmira, N. Y., and made three flights of over one hour each, full soaring. What else can be done with a glider? If one has a great soarer built for certain conditions, he dare not risk it under different conditions. If one has a primary and is wise, he will train cautiously and nothing else. But one can go the whole route in the utility at less total cost because of fewer accidents, and if the insurance man is wise he will give ample coverage on life and limb and will gather in many premiums and pass out few benefits.

And, finally, if a glider can be towed by airplane, it is an airplane. The glider fraternity await only the appearance of the right motor to show how to shift quickly and easily from a glider to a light airplane.

AER-52-28

Some Improvements in Wind-Tunnel Balances Made Possible by Use of Vacuum Tubes

By FRED S. EASTMAN,[1] SEATTLE, WASH.

The paper presents some novel ideas on wind-tunnel balance design. It describes a new electromagnetic balance, which determines the current required to produce a balancing force. The claims are made, as against the motor-driven balance, that it is less costly, that it has no moving parts in the beam, is practically instantaneous in its response to sudden changes in force, can be quickly adjusted to almost any desired degree of sensitivity, and is readily adapted to recording devices to do away with the point-by-point method of obtaining curves.



THIS PAPER is divided into three parts. Part 1 deals with the difficulties encountered in the ordinary motor-driven balance; part 2 describes a means of improving such a device by using vacuum-tube control; and part 3 introduces a newly developed electromagnetic balance.

1 The Motor-Driven Balance

The automatic electric balance which is now in common use for measuring forces on wind-tunnel models may seem to be almost superhuman to any one who has had no experience with them. This is because the unsuspecting bystander sees a man press a single button, and immediately afterward several small weights begin to move to and fro on the delicate wind-tunnel balances. There may be as many as four or five of these balances operating at the same time, and each one moves its own little weight until its beam remains quite steady, with the little movable weight on one side of the fulcrum exactly balancing the aerodynamic force which is transmitted from the wind-tunnel model through a small wire to the other side of the fulcrum. As soon as all the balances become steady, the operator pushes another button, which stops any further movement of the little weights. He then proceeds in a leisurely manner to determine the forces which were acting on the balances, by noting the position of each of the several small weights.

This all seems so simple and yet so remarkable to the stranger that he fails to notice how long and patiently the operator has to wait before the moving weights become steady, and of course it would hardly occur to the bystander that the elaborate equipment before him may require frequent and careful attention to keep it in such perfect running order. Nevertheless, this is all perfectly true, and a brief explanation of the way in which these automatic balances operate is enough to suggest that a good supply of patience and persistence is required of any one who wishes to get satisfactory results from them.

A diagram of a common method of connecting these balances to a source of electrical energy is shown in Fig. 1. Since the real aim is to make the small electric motor, which is attached to the balance beam, drive the movable weight to the left when the beam tips clockwise and to the right when the beam tips counterclockwise, it is necessary to provide a means of automatically reversing the motor whenever the beam moves slightly one way or another. The control contacts at the right-hand end of the beam serve this purpose, since the current in the armature flows in one direction when the beam touches the upper contact and in the opposite direction when the beam touches the lower contact. The result is that, as long as the beam is balanced in the mid-position between the two control contacts, no current flows in the armature of the small motor, but the instant the beam becomes unbalanced in either direction, one of the control contacts closes the circuit through the armature, and since the magnetic field is present at all times, the motor starts to rotate and moves the traveling weight in the proper direction to balance the beam once more.

This much of the action of this weighing device is exactly as is desired, but unfortunately the traveling weight does not stop the very instant that it reaches the new position for balance, because its inertia carries it a little too far, so that the beam has

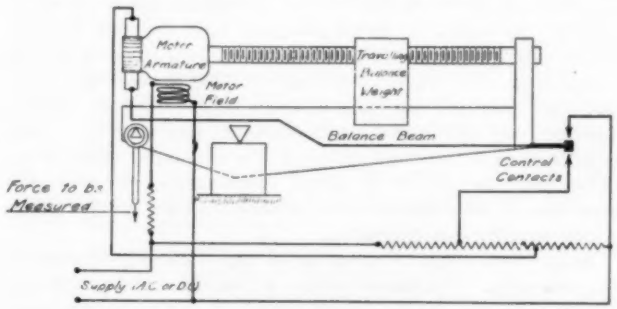


Fig. 1 Motor-Driven Balance

to swing slowly over to the other contact to reverse the motor and bring the weight back toward the proper position. If the traveling weight would always stop at the precise spot for perfect equilibrium, after the second or even the third reversal of the motor, the situation probably would not be particularly objectionable, but it must be remembered that the air forces on the wind-tunnel models are never absolutely constant, so that the position for equilibrium is continually changing very slightly, with the distressing result that if the traveling weight ever does reach a really steady condition, the chance of its remaining still for any appreciable time is extremely remote. To make matters worse, most of the testing work which is done with wind tunnels requires the use of three or more of these balances, and to be sure of accurate results all of them must be steady at the same instant,

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

so that the person operating the balances must wait patiently, with his finger constantly in readiness to push the button, if he hopes to catch a single instant during which all the traveling weights are motionless.

There is one very simple way of remedying this difficulty, but it seriously impairs the sensitivity of the balance. However, if great accuracy is not necessary, the control contacts may be separated so as to allow an appreciable movement of the beam, with the result that if the beam is slightly stable, small changes of force do not move it far enough to touch the control contacts. This has another great advantage because the contacts touch less frequently, so that there is less tendency for them to become burned and pitted when the connection is broken.

The importance of this last remark will be apparent if the action of the contacts is thoroughly understood. This action is best illustrated by imagining the beam unbalanced and one contact conducting the full armature current to the motor, which is driving the little weight toward the position which would balance the beam. As the weight approaches the proper position, the force between the contacts becomes less and less, until at the instant that the beam begins to swing there is no force pressing the two contacts together. Since the beam has a very large inertia compared with the very small unbalancing force, it moves very slowly and gives ample time for a very short electric arc to do some slight damage to the contact surface, even though the current is very small, probably less than a quarter of an ampere. Although the damage done by this one tiny arc may not be enough to harm the action of the balance, it is apparent that it will be a rapidly accumulating action, because the next time contact is made the sparking may start to occur slightly before the beam begins to drop. This is because the force between the contacts is not sufficient to press out impurities or roughness, and so the damage done this time will be much greater, and each succeeding time still greater, until finally the motor will operate sluggishly or not at all, because only a poor connection is made. It will then be necessary to clean the contacts carefully by hand, and this is not only a nuisance, but the frequent handling of the balance is sure to do some damage to the hardened knife edges which support the beam. It seems odd that these contacts cause so much more trouble than those on spark-ignition systems, but the reason is apparent when it is remembered that the ordinary make-and-break contacts are quite different in that they are pressed together very tightly and are drawn apart very quickly. These conditions tend to reduce sparking to a minimum, while on the other hand the extremely gentle touching and the very slow separating of the contacts on the balance beam are the very things which magnify this difficulty.

So far there seems to be no simple way of entirely eliminating this sparking. Condensers placed across the contacts will help the situation, but cannot entirely remedy it. Of course it is possible to reduce the current handled by the contacts, if the motor is reversed by controlling the field current instead of the armature current, but since the field has a higher inductance, sparking may be just as serious as in the case of higher current and less inductance. Moreover, it is very desirable to maintain at all times a very strong magnetic field to start the motor quickly and to aid in stopping it by means of the resulting magnetic losses.

Apparently no motor-driven balance can have all the desired qualities at one time. If extreme sensitivity is desired, the contacts must be close together, and the result is that more attention will have to be given to the contacts and more time will be required for the balance beam to obtain equilibrium. Assuming, for the moment, that the contacts may be kept clean and smooth, a reduction in the time to reach equilibrium without change in sensitivity might be obtained by decreasing the inertia of the moving parts or by increasing the accelerating and decelerating forces on the motor, and obviously neither of these alternatives allows much room for improvement. However, both the sensitivity and the time to reach equilibrium depend largely upon the ease and rapidity with which the circuit through the control contacts can be opened and closed, so that even the slightest sparking at these contacts is a serious disadvantage.

In conclusion, it may be said that aside from faults arising from mechanical design, such as lost motion in moving parts or improper damping devices, practically all the difficulties which occur in the ordinary motor-driven balance may be traced either to the inertia of the moving parts or to sparking at the control contacts, or to both.

2 Vacuum-Tube Control for Motor-Driven Balances

Granting that the two inherent faults of the motor-driven balance are the inertia of the moving parts and the sparking at the control contacts, it is apparent that, once a balance is constructed, the only way to improve its operating qualities without making any radical changes in its mechanical design is to reduce the contact difficulties to an absolute minimum.

As a possible means to this end an attempt was made to operate the motor-driven balance through a system of sensitive magnetic relays, so that the contacts would have to control

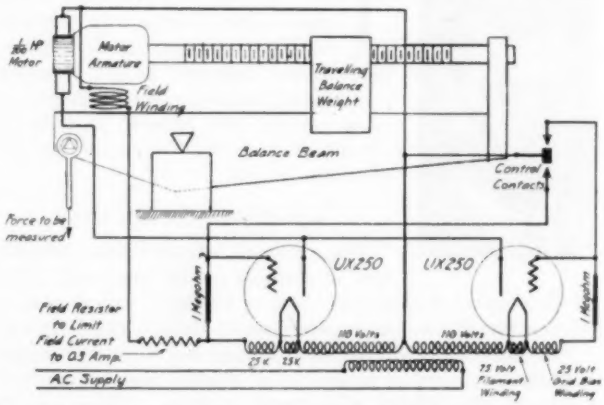


Fig. 2 Vacuum-Tube Control for Motor-Driven Balance, Alternating-Current Operation

only a very small current. However, a little experimenting showed that the brief interval of time lost in closing and opening the relays caused the weights to overrun the balance position before the current in the motor circuit was reversed, and the result was that the motor-driven weights had a very much greater tendency to oscillate back and forth about the position of balance. No attempt was made to remedy this situation by using quicker-acting magnetic relays, because the evident desirability of having absolutely instantaneous control over the current passing through the motor suggested the possibility of using the three-element vacuum-tube relay.

It is well known that the plate current of a given vacuum tube may be varied from zero to the maximum safe value for the tube by merely changing the voltage impressed on the grid, and, moreover, there is no measurable time lag between the change in grid voltage and the change in plate current. In this respect the vacuum tube is admirably suited to the needs of the electric balance, but it might be supposed that it would be impossible to operate the motors by means of the small plate currents obtainable from vacuum tubes using reasonably low plate voltages, preferably between 100 and 200 volts. However, the UX-250 vacuum tube will deliver a current of 0.25 amp. when a potential of about 95 volts is impressed on its grid and plate together, and it was found that this was ample to operate the

small $^1/_{200}$-hp. 110-volt motors used on the balances built at the University of Washington.

A good many different circuits were tried out while attempting to determine the simplest and most satisfactory method of connecting two UX-250 vacuum tubes so that they would control the motor on one of the balances. Figs. 2, 3, and 4 show three of these circuits which have been found to be satisfactory.

Fig. 2 shows what is probably the simplest way of connecting the vacuum tubes to the usual motor-driven balance. Since alternating current is used, any desired voltages may be obtained by means of a single transformer with a tapped secondary winding, as shown, or by the use of several separate transformers. To prolong the life of the vacuum tubes it is best to use the lowest plate voltage that will pass sufficient current to operate the motors, and it was found that 110 volts was sufficient for the particular balance used in the preliminary experiments. Therefore, the transformer shown in Fig. 2 has a 220-volt winding, mid-tapped to give two 110-volt windings, one for the plate of each tube; and at the two ends of this winding are the 7.5-volt filament supply windings which must be able to carry a current of 1.25 amp. The extra 25-volt windings are to furnish the necessary grid bias to cut off the plate current when the beam is balanced, and could readily be replaced with two $22^1/_2$-volt dry batteries, since very little current flows in the grid circuits.

When the balance beam dips down, a connection is made between the grid of the left-hand tube and the motor armature, which is at the same potential as the plate of the same tube. This means that both grid and plate receive the same alternating voltage, and since current will flow only when the plate is positive with respect to the filament, the motor armature receives a pulse of direct current during one-half of each cycle of the

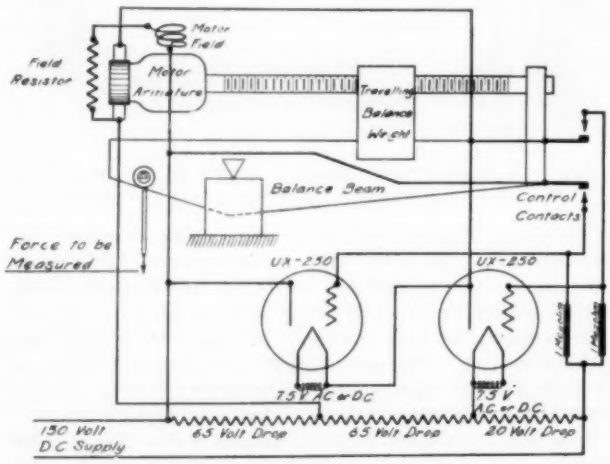


FIG. 3 VACUUM-TUBE CONTROL FOR MOTOR-DRIVEN BALANCE, DIRECT-CURRENT OPERATION

alternating voltage. Moreover, the same thing is true if the beam tips up, but in this case the right-hand tube will supply pulses of direct current which will be in the same direction through the armature, but which will occur in the opposite half-cycles of the alternating voltage. The result is that the motor develops a pulsating torque, the direction of which depends upon the polarity which the alternating field maintains during the particular half-cycles that the armature receives its pulses of direct current.

If this scheme is studied carefully, it will be apparent that the field current must be in phase with the armature current in order that the motor may develop the maximum possible torque. This was verified while trying out this circuit, and it was found that it is quite important to have a resistance of 400 ohms or more in series with the field winding.

No appreciable vibration was noticed which could be attributed to the alternating magnetic field or the pulsating torque of the motor, but since it is quite possible that some parts of the model suspension system in a wind tunnel might resonate with the slight vibrations that must exist in any device using alternating current, the direct current circuits shown in Figs. 3 and 4 were devised.

The operation of these two circuits is easily understood. Briefly stated, the vacuum-tube relays in Fig. 3 perform the

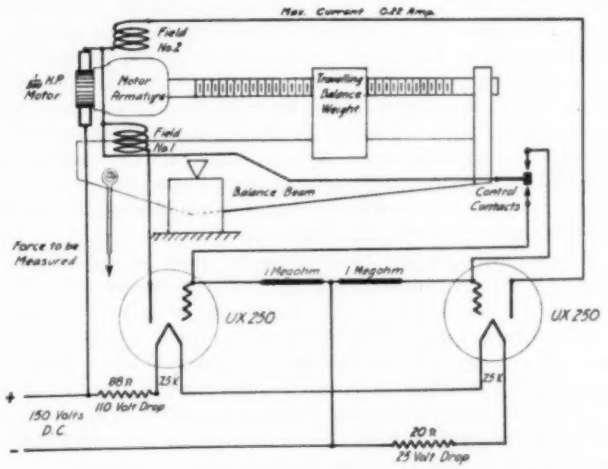


FIG. 4 VACUUM-TUBE CONTROL FOR MOTOR-DRIVEN BALANCE, DIRECT-CURRENT OPERATION

same duties as the control contacts in Fig. 1, so that the armature circuit is opened and closed by relays which operate instantaneously and have no troublesome contacts. The current which must be handled by the contact points on the balance is less than 15 milli-amp., and therefore these points may be made of very fine platinum wire striking against a tiny piece of platinum foil. Fine contacts, slightly pointed, will give better results than large flat contacts, because higher unit pressures may be obtained, thus insuring excellent connection even though the beam swings very gently one way or another. Moreover, positive and instantaneous control over the motors is practically insured by the fact that the sensitivity of the grid circuit of a vacuum tube is so great that contact resistances even in excess of 1000 ohms would not seriously impair the operation of the vacuum tubes as relays.

Turning now to Fig. 4, which is a diagram of the balance now in use at the University of Washington, attention is called to the extreme simplicity of connections. All the equipment which is required consists of the two UX-250 vacuum tubes, two 1-megohm grid-lead resistors, and two resistors capable of carrying 1.25 amp., one of which has a resistance of 88 ohms and the other a resistance of 20 ohms. Light globes may be used for the latter two resistors if desired. Two 75-watt mazdas in parallel will approximate the 88-ohm resistance, and one 50-watt 28-to-32-volt lamp will serve for the 20-ohm resistor.

The only alteration required to convert the ordinary electric balance to this new circuit is to replace the usual motor with one equipped with two field windings. In case it is not possible to obtain such a motor, an extra winding can often be wound on a form and slipped over the top of the original field coils of the motor. This was done for the balance now in operation at the University of Washington.

The reversing of the motor is accomplished by simply allowing

the current to flow through the armature and one or the other field windings, according to which vacuum tube receives a positive grid potential from the control contacts. The direction of the current is always the same through both armature and field windings, but one field is connected to drive the motor in one direction and the other field is connected to drive it in the opposite direction.

One difficulty which might become objectionable is the fact that, because the filament of the two vacuum tubes are connected in series for the purpose of decreasing the current taken from the line, the filament of No. 1 in Fig. 4 will be at a higher potential than that of No. 2. The result is that the potential, and therefore the current, between plate and filament is higher in tube No. 2 than in tube No. 1, so that the motor will tend to run a little faster in one direction than in the other. However, if the characteristics of the two vacuum tubes differ sufficiently, the plate currents can be practically equalized by simply putting the more active tube where it receives the least plate voltage. In any case it is a simple matter to equalize the current flowing through each tube by placing a few hundred ohms resistance between the upper control contact and the grid of tube No. 2.

Any ordinary electric balance can be made almost entirely free from contact troubles by using one of the vacuum-tube control circuits described here. Of course if the motor requires more than 0.25 amp. for satisfactory operation, it will be necessary to use two or more UX-250 vacuum tubes in parallel in place of each one shown in Figs. 2, 3, and 4. Moreover, it may be found necessary to alter the control contacts in some such manner as has been previously suggested, because some of the metals which are commonly used for contact points tend to coat over with a thin layer of a poor conducting oxide, and while this coating may not seem to have any harmful effect when the contacts handle the full armature current of the motor, it may be quite troublesome when the current is too small to cause any appreciable heating or sparking.

The use of the vacuum tubes as described here should cause

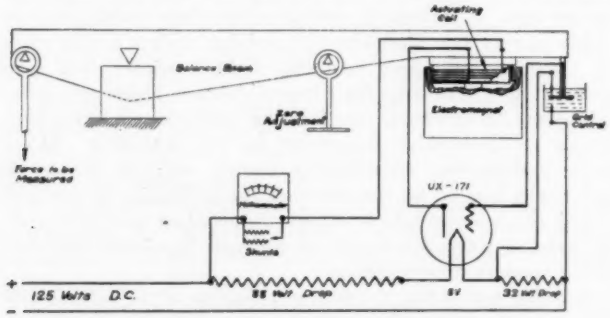


FIG. 5 ELECTROMAGNETIC BALANCE

a decided increase in the sensitivity of any motor-driven electric balance, because the motor will respond instantly to even the most gentle pressure between the contacts. Furthermore, since the contacts can be placed very close together without any danger of an arc forming between them, the very slightest movement of the beam will actuate the motor. Of course this will make the balance more active, and therefore more time may be required for the traveling weight to find a steady position, but it has been found that it is possible to improve this condition by shifting the center of gravity of the beam slightly below the fulcrum, so that the beam will be in stable equilibrium. The sensitivity is impaired very little by this procedure when the contacts are very close together, and it is possible to obtain remarkably quick readings, when great accuracy is not necessary, by simply separating the control contacts any desired amount.

3 THE ELECTROMAGNETIC BALANCE

The new electromagnetic balance is still in the experimental stage, but it should attract a good deal of interest because it is quite different from any weighing device that is known to have been used before. Comparing it with the previously described motor-driven balances, the electromagnetic balance is less costly, has no moving parts on the beam, is practically instantaneous in its response to sudden changes in force, can be quickly adjusted to almost any desired degree of sensitivity, and is readily adapted to recording devices to do away with the point-by-point method of obtaining curves. These are all distinct advantages, and so far no serious disadvantages have been foreseen.

The experimental balance of this type which is now in operation at the University of Washington is shown in Fig. 5. The beam is balanced against the unknown force by a magnetic pull on the "actuating coil" which is attached to the right-hand end

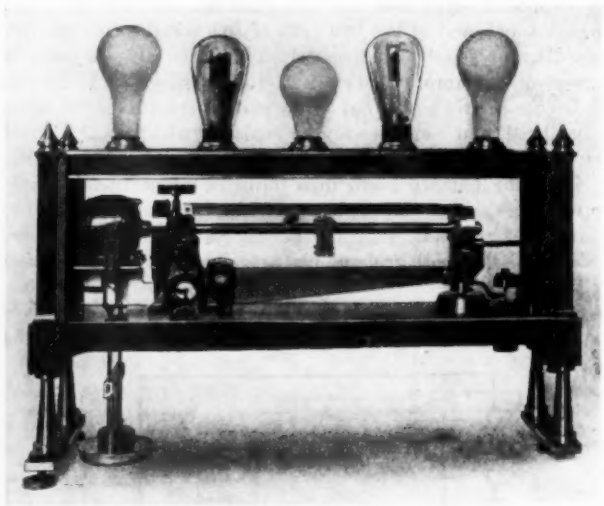


FIG. 6 VACUUM-TUBE-CONTROLLED MOTOR-DRIVEN BALANCE
(This was designed and built at the University of Washington.)

of the beam. This coil consists of a few hundred turns of very fine wire wound on a very thin cylindrical tube. The overall thickness of the wire and tube is 0.06 in., and this whole ring-shaped winding is placed in the circular air gap of a powerful electromagnet, allowing sufficient clearance to prevent any possible rubbing. The electromagnet is so designed that most of the iron will be saturated, and therefore any small fluctuations in its exciting current will scarcely affect the magnetic flux.

This part of the electromagnetic balance is very similar to the electrodynamic loud speaker which is so commonly used in radios and electric phonographs. Moreover, the magnetic pull on the actuating coil of the balance is obtained in exactly the same manner that the motive force is obtained to drive the vibrating cone of the loud speaker. The important thing to remember in this connection is that the force is proportional to the current flowing through the little coil in the air gap, and also to the strength of the magnetic field passing through this little coil. In the case of the electromagnetic balance special care is taken to maintain a practically constant magnetic field, so that the force which is transmitted to the balance beam should be directly proportional to the current in the actuating coil. It follows that the current required to balance the beam is a direct measure of the unknown force, and if this current can be automatically adjusted to keep the beam in perfect balance, we will have an ideal device for measuring forces.

This has been done in the experimental balance shown in Fig. 5 by the use of a UX-171 vacuum tube and a special liquid potentiometer. The moving electrode of the potentiometer is attached to the balance beam, and its purpose is to decrease the bias voltage on the grid of the vacuum tube in case the beam tips up, and to increase this voltage in case the beam tips down. The plate current of the vacuum tube passes through the actuating coil of the balance, thus producing a downward electromagnetic force on the beam, the strength of which depends upon the value of the plate current. The result is that if a sufficient load is applied to the left-hand end of the beam the moving electrode of the liquid potentiometer will rise a little, and in so doing it will reduce the negative bias on the grid of the vacuum tube, and therefore increase the plate current and create a downward electromagnetic force which will balance the applied load. A milliammeter placed in the plate circuit measures the current flowing through the actuating coil, and it is a simple matter to furnish the proper shunts to make this meter read directly in grams or in pounds.

One of the remarkable features of this type of balance is the rapidity with which it responds to sudden changes in force. Even when the load is suddenly increased from zero to full-scale deflection, only a fraction of a second is required for the pointer of the milliammeter to come to rest at its new position. Moreover, the pointer should be absolutely steady as long as there is no change in the applied load, because the beam is in a truly stable condition and any tendency to swing in either direction is instantly opposed by a change in electromagnetic force.

Another notable feature is the ease with which the sensitivity of the device can be varied. If the maximum plate current available from the UX-171 tube is 50 milliamp., a meter having a range of not over 0-5 milliamp. should be used, so that the sensitivity may be adjusted by using different shunts. For example, if the balance could measure a force of 500 grams with 50 milliamp. flowing in the actuating coil, the 0-5 milliammeter would measure this with a suitable shunt across it, and in this case if the meter had 100 divisions on its scale, the finest division would represent 5 grams. If greater accuracy should be desired, the shunt could then be removed, and full-scale deflection would then represent 50 grams instead of 500 grams, and the finest division would represent 0.5 gram instead of 5 grams. Even greater accuracy might be had if a more sensitive meter is used, but this should not be carried too far without proper precautions, because any accidental increase in load might easily burn out the more sensitive instruments. Another alternative is to place a shunt across the actuating coil instead of the meter, and in this way no harm can come to the electrical instrument, because the 0-50 milliammeter would be used, and the current can never rise above the maximum which the vacuum tube will pass. However, it has been found that the balance begins to act sluggishly if the actuating coil is shunted by too low a resistance, so that this method of varying sensitivity is also limited.

It might be supposed that the electromagnetic balance cannot combine great accuracy with large range because the range is fixed by the maximum reading of the milliammeter. The fact is that the range can easily be extended indefinitely by adding weights to a pan hung from the beam on the right-hand side of the fulcrum, as in Fig. 5. If the distance from this pan to the fulcrum is made exactly equal to the distance from the fulcrum to the point of suspension of the unknown load, then in case the milliammeter should indicate a load of 500 grams, a standard 500-gram weight may be added to the pan to bring the needle of the meter back to zero. This addition of a known weight to the pan will also afford a quick and convenient check on the calibration of the meter.

In spite of the fact that the experimental electromagnetic balance has been operated successfully for more than a month, there is much room for improvement in the liquid potentiometer for controlling the grid bias of the vacuum tube. So far no liquid has been found that will continually maintain a constant potential gradient of 50 volts across a distance of less than 0.01 in. Glycerin has been used with the greatest success, and several designs for the moving electrode have been experimented with in an attempt to reduce the irregularity in potential gradient which is apparently caused by small gas bubbles. However, there seems to be little hope of obtaining an entirely satisfactory liquid potentiometer for the circuit shown in Fig. 5, because such a large voltage must be impressed across a very thin layer of liquid. Fortunately there is a simple remedy for this difficulty, the success of which is almost certain, due to the fact that the bubbling in any liquid becomes less when the impressed voltage is reduced. It is therefore proposed to decrease the voltage across the potentiometer liquid from 50 volts to 5 volts, or less if necessary, and insert a vacuum-tube amplifier between the moving electrode and the grid of the UX-171. In this way the very small changes in voltage of the electrode attached to the beam can be amplified to the value necessary to control the current in the actuating coil of the balance.

This idea has not been tried out yet because the simpler circuit in Fig. 5 was found to be good enough for the preliminary experiments, but there is very little reason to doubt that absolute satisfaction can be obtained if the potential drop through the liquid can be kept very low. Moreover, if necessary it would probably be possible to do away with the liquid potentiometer entirely by devising a means of controlling the grid voltage of the UX-171 tube by allowing the beam to change the capacity of a small condenser. This would require the use of a more complicated circuit using several vacuum tubes, but in case liquid potentiometers should eventually prove to be troublesome in any way, which is very unlikely, this additional complication would hardly be objectionable.

The experimental electromagnetic balance described in this paper has already proved the practicability of measuring forces by determining the current required to produce a balancing electromagnetic force. In spite of the fact that when the device is used for a considerable period of time it becomes slightly unsteady, due to the formation of gas bubbles in the liquid potentiometer, the force measurements were in all cases perfectly reliable, and if the equipment was allowed to stand idle for a brief period of time, the gas bubbles would pass off and the balance would give perfectly steady readings once more. The methods suggested for overcoming the difficulties due to the gassing of the potentiometer liquid are practically certain to give absolute satisfaction at all times, and therefore it is believed that an electromagnetic balance can be built which will have several very decided advantages over any of the wind-tunnel balances in common use today. A brief summary of some of the more important of these advantages follows:

1 There are no moving parts on the device except the balancing beam, and this part moves only very slightly, in accordance with the applied load.

2 The initial cost of one of these balances should be far less than that of the average motor-driven balance, because there are fewer parts to make, and the size and weight of these parts are relatively unimportant because the balance is finally calibrated by choosing a suitable shunt for the milliammeter.

3 A wide range of sensitivity can be had from a given electromagnetic balance by simply using different shunts across the milliammeter. This is of special importance in measuring the small changes of the drag of a model due to slight modifications of its form.

4 The electromagnetic type of instrument gives an almost instantaneous indication of sudden changes in applied load, so

that quite rapid fluctuations in force can be followed by observing the pointer of the milliammeter.

5 The calibration of the device is practically as permanent as the calibration of the electrical indicating device and can be checked and readjusted in a moment's time.

6 It should be possible to make the electromagnetic balance operate graphical recording electrical instruments, so that the variation of the lift and drag forces on a wind-tunnel model could be automatically recorded as the angle of attack is changed. If a similar device could be used to record the air velocity at the same time, all the necessary engineering data could be obtained from the resulting graphs, so that the human element would be entirely eliminated from the wind-tunnel tests, and all the data could be obtained in a few minutes instead of several hours.

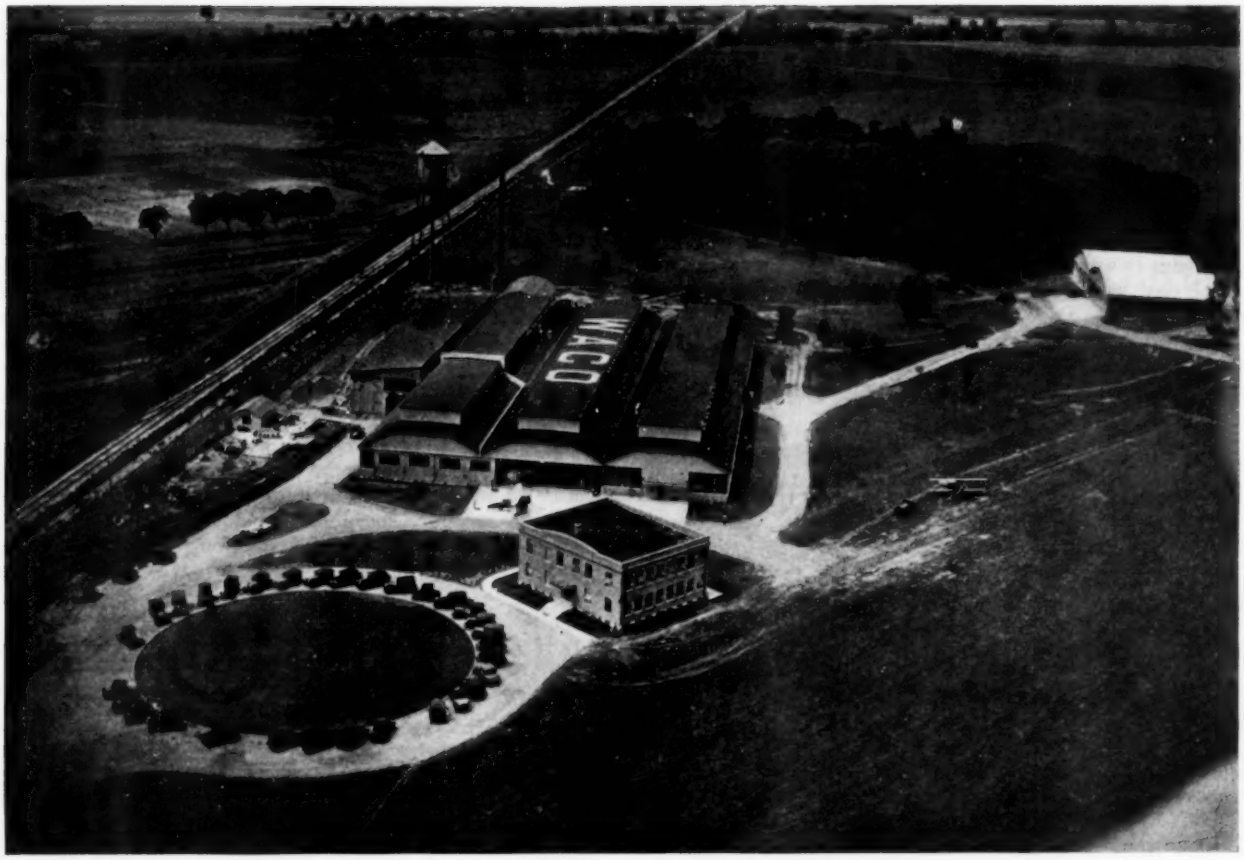


FIG. 1 THE WACO AIRCRAFT COMPANY'S BUILDINGS AND FIELD, TROY, OHIO

Design and Manufacture of Airplanes From the Point of View of Low Production Costs

BY RUSSELL F. HARDY,[1] TROY, OHIO

EXPERIENCE gained in the automotive industry has saved the aircraft industry many millions of dollars. The most valuable information obtained has been along the lines of production. A large quantity of any product built with up-to-date methods usually results in low manufacturing costs. Consequently, to build airplanes having a low manufacturing cost, one of the first things to consider is: Will they be produced in sufficient quantity so that the cost per unit will be appreciably low?

This question, of course, involves many factors including the facilities for quantity production as well as the capability of the entire sales organization in disposing of the product. On the other hand, the sales department will not be able to sell quantities of a product unless it takes well with the general public.

Assuming that the product will be accepted by the general public, various items of importance in the design and manufacture of an airplane should be considered.

The first thing to be decided is the type of aircraft which is to be built—whether it will be military, cargo carrying, passenger carrying, or for the private individual. Assume that it is desired to produce a great number of a certain type. At the present time it should be the objective to sell the product to the private individual who perhaps has never piloted an airplane and has had only an occasional ride. This will necessitate that the airplane be at least of the two-place type or more. The two-place type will be perfectly satisfactory during the time in which the purchaser is receiving his instructions. At this point opinions will vary. What type of airplane will best suit the purpose of training the purchaser as well as giving satisfactory service after he has obtained his license? Surely he will not object to having room for one passenger, and very likely he would be more desirous of carrying two if the initial cost of the airplane is only slightly more and the performance difference very small.

A few engineering investigations may next be made to ascer-

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[1] Chief Engineer, The WACO Aircraft Co.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

tain the difference in weight of the airplane structure due to the increased pay load of one additional passenger and baggage and also to determine the difference in performance. If these differences are small, it is very likely that the airplane will be of, at least, the three-place type, providing the engine to be used is capable of carrying the load with satisfactory performance.

Having decided upon the number of persons which the airplane is to carry, one might next consider whether it is to be of the open or closed type. Again there will be many differences of opinion, and very likely most of them may be justified. It would appear that most of the training airplanes in the past have been of the open-cockpit type. However, at the present time there are a few small planes used in training which are of the closed type. Both have their advantages, to



FIG. 2 A FLEET OF WACO AIRPLANES DELIVERED TO THE AVIATION BUREAU OF CANTON, CHINA

be sure. It is almost certain that the open type can be built at a lower cost, a fact which is also true in the automotive industry.

There have been many articles written on the subject of monoplanes versus biplanes, and it will be the duty of the designer to decide which type will most satisfactorily meet the requirements. One can now begin to picture what the airplane is to look like when completed. Perhaps it will be a two-place closed-type monoplane or a three-place open biplane. After the general type has been decided upon, line drawings of the fuselage, wings, and empennage can be started. The type and weight of the engine to be used must be known at this stage so that a balance diagram may be obtained. Since the balance diagram should be constructed before the airplane is built, the various weights and their locations will necessarily be approximated. This would seem to be a rather difficult task, but with a little experience a designer can estimate the weight empty of the airplane very accurately. There are many other items which must be decided upon at this time, some of which are airfoil section, area and type of wings, and location and capacity of fuel and oil tanks.

It is very important at this stage of the design to make certain that the general proportions and arrangement of the wings, landing gear, fuselage, and empennage present a satisfactory appearance. This is best accomplished by constructing a full-size mock-up of the airplane which incorporates the features already decided upon. While a few airplanes are said to be built entirely from drawings, the author has usually found that much time can be saved through the aid of an inexpensive mock-up. A mock-up should at least include that portion of the fuselage in front of and including the space occupied by the rearmost occupant. Stub wings should be placed in the proper location so that visibility and ease of entrance to the airplane can be determined. The arrangement of the seats is of utmost importance so that plenty of leg room and comfort for the occupants will be assured. In some cases the mock-up might include the structure back to and including the empennage. The turtle deck can be covered with thick paper and in this manner the general lines of the airplane may be obtained. The structural parts of the mock-up can be made of wood and these wood members will be in the same location as those which are later to be made of steel tubing.

After the mock-up has proved to be satisfactory with regard to general dimensions and appearances, line diagrams of the fuselage and landing gear may be drawn and from these the preliminary analysis may be made. Stress diagrams can be made with unit load conditions and from these diagrams it may be determined whether any of the members take excessive loads. If it is found that they do, then a different arrangement of the structural members may be necessary.

Before any final analysis is made, however, the maximum horsepower of the engines to be used must be known, in order that the proper load factors may be obtained. Since many manufacturers of aircraft rely on outside sources for their engine supply, it is usually hard to decide what horsepower engines should be used in a particular design. From a standpoint of flexibility it is desirable to be able to equip the airplane with engines having a varying range of power. However, the higher powered engines determine the maximum load factor to be used, and as a consequence the structural weight will be greater than would be necessary for a lesser powered engine. Assuming that the entire structure will be identical for different engines with the exception of the engine mount, the airplane having the lesser power will be penalized by the unnecessary structural weight.

It is usually desirable to build an experimental airplane be-

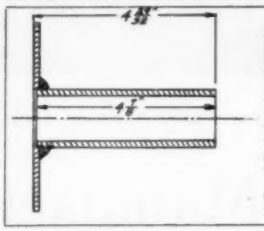
THE WACO AIRCRAFT CO.

ENGINEERING CHANGE RECORD

Part No. 2001 Date 3/14/30

Dwg. No. Letter D

Part Name Tube, Axle Retainer Models Effected R, W, & F



Due to the change in length of the axle #1878, it will be necessary to change the length of the axle retainer tube from 4 7/8" to 4 29/32". This will permit the flange to be welded flush with the end of the retainer tube.

The flange may then be machined even with the end of the retainer tube which will make a satisfactory backing for the thrust washer.

This change supersedes the change dated 11/21/29.

Submitted By John Doe Checked By George Jones Approved By Richard Roe

Effective At once.

FIG. 3 DESIGN CHANGES ARE RECORDED ON A CARD OF THIS TYPE

fore too much work is done on the analysis, in order that the test pilot can determine whether the balance, performance, and stability are satisfactory. Approximate methods may be used to determine sizes of the primary members such as wing spars, longerons, empennage members, and others. It is understood that the wing section and area have already been decided upon. The experimental model should be constructed so that the dihedral, incidence, and perhaps the stagger may be varied in order that the characteristics previously mentioned may be studied. It is very seldom that the approved airplane, later to be built in quantities, is the same as the one upon which the first tests are run.

After the tests have been run, it is advisable to complete the stress analysis of the various primary and secondary structural units. The primary units comprise the main structural members of the wing cellule, fuselage, and landing gear. The secondary units consist of fittings, lugs, clips, and other small members.

After the sizes of the fuselage and landing gear members have been determined from the stress analysis, complete units should be built up and drop tests conducted in order to determine whether they will later pass the Department of Commerce requirements. The ailerons and empennage surfaces

should also be static-tested so that there will be no delay due to insufficient strength or unsatisfactory action at a later date when these units are tested in the presence of the Department of Commerce inspector.

Even though the fittings, lugs, clips, and other secondary units have been analyzed, it is very advisable when possible to perform tests on them to determine their actual strength. These parts may have shown ample margins in the stress analysis, but much can be learned about them during destruction tests. Aileron control mechanisms may be connected to a slow-speed motor through a series of reduction gears, and operated over long periods of time under an imposed load in order to be assured of the proper functioning of all parts.

After the completion of satisfactory load tests, the stress analysis and the necessary drawings should be submitted to the Department of Commerce for checking and approval. When the approval is obtained, the airplane may be subjected to the various flight tests, and if these are satisfactory an approved-type certificate will be issued to the manufacturer.

Before the airplane can be produced satisfactorily in quantities it is necessary to have detail drawings of the various parts. The experimental drawings are checked very carefully and turned over to the production drafting department where production part numbers are assigned and detail drawings made. The part names and numbers are recorded alphabetically and numerically so that information may be quickly given to various departments requesting it. Specifications in accordance with these names and numbers are compiled by the planning department, and production orders may then be issued to the various departments.

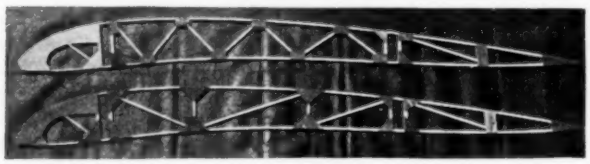


FIG. 4 TWO TYPES OF RIBS HAVING THE SAME WEIGHT
(The upper rib is less costly to build since eight of the web members have the same length. It also has much greater strength due to the smaller span between members.)



FIG. 5 ABOVE, ALUMINUM-ALLOY CASTINGS OF STICK SOCKET AND CONTROL-SHAFT BEARING—BELOW, AILERON HINGE MADE FROM DROP FORGING

Any changes in design or construction must be recorded on an "engineering change card" and approved by one of authority in the engineering department. A permanent record of the changes should be kept in order that service orders may be filled correctly at a later date.

Throughout the entire design, standardization should be kept vividly in mind. If the airplane is of the biplane type, the upper and lower wings should be as nearly identical as is possible and practical. The spar fittings should be made up from blanks and plates which may be used interchangeably throughout the design. Compression ribs should be spaced so that different length internal drag wires may be reduced to a minimum. Ribs should be identically the same as far as possible. It may also be desirable to use metal tubes, preferably duralumin, for compression struts throughout the wing, and in this manner eliminate the special compression ribs. If several experimental wings having different airfoil sections are to be built up, it is cheaper to construct them with a plywood web instead of the truss type, for with this method of construction the airfoil section may be cut out on a bandsaw and the cap strips bent to conform with the contour of the rib. If the truss-type rib is to be employed, a jig is necessary to hold the various members in position during gluing and nailing operations.

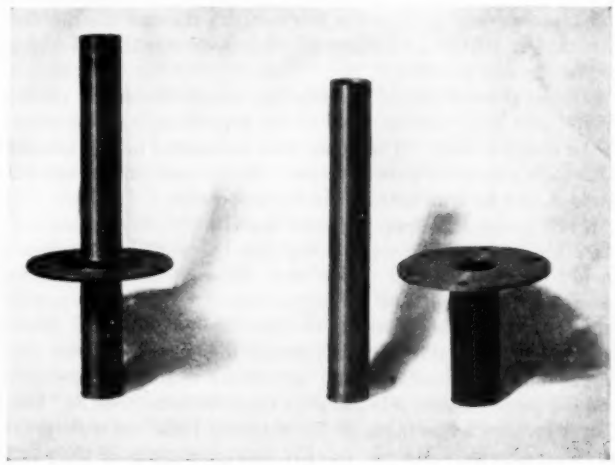


FIG. 6 TWO TYPES OF CONSTRUCTION FOR BRAKE-TORQUE FLANGE AND AXLE
(At left, brake-torque flange welded to the axle. The entire unit is then heat-treated and finished to size. As an improvement an axle which is heat-treated and machined to size, shown in the center, is later pressed into the retainer and flange unit shown at extreme right. There is no welding on this axle, which results in greater strength and cheaper production costs.)

If the truss-type rib is to be adopted as standard, it should be designed so that its production costs will be reduced to a minimum. All web members should have the same length as far as possible so as to reduce costs. Also, a standard gusset should be used in as many places as possible. Various types of ribs should be built up, and a decision as to which should be used can be made after characteristics of strength, weight, and costs have been determined. The internal brace-wire fittings can usually be standardized and this will reduce wing costs materially.

Parts in the nature of aileron hinges, of which there are usually a great number per airplane, can be produced very economically from drop forgings. The die cost is the main item to be considered, but when it is distributed over thousands of parts, an appreciable saving in production costs will be realized.

Many parts, such as stick sockets, can best be produced in the form of an aluminum-alloy casting. The unit is light in weight and its strength can be determined by severe static and fatigue tests imposed upon the part in a manner similar to that which it receives in service on the airplane.

Parts, such as spar bracket, side plates, and clips, can either be stamped out or nibbled. It is always well to have a very accurate estimate of the quantity to be produced so that cost figures may be obtained which will make it possible to determine whether the part should be produced on a press or on a nibbler. The nibbled part has to be finished by filing, while the stamped part may be used as it comes from the press. A change in design is more easily made and is less costly in the case of a nibbled part than in that of a stamping which necessitates the reworking of the die and punch.

Structural members of the fuselage frame should be cut to length, and have their end cut to the proper angle and contour in the machine shop. They may then be stocked in bins labeled with their respective part numbers. Hand work on the various parts should be eliminated as much as possible.

If it is decided to equip the same model with different engines, then the question arises as to whether the complete fuselages should be built up in units with different integral engine mounts or in a single unit up to the engine mount which at this point is equipped with fittings to take different mounts. From a standpoint of production it is desirable to have the demountable engine mount, as fuselages may be built up in advance of orders and the proper mount selected after the order is received. Then, too, there are advantages to be obtained from the standpoint of the purchaser. He can quickly change engines of the same type in the event of a necessary overhaul or can install an entirely different type of engine by purchasing the proper replacement mount.

The merits of removable mounts are opposed by certain undesirable features, one of which is the tendency of the mount to become loosened at the points where it connects the fuselage unless a great amount of care is taken in the design of the fittings. Then, too, the cost of building a fuselage and separate mount is greater than if the two were built in one unit. Parts fabricated by welding should be held in rigid jigs so that variations in the finished product may be reduced to a minimum.

Welding jigs should be designed so that the parts may be easily inserted, welded, and removed. In the welding of large units, such as landing-gear vee struts, it is often advisable to check them, after they have been completed, in a test jig to determine whether they are within the tolerances. It is also advisable to weld up a master fuselage which is exactly correct with regard to tube sizes and dimensional distances. This fuselage is used as a check on the welding jigs which may be found in error after repeated use. If at any time assemblies are made with difficulty, the part in question may be quickly checked against the master fuselage.

The inspection department plays a very important part in keeping production costs down to a minimum. All material should be inspected before it is fabricated into the various parts. If the part happens to be a welded assembly, it must later be inspected from the standpoint of a weld, and in addition must be checked against the blueprint specifications in order that it will fit properly when assembled on the airplane. The inspection department should be under the supervision of the engineering department, for in this manner any questions which may arise may be settled directly and promptly.

No matter how perfectly an airplane may be designed from the standpoint of standardization of parts, the production costs will be high unless proper routing of stock and materials is employed. The raw stock should pass through the receiving department to stockrooms which should be adjacent to the various departments in which it is to be made up. The stock is then requisitioned from the stockroom and put through the various working processes. If the parts are completed in the machine shop they may then pass to the stockroom in which completed parts are kept. If the part is not completed until it passes through welding operations, it may then be returned to the stockroom to be later requisitioned by the welding department. The part may then be welded, sandblasted, inspected, rustproofed, and stored in the stockroom for finished parts. The various departments and stockrooms should be arranged so that the part will travel the least possible distance.

Thus, it is seen that the essential factors which contribute to low production costs in the manufacture of aircraft are: First, an acceptable design which will lead to quantity production; second, correct procedure with regard to designing, standardizing, experimenting, and testing; and third, efficient production methods including capable personnel, satisfactory equipment, rigid inspection, and proper routing of materials.

Discussion

H. Rawdon.[2] The writer can in one or two instances take an opposite stand to the statements contained in the paper on the design and production of airplanes for low cost. Anything economically accomplished must be systematically done. The exact system to be used is not definite, like a mathematical proposition. There are, however, general principles which can be considered.

The design which would be an economical job for a production of 500 units per year would not necessarily be of the same relative production cheapness when the production had reached an annual volume of 10,000 units. Likewise, the manufacturing system will have to be modified in the two cases, though they may fundamentally be similar.

The first step in producing an airplane is the determination of the requirements of the buying public. With this information at hand it is possible to proceed with the details. The writer will elaborate on some points in accordance with his own personal opinions and observations.

It should be the duty of the sales department to analyze the requirements of the buying public, and to intelligently predict the volume of consumption of the article to be manufactured. In the case of the airplane it is very essential to have the foregoing information accurate if the design is to be intelligently developed. It is the writer's opinion and observation that the lack of such analyses has in many cases resulted in very costly experiments, both in design and production.

It has been the policy of most companies to cater to "custom built" trade. This process is evidently not economical in production. The unit cost of construction runs high for such type of work. The greatest profit is to be made with standardized products. There is also another difficulty encountered with these special airplanes, that of the legal aspect. The requirements of the United States Department of Commerce that a machine is built in exact accordance with the approved type means that the special-built airplanes will require additional work and expense in preparation and submission of data necessary to secure the commercial-type license. Requirements and fancies of the general buyer have been varied and fluctuating in the past and have been difficult problems with which to deal.

The extent that standardization of parts can be carried is also a function of the number of individual models built by the manufacturer. The writer cannot agree with the idea of identical upper and lower wings. There seem to me to be other

[2] Travel Air Co., Wichita, Kansas.

problems, aerodynamical, appearance, and even fabrication considerations, that would offset the small advantages from the manufacturing end. Would it not be of more definite advantage to have it said, the same and identical lower wing on two separate types? This presupposes a biplane design. The writer personally is not a monoplane enthusiast, while he takes it that the author favors the biplane type. Both types have their advantages and adherents.

The consideration of the mock-up is essential, and usually it does not pay, in the experimental design, to go too far in making production drawings. The mistake is often made of releasing airplanes for production and sale before they have been thoroughly tested and corrected for minor defects with details. Not only should cheapness of production be considered, but also that of servicing those articles that are in the hands of the customer. It will prove more expensive to the manufacturer who has to replace parts due to faulty design than it will to the manufacturer who has the design faults found and corrected before he has sold his article to the public. In the end it is the article that makes the most profit for a given sale price. This profit is a function of other considerations as well as the cost of manufacturing. In general it pays the manufacturer to proceed with some caution. Demand that exceeds production output is usually conducive to expensive manufacturing policies. This has been the situation the past two years.

Very often it happens that temporary dies can be made and simple fitting blanks can be stamped at a saving in lots as small as 15 airplanes. In the initial design a certain amount of flexibility should be allowed for, so that later on if it is proved that the airplane is in popular demand the details can be redesigned to adapt them to methods of quantity production. In this way it is often possible to save much expense in die construction and various tooling-up processes where the demand proved less than anticipation.

Functionally the inspection department reports to the engineering department, since the drawing is the primary source of information. However, the inspection may report to the chief engineer or the general manager, and in either case can be found examples of successful functioning. Much importance is attached to what is to be inspected and the stage of construction at which the inspection takes place. Material inspected at the proper stage usually saves much expense later on.

The assumption of acceptable design, that is, in regard to the quantity to be consumed, is usually open to much questioning, and is best proved by actual trial and experience.

Author's Closure

The author wishes to take this opportunity to thank Mr. Rawdon for his discussion. While in one or two instances he has seemingly taken an opposite stand, these differences in opinion might be attributed to a lack of clarity or completeness in writing the paper.

Mr. Rawdon states, and justly so, that "It should be the duty of the sales department to analyze the requirements of the buying public." The assumption that this had been done was intended to have been brought out in the third paragraph of the paper, which reads: "Assuming that the product will be accepted by the general public, various items of importance in the design and manufacture of an airplane should be considered."

It is not always easy to estimate, even roughly, the demands of the buying public. Last year and the present year have been excellent examples of this, but we can attribute this exception of the rule to the financial unrest throughout the country. Many production schedules if carried out as originally planned this spring would have left, and in some cases have actually left, the company with large numbers of unsold articles.

In the paper it is stated: "If the airplane is of the biplane type, the upper and lower wings should be as nearly identical as is possible and practical."

Aerodynamically, there may be some slight advantage in having different upper and lower wings, but the author does not believe that this advantage will offset that of being able to keep the majority of parts interchangeable. The appearance is usually taken care of by separating the upper wing panels with a center section of span equal to or greater than the width of the fuselage which separates the two lower wings. Standardization should not in any case be carried out to the extent that the impractical features outclass the advantages.

AER-52-30

High-Performance Gasoline Aircraft Engine

Its Problems of Fuel, Oil, and Materials of Cylinder Construction

By S. D. HERON,[1] DAYTON, OHIO

The paper discusses the available fuels and anti-knock compounds for aircraft use. The question of the anti-knock values of widely different fuels is one concerning which but little is known. The determination of these knock values was the object of the experiments conducted.



THE constant striving for increase of output from aircraft engines of given size and displacement has led to increase of compression ratio, improvement in volumetric efficiency, supercharging, and increase of engine speed. The first three factors, other things being equal, demand improvement in the anti-knock quality of the fuel used. The fourth factor tends to diminish the required quality of the fuel up to the point where the increased rate of heat flux to the cylinder and piston causes overheating. When overheating thus occurs, it is susceptible to fuel changes. Although the reduction or elimination of such overheating by means of fuel can be and has been obtained in high-performance engines, for aircraft and other purposes, by the use of alcohol and alcohol mixtures, the method is hardly practical for service aircraft engines on account of the resulting high specific fuel consumption.

That increase of compression ratio, up to the point where detonation occurs, results in increase of output has long been known, but so far as the author knows, A. H. Gibson[2] was the first to discover that such increase possessed peculiar advantages for aircraft use, inasmuch as it diminished the specific heat flow to the cylinder walls and thus reduced the problem of disposing of the waste heat. In the case of some air-cooled cylinders, Gibson in 1916–17 discovered that not merely was the specific heat flow diminished, but that apparently the total flow also was sharply reduced, as in several air-cooled cylinders the mean and maximum cylinder wall temperatures dropped as the compression ratio increased with otherwise constant conditions. Increase of compression to the point just below that at which detonation starts results in increase of power output, reduction of specific fuel consumption, and reduction of specific heat rejection. Reduction of both engine weight and fuel consumption are of primary importance in aircraft. Reduction of either specific or total heat rejection by the engine is of outstanding importance, as the amount of parasitic resistance involved in transferring to the atmosphere the heat rejected by the cylinder-cooling system is thus reduced. Supercharging, while it allows of greater increase of output with a given fuel improvement than is possible with high compression, does not improve fuel consumption—in general, rather the reverse—and it increases the heat rejection to the cylinder walls in some cases to such an extent that marked increase of fuel consumption is necessary in order to prevent failure from overheating.

The limiting factor in the use of any fuel in an aircraft engine is its tendency to detonate and thus heat the cylinder and piston unit. There is some evidence that, while under a given set of conditions two fuels may have equal tendencies to audible detonation, their effects in heating the cylinder and piston unit may show somewhat decided differences.

Until recently, the most widely used fuel in American aircraft engines has been straight-run Mid-Continent domestic aviation gasoline, meeting the Federal Specification Board requirements for distillation. In the past, when a fuel superior to this has been required, the addition of benzol has been general. Average straight-run Mid-Continent D.A.G., from the anti-knock standpoint, is an exceedingly poor fuel for modern aircraft-engine use. It is inferior to much of the low-test gasoline available for automobile purposes. Large supplies of automobile anti-knock gasoline are available which approximate in anti-knock value a mixture of 70 per cent average Mid-Continent D.A.G. with 30 per cent benzol. With increase of compression ratio, volumetric efficiency, and cylinder-wall temperature, Mid-Continent D.A.G. has ceased to be a satisfactory fuel for a majority of modern engines. A large amount of the modern equipment in use in the Air Corps will not operate on Mid-Continent D.A.G., even when throttled at ground level and not opened out fully below an altitude of 8000 ft.

Pending an investigation of the most suitable fuel for present and future equipment, the Air Corps decided to add 3 cc. of tetraethyl lead per gallon to all gasoline for use in such equipment as would not operate on the fuel procured to its Specification 2-40F, which specification, in general, results in the supply of average Mid-Continent D.A.G. The arbitrary addition of 3 cc. of tetraethyl lead to gasoline meeting Specification 2-40F in no sense represents an anti-knock value, but the resultant fuel is at least 3 cc. of lead better than it otherwise would be.

At the time the invitation to write this paper was accepted, the author hoped to be able to be much more definite on the subject of available fuels and anti-knocks for aircraft use than is now possible. The question of the anti-knock values of widely different fuels in both aircraft and fuel-test engines is one concerning which but little is known, nor is a thorough understanding likely to be gained for the next two or three years. Therefore, it is at present only possible to indulge in generalities on the subject.

SECTION 1 DETERMINATION OF ANTI-KNOCK VALUE

Until recently, it has been generally supposed that the relative anti-knock value of two fuels measured under one set of conditions was approximately similiar under other conditions. In practice, different laboratories, working with varying equipment and technique, were unable to agree on the relative ratings of fuels of widely different character, notably lead blends versus benzol blends. In general, the attitude of each laboratory was to suspect the other of careless or inaccurate test methods.

In the latter part of 1928, the Matériel Division, partly as a

[1] Mechanical Engineer, Power Plant Branch, Air Corps, Wright Field.

[2] "The Air Cooling of Petrol Engines," A. H. Gibson, D.Sc., Proceedings Institution of Automobile Engineers (London), vol. 14, 1919–20.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

result of its work on the use of liquid cooling media at temperatures in excess of 180 deg. fahr. and partly as a matter of accident, found that cylinder temperature had an extremely market effect upon the relative ratings of fuels. The engine used was an early form of the Series 30 Ethyl Gasoline Corporation Knock Test Engine, with 8 to 1 compression ratio and an operating speed of 600 r.p. m. Detonation was measured with a bouncing pin. The amounts of benzol and lead required in two different gasolines to equal a standard consisting of 80 parts (by volume) of 2–2–4–trimethyl pentane and 20 parts normal heptane were obtained at cooling-liquid temperatures varying between the limits of 150 and 350 deg. fahr. (with circulating evaporative cooling). The tests in question were rather crude and could not be repeated to close limits. However, repeat tests showed that there was no question of the general order of the effect, and investigation by other laboratories has shown similar effects. The results of these tests are shown in Fig. 1. The outstanding feature is the rapid relative loss of anti-knock value of benzol with increase of cylinder temperature. Other aromatic compounds, such as ethyl benzene and a mixture of alkyl benzenes, both produced by organic synthesis by the Carbide and Carbon Chemicals Corporation, showed a similar loss of anti-knock effect with increase of cylinder temperature.

The results shown in Fig. 1 at once indicated that, with the

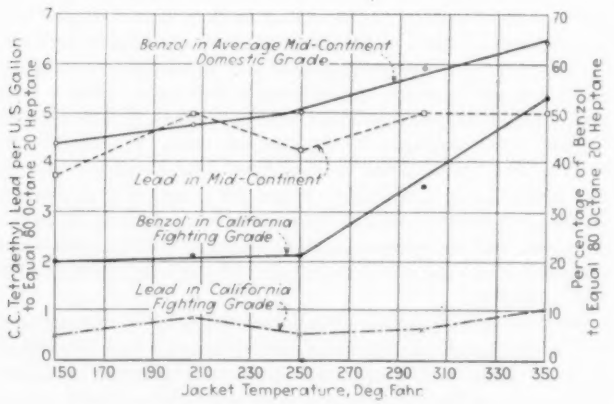


FIG. 1

wide variation of cylinder temperature found in aircraft engines, fuel-test engine conditions would need to be so adjusted as to strike an average of the relative effectiveness of widely varying fuels as determined in multicylinder aircraft engines. It appears obvious that with the wide differences in cylinder temperature, compression ratio, volumetric efficiency, rate of revolution, etc., found in various types of aircraft engines, the relation between lead and benzol will vary widely and that it will not be possible to do more then set test-engine conditions to average the results obtained.

The Matériel Division has started a program to determine the relation between the anti-knock effects of widely different fuels in multicylinder aircraft engines. For the present, the method in use is to determine the minimum amounts of tetraethyl lead and benzol in a base gasoline which, from the anti-knock standpoint, is so poor that the engine under test will not operate on it without substantial anti-knock additions. It is planned to test a variety of types of engines in this manner, if possible, with different base gasolines and then to adjust test-engine conditions so that they reproduce the average anti-knock relation between lead and benzol found in multicylinder engines. As later discussion will show, almost any variation of test-engine conditions seems to affect the relation between lead and benzol. It may therefore prove necessary to attack the problem from the standpoint of obtaining the amounts of a given anti-knock required in a series of widely differing base gasolines to produce maximum output, minimum fuel consumption, etc., in any given engine. The relation between fuels is obtained by determining the minimum amount of anti-knock that will give maximum output, minimum fuel consumption, and minimum cylinder temperature with the carburetor so leaned out as to drop the power 1 per cent (best setting) from the maximum available at any fuel consumption on the fuel under test. Maximum power, minimum fuel consumption, and minimum cylinder temperature are usually obtained simultaneously (at best setting) when sufficient anti-knock compound has been added to the base fuel, and further anti-knock additions do not produce any measurable effect. As an alternative to changing the method of determination of relationship between fuels of different types in multicylinder engines, it may be necessary to entirely change the method of single-cylinder test of fuels.

In all single-cylinder engine tests of fuels, the Matériel Division has adopted the procedure of testing at that air-to-fuel ratio which produces maximum tendency to knock. The necessity for this has been clearly shown by Campbell, Lovell, and Boyd.[3]

In the early work with high cooling-liquid temperatures in fuel-test engines, the Matériel Division installed cylinder-head thermocouples to determine the relation of test-engine cylinder temperatures and those of hot running aircraft engines. In determining maximum knock mixtures strength for a given fuel and comparing fuels of diverse knock characteristics, it was noted that differences in knock intensity were invariably accompanied by differences of cylinder-head temperature. As this offered a possibility for an even more direct method of determining detonation than the bouncing pin, a temperature method was sought which would show much smaller changes of knock intensity than could be detected by the cylinder-head temperature. Furthermore, as the limiting factor in the use of any fuel in an aircraft engine appears to be the tendency to heat the cylinder and piston unit, the method possibly has a sounder basis for evaluating aircraft-engine fuel than one relying upon detection of audible detonation or rate of pressure rise.

In order to magnify temperature differences for small changes of knock intensity, temperature measurement of an indirectly cooled surface within the combustion chamber seemed the best line of attack. The plugs shown in Fig. 2 were therefore constructed and fitted into the bouncing pin hole in the Series 30 Ethyl Gasoline Knock Test Engine. Operating at 900 r.p.m., all three types of plug show a minimum change of 3 deg. fahr. for a change from 3 to 3⅛ cc. of tetraethyl lead in several gasolines. Up to 3 cc. of lead, the sensitivity shown for a change of concentration of ⅛ cc. of lead appears to be independent of lead concentration.

At 350 deg. fahr. jacket, the type 3 plug shows a temperature change of 5 to 7 deg. fahr. for a change from 50 to 51 per cent benzol in California Fighting Grade gasoline.

Occasionally, the fuels were found that, when matched by the bouncing pin, did not match audibly or by cylinder-head temperature. Fuels which match by the cylinder-temperature method will sometimes show a pronounced difference in audible detonation.

A considerable amount of investigation will be necessary before it is possible to determine whether the bouncing-pin method, the temperature method, or, possibly, the method of testing by incipient knock, gives the closest correlation with the tendency of fuels to limit aircraft-engine output.

[3] "Importance of Mixture Ratio in Rating Fuels for Knock," John M. Campbell, Wheeler G. Lovell, and T. A. Boyd, *Industrial and Engineering Chemistry*, vol. 20, no. 10, October, 1928.

Using the temperature plug in a Series 30 Ethyl Gasoline Knock Test engine, running at 900 r.p.m., the Matériel Division obtained the tetraethyl lead equivalent of 50 per cent C.P. Benzol in a California Fighting Grade and a Mid-Continent Domestic Aviation Gasoline at cooling-liquid temperatures varying between 150 and 400 deg. fahr. The results shown in Fig. 3 indicate very large variations with change of cylinder temperature by this method of test. Preliminary tests made by the Ethyl Gasoline Corporation, and shown in Table 1, using a similar engine at 350 deg. fahr. jacket temperature and the same Mid-Continent D.A.G. and benzol, but different California Fighting Grade gasoline, show the effects of engine speed and knock intensity upon lead-benzol ratings by both bouncing-pin and temperature-plug methods of test. The lead equivalents of 50 per cent of benzol in the two gasolines were determined by both methods at two engine speeds and at two knock intensities. The effect of engine speed in changing the relative rating between lead and benzol, shown in Table 1, has been observed in the testing of undoped fuel. A highly naphthenic California gasoline was matched with a benzol Mid-Continent D.A.G. blend at 600 r.p.m. and 350 deg. fahr. jacket temperature. On raising the engine speed to 900 r.p.m. without other change than increase of throttle opening, the amount of benzol in the D.A.G. required to match the California was found to be considerably increased.

Some investigators have obtained a sensitivity of knock determination of considerably less than 1/2 per cent of benzol in a given fuel under fixed engine conditions. While this is a remarkable achievement, its usefulness as applied to aviation fuels would, from the effects shown in Figs. 1 and 3 and Table 1, appear to be open to doubt.

It would seem that a series of tests in a variable compression engine working at full throttle with both varying jacket temperature and engine speed would be of much value. The Air Corps, however, is not in a position to carry out such tests at present.

The foregoing discussion of determination of anti-knock value and the effect of test-engine conditions upon it serves to show that it is not possible to do much more than generalize upon the subject of anti-knock values of fuels for aircraft engines and that a large amount of multicylinder-engine data will be necessary for any-

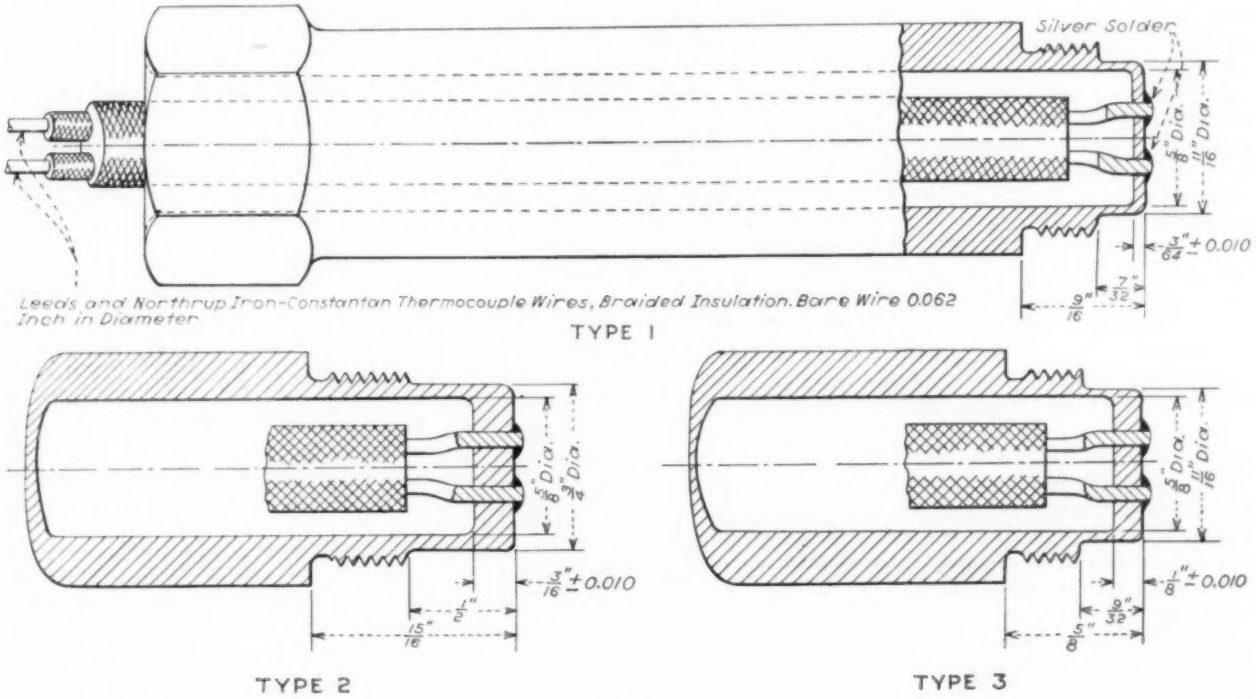


FIG. 2

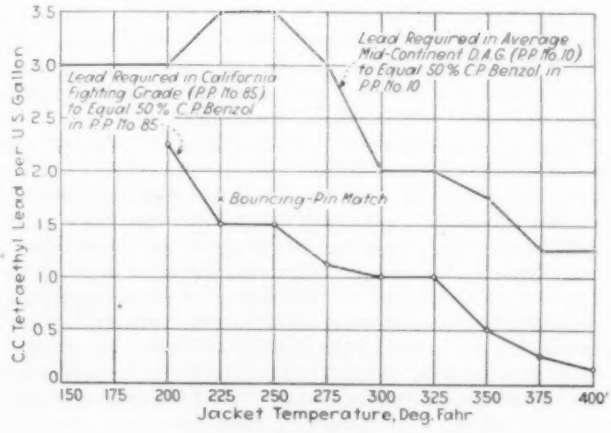


FIG. 3

TABLE 1

Fuel	Engine speed, r.p.m.	Throttle opening, deg.	Compression pressure, lb. per sq. in.	Spark advance, deg.	Cc. of lead to equal 50 per cent benzol By bouncing pin	By temperature plug	Plug temperature, deg. fahr.
P.P. No. 10 (Mid-Continent D.A.G.)	600	8 1/2	86	20	1.70	1.20	454
	600	11	106	20	2.10	1.70	556
	835	11	77	20	0.85	0.60	507
	835	13	87	20	1.10	1.00	522
P.P. No. 104 (California Fighting Grade)	600	10	102	20	1.05	0.50	512
	600	12	115	20	1.30	0.80	543
	835	14	94	20	0.45	0.20	522
	835	16	104	20	0.80	0.70	558

thing like a complete understanding of the problem. Some sort of knock test, however, for fuel for the modern aircraft engine of high performance is vastly better than no test whatever, as it will provide at least a partial control of anti-knock value. The Air Corps has equipment now in service which will fail in a few minutes if operated on fuel of lower anti-knock value than that on which it has been developed. The Air Corps has therefore issued tentative specifications for anti-knock fuel, using its best present judgment as to the specified test conditions. Revision of these specifications as further experience is obtained will undoubtedly be necessary.

In setting any specification for anti-knock value, it is highly desirable to have a reproducible primary knock standard consisting of a mixture of two pure compounds for reference purposes. The Heptane-Octane (2–2–4–trimethyl pentane) scale, first proposed by Graham Edgar,[4] has for the present been adopted by the Air Corps. This scale is at present considered preferable to the normal Heptane C.P. Benzol scale used by Brown,[5] Stansfield and Thole,[6] and others, as the temperature effect shown in Figs. 1 and 3 does not cause the disturbance with the Heptane-Octane scale that it possibly does with a scale involving the use of aromatics.

SECTION 2 AVAILABLE FUELS AND ANTI-KNOCKS

1 Fuels

The Matériel Division has taken the standpoint that it will adopt the best possible base gasoline having any wide availability and to this add the maximum usable quantity of anti-knock material, of which large supplies can, if necessary, be obtained.

The practice of the Air Corps until recently was to use Mid-Continent D.A.G. and to add benzol to this when fuel of superior anti-knock value was required. For service purposes, benzol in excess of 20 per cent cannot be used owing to freezing in the fuel tanks and lines, and even this amount causes considerable additional trouble with ice formation in the carburetors and induction systems when the water content of the atmosphere is high. For special purposes, such as racing, benzol contents have occasionally been as high as 80 per cent. In service, if trouble with ice formation in the induction system is to be avoided, the freezing point of the fuel should not be much in excess of —60 deg. cent., and this limits the benzol concentration to approximately 20 per cent.

Other aromatic compounds, such as toluene, are free from the freezing disadvantages of benzol. Alkyl benzene has a freezing point of approximately —60 deg. cent., and in concentrations up to about 50 per cent, apparently has a better anti-knock value than benzol. The production of toluene is quite limited and hardly equal to the demand and would likely not be available at all in time of emergency. The supply of alkyl benzene depends upon benzol production, but due to the increase of volume as a result of synthesis and the higher anti-knock value, 100 gal. of benzol synthesized to alkyl benzene produces approximately the same effect as 270 gal. of benzol. Alkyl benzene apparently could be competitive in price with benzol for equal anti-knock effect.

It is not considered that present or possible production of any aromatic anti-knock would warrant relying upon the exclusive use of such compounds, even in the best available gasolines for the anti-knock fuel requirements of the Air Corps. A service-test program of alkyl benzene added to highly anti-knock gasoline in supercharged air-cooled engines is shortly to be undertaken.

When the Air Corps started to investigate fuel problems, the first step was to test as many of the available aviation gasolines as possible with a view to obtaining as good a base fuel as possible. The result of this test program was to show that available California straight-run aviation gasolines were equal to average Mid-Continent D.A.G. plus 25 to 50 per cent benzol. Some cracked gasolines at lower cylinder temperatures were found to be better than the best California gasolines. As the California gasolines with tetraethyl lead added seemed to be subject to relatively less loss of anti-knock value with increase of cylinder temperature in terms of Heptane-Octane than any fuel blend tested, a first-class California Fighting Grade gasoline with 3 cc. of lead added was tentatively chosen as a standard. A variety of cracked gasolines have been rated at 900 r.p.m. and 350 deg. fahr. jacket temperature on the Series 30 Ethyl Gasoline engine. Only one cracked gasoline that will meet Air Corps requirements for gum has been found that will equal the tentative standard with less than 6 cc. of lead per gallon. Present Air Corps requirements for anti-knock gasoline specify the Series 30 Ethyl Gasoline Knock Test Engine with 300 deg. fahr. jacket temperature at 600 r.p.m., and it has been found that this reduction of jacket temperature and engine speed results in several gum-stable, cracked gasolines meeting the specified anti-knock rating of 87 Octane-13 Heptane with less than the allowable maximum lead concentration of 6 cc.

Cracked gasolines have been little used for aviation purposes, as the Federal specifications for aviation gasolines allow only 3 mg. of copper-dish gum. While this requirement is almost impossible to meet with cracked fuels, the Matériel Division has found cracked gasolines that do not increase gum in storage, and while having higher values than the Federal specifications allow, do not cause trouble from gum deposits in service. The Matériel Division operated one 9-cylinder 430-hp. air-cooled radial engine and one 12-cylinder 425-hp. water-cooled vee engine on cracked gasoline, showing a glass-dish gum value of 10 mg. after accelerated aging (test described later) and containing 3 cc. of lead. The air-cooled engine was flown for 66 hours and the water-cooled engine for 128 hours, both exclusively on this fuel, and on stripping the engines, no difference in condition due to the use of cracked gasoline in place of straight-run fuel could be detected.

Low and stable gum values in cracked gasoline appear to be obtainable. Even completely vapor-phase cracked gasolines of exceedingly high anti-knock value can apparently be made gum-stable by the use of gum inhibitors. It would seem that the use of such gasolines for anti-knock aviation fuels is worthy of study and test. So far, the cracked gasolines tested by the Matériel Division at high cylinder temperature have proved inferior to straight-run California gasolines when lead is added to both. Whether the relative inferiority is due to loss of anti-knock value of the cracked base as a result of the temperature effect or whether tetraethyl lead is less effective in raising the anti-knock value of such bases has yet to be determined, at least so far as the Matériel Division is concerned.

While the California gasolines, so far, appear to be the best available base, the problem of supply of such fuel in many of the states away from the seaboard is rather difficult, and cracked gasolines appear to offer a possible alternative. Developments in cracking may result in the production of gasolines that are less liable to loss of anti-knock value at high cylinder temperature, and investigation of this phase would seem desirable.

The depreciation of anti-knock value of cracked gasolines at high charge temperatures appears to be a generally observed

[4] "Measurement of Knock Characteristics of Gasoline in Terms of a Standard Fuel," Graham Edgar, *Industrial and Engineering Chemistry*, January, 1927.

[5] "Knock Test Apparatus and Methods Are Standardized by Michigan Engineering Experiment Station," George Granger Brown, *Automotive Industries*, January 25, 1930.

[6] "Standardization of Conditions for Measuring the Detonation Characteristics of Motor Fuels," R. Stansfield and F. B. Thole, *Industrial and Engineering Chemistry*, Analytical Edition, vol. 1, no. 2, April, 1929.

effect. Other laboratories, using test methods quite different from those of the Matériel Division, have found this.

The anti-knock value at fairly high cylinder-temperature conditions of the best base fuel that the Matériel Division feels can have any wide availability is approximately 75 Octane-25 Heptane. This is about equal to that of doped and cracked automobile gasolines now sold in large quantity, although many of these, while giving this anti-knock value and higher at 212 deg. fahr. jacket temperature, will not do so at 300 deg. fahr. While the initial anti-knock value of the base may be roughly 75 Octane-25 Heptane, different fuels of this initial anti-knock value may, and do, require considerably varying additions of anti-knock compound to bring them to the present Air Corps standard of 87 Octane 13 Heptane.

Volatility. The Air Corps has tentatively adopted Federal Fighting Grade volatility requirements (modified as discussed under Vapor Lock, Section 5) for its anti-knock gasoline specification as it has been found that even such volatile fuel as this requires considerable heating of the intake system for maximum fuel economy.

The use of fuel of average automobile volatility (30 per cent over at 100 deg. cent. and 200 deg. cent. plus end point) in military aircraft engines almost always involves a very considerable drop in output. A recent test on a Curtiss D-12 engine with an automobile gasoline of ample anti-knock value showed a drop in power of 10 per cent over that given by D.A.G. In the same engine, the use of very volatile natural gasoline (which cannot be considered for service use with present fuel systems) gives an increase of power and reduction of fuel consumption over that obtainable with D.A.G., the improvement in power no doubt being due to increase of volumetric efficiency as a result of greater evaporation and increase of charge density prior to entrance to the cylinder and the reduction of fuel consumption to improved distribution.

Gum. The recently issued tentative Air Corps Specification for Anti-Knock Fighting Grade Gasoline has eliminated the copper-dish gum requirement and tentatively substituted an accelerated aging test with oxygen. This test consists essentially of treatment with oxygen at 100 lb. pressure for two hours at 212 deg. fahr., followed by evaporation of 100 cc. in a glass dish, the maximum allowable gum being 15 mg. This test is an early form of one now in use by the Standard Oil Company of New Jersey[7] and to whom its use by the Air Corps is due.

The anti-knock requirements of the recently issued specification are such that some refiners will probably supply cracked gasoline. The accelerated oxygen aging test has therefore been adopted in view of the known unreliability of the copper-dish test for other than straight-run gasolines. So far, no gasoline has been found that will pass the accelerated aging test specified and develop excess gum in storage, although there is some suspicion that some gasolines which will not pass it may not develop excess gum in storage.

The test has been adopted tentatively, pending an investigation by the Bureau of Standards of accelerated aging tests which will duplicate gum stability in storage.

The Air Corps has subjected samples of straight-run and cracked gasolines to accelerated aging and tested the anti-knock rating of the aged against the unaged samples. In the case of a straight-run California Fighting Grade gasoline, the aged sample required less than 0.2 cc. lead to equal the unaged. In the case of a vapor-phase cracked gasoline which increased its gum value by approximately 1700 mg. on aging, over 10 cc. lead was required to bring the aged sample up to the unaged.

[7] "Gum in Gasoline," E. B. Hunn, H. G. M. Fischer, and A. J. Blackwood, *S.A.E. Journal*, January, 1930.

2 Anti-Knocks

The available anti-knocks at present are: Aromatics, organo-metallics, carbonyls, aromatic amines, and the alcohols. Of the organo-metallic compounds, only tetraethyl lead has been used commercially. Iron carbonyl has been used to some extent in Germany. Nickel carbonyl has been investigated as regards its anti-knock value, but beyond this does not seem to have been used in multicylinder engines. The aromatic amines, such as aniline, ortho-toluidine, xylidine, etc., have been the subject of considerable research and have been used to some degree for racing and special purposes, but have found no general application. Anhydrous ethyl alcohol in the smaller concentrations is, apparently, a better anti-knock than benzol. Owing to its high latent heat of evaporation, it increases volumetric efficiency and internally cools the cylinder unit, and on this account as been used largely for racing purposes in high concentrations, mainly in air-cooled motorcycle engines. In some racing-aircraft engines, overheating and failure have been avoidable only by the use of fairly heavy alcohol concentrations. Only anhydrous alcohol is soluble in gasoline, and even in mixture is exceedingly hydroscopic and picks up water from the atmosphere, with the result that the alcohol is almost entirely thrown out of solution and the anti-knock effect confined to one layer of the fuel.

The viewpoint and experience of the Matériel Division with the aforementioned anti-knock compounds are as follows:

(*a*) *Aromatics.* Aromatic compounds, particularly those with low freezing points, are mainly of use for adding to gasolines in cases where the base fuel cannot be brought up to requirements by the use of dopes alone. Aromatic compounds added to doped fuels when the aromatic concentration is of the order of 30 per cent have in several cases resulted in an increase of fuel consumption of the order of 3 per cent over that obtained with a doped fuel of similar anti-knock value and low aromatic content.

(*b*) *Organo-Metallics.* Tetraethyl lead in the form of Ethyl fluid has been rather extensively used by the Air Corps in the last year. This, particularly when added to highly naphthenic fuels such as the California gasolines, makes an exceedingly good fuel for hot-running, high-compression, supercharged engines. It has not been found possible to exceed a rating of about 95 Octane 5 Heptane with any gasoline and 6 cc. of lead at high cylinder temperature, and this only with the best of the California gasolines. When ratings in excess of this are required, it is the practice of the Matériel Division, for the present, to use benzol, toluene, or alkyl benzene in California gasoline with 6 cc. of lead added to the mixture. A detailed discussion of the operating results with Ethyl aviation gasoline is given later.

(*c*) *Carbonyls.* The experience of the Matériel Division is confined to iron carbonyl, and this only in knock-test engines. At 350 deg. fahr. jacket temperature, it was found that spark-plug missing occurred within 3 or 4 minutes after switching over to fuel containing approximately 7 cc. of iron carbonyl per gallon. Experiment with this material has been abandoned for the moment.

(*d*) *Aromatic Amines.* The Matériel Division has used commercial mixed xylidines and commercial ortho-toluidine in a few high-compression engines for racing and has done some single-cylinder endurance testing. The experience is too slight to draw conclusions from except that the single-cylinder tests showed a tendency to foul the engine, particularly the spark plugs. Tests made at 350 deg. fahr. jacket temperature at 900 r.p.m. in the Series 30 Ethyl Gasoline engine, using the temperature plug for determination of detonation, showed that 3 cc. of lead in a high-grade California Fighting Grade gasoline was equaled by 11 per cent of commercial ortho-toluidine and 89 per cent of the same gasoline. Thus, for the same anti-knock effect, ortho-toluidine would cost at least ten times as much as lead.

(*e*) *Alcohols.* The Matériel Division has no experience to date with alcohols as anti-knocks and for the present does not plan to investigate their possibilities.

The experience of the Matériel Division with anti-knocks to date would indicate that, as regards availability and usefulness in modern air-craft engines, tetraethyl lead has the greatest possibilities in raising general service-fuel standards. For limited use, to supplement lead, aromatics have a distinct field which would be greatly increased by increased production and availability of aromatic compounds, in particular those of low freezing point.

Furthermore, it is the opinion of the Matériel Division that in raising the anti-knock standard of service fuel to the highest possible standard of any general availability, dopes are not a remedy for the use of a poor base fuel. Rather, as good a base as possible should be employed and to this be added the maximum usable quantity of dope, and if further improvement in anti-knock value is required, aromatic compounds may be added. In this connection, a test of some interest may be quoted. A high-grade California gasoline plus 6 cc. of lead was tested against the same gasoline-lead mixture plus 30 per cent C.P. Benzol, i.e., 4.2 cc. of lead per gallon. The test was carried out at 350 deg. fahr. jacket temperature and detonation measured by the temperature-plug method. The gasoline-benzol-lead blend was decidedly inferior to the gasoline-lead blend, although exactly how much so was not determined. Thus, the last 1.8 cc. of lead produced more anti-knock effect than 30 per cent of benzol.

Tests of the same California gasoline containing 6 cc. of lead against a blend of 70 per cent of this gasoline, 30 per cent commercial toluene, and 4.2 cc. of lead showed a match by the same method as used for the benzol blend. A match as regards power and cylinder temperature was also obtained in a multicylinder liquid-cooled engine with 7.3 compression ratio and operating at a jacket temperature of 300 deg. fahr. The aromatic blend, however, gave a 3 per cent higher fuel consumption at best setting.

Tests in a highly supercharged water-cooled engine of 5.8 compression ratio, equipped with a gas-turbine supercharger showed that, with another California gasoline, the gasoline with 6 cc. of lead was superior to a blend of 80 per cent this gasoline, 20 per cent motor benzol, and 4.8 cc. of lead.

SECTION 3 FUEL REQUIREMENTS OF MODERN AIR-CRAFT ENGINES

The anti-knock value of the fuel required by aircraft engines of even similar type varies considerably. The fuel required will depend upon the compression ratio, the volumetric efficiency (including the use of supercharging to increase this factor), the engine speed, the design and size of the cylinder and piston unit, and the operating temperature of the cylinder unit.

The Curtiss D-12, water-cooled, 12-cylinder, vee engine of 1160 cu. in. capacity and 5.6 to 1 compression ratio, is a most remarkable engine as regards its ability to give high output on relatively poor fuel. This engine at 180 deg. fahr. jacket temperature has a peak b.m.e.p. of 138 lb. per sq. in. at approximately 1900 r.p.m. and develops this on average Mid-Continent D.A.G. (approximately 48 Octane-52 Heptane knock rating) at a specific fuel consumption of about 0.52 lb. per b.hp-hr. At 2300 r.p.m., 130 lb. b.m.e.p., and 435 b.hp. are developed, with a specific fuel consumption of approximately 0.5 lb. on Mid-Continent D.A.G. With 7.3 compression ratio and 180 deg. fahr. jacket temperature, approximately 4 cc. of lead is required in Mid-Continent D.A.G. (approximately 77 Octane-23 Heptane knock rating). 512 b.hp. or 152 lb b.m.e.p. is developed at 2300 r.p.m. on a specific fuel consumption of 0.45 lb. At 5 6 compression ratio and 300 deg. fahr. jacket temperature, 425 b-hp. or 127 lb. b.m.e.p. is developed at 2300 r.p.m. on a specific fuel consumption of 0.44 lb. with approximately 2 cc. of lead in Mid-Continent D.A.G. (approximately 73 Octane-27 Heptane knock rating). The same performance is obtained with 40 per cent of Motor Benzol and 60 per cent Mid-Continent D.A.G. At 7.3 compression ratio and 300 deg. fahr. jacket temperature, 488 b-hp. or 144 lb. b.m.e.p. is developed at 2300 r.p.m. on a specific fuel consumption of 0.41 lb., the fuel being California Fighting Grade, plus 6 cc. of lead (knock rating approximately 91 Octane-9 Heptane). In this case, by leaning the mixture out so as to produce a 3 per cent power drop from the maximum, a specific fuel consumption of 0.39 lb. has been obtained.

The Curtiss V-1570, 12-cylinder, vee, water-cooled engine is similar to the Curtiss D-12 engine, except that it has open-end sleeve aluminum-block cylinder construction in place of closed-end sleeve, has 1570 cu. in. capacity instead of 1160 cu. in., and the compression ratio is 5.8 to 1. At 2400 r.p.m. and 180 deg. fahr. jacket temperature, 620 b-hp. or 130 lb. b.m.e.p. is developed on Mid-Continent D.A.G. with approximately $^1/_2$ cc. of lead, with a specific fuel consumption of 0.47 lb.

With 8.3 compression ratio at 180 deg. fahr. jacket temperature, 700 b.hp. or 147 lb. b.m.e.p. is developed on a specific fuel consumption of 0.41 lb. with a Fighting Grade California Gasoline plus 6 cc. of lead, having an approximate knock rating of 88 Octane-12 Heptane, this fuel being barely good enough from the anti-knock standpoint, but the engine condition in this test was somewhat doubtful.

A somewhat modified version of this engine, with a geared supercharger and 5.8 compression ratio, gave the following performance at 180 deg. fahr. jacket temperature and 2400 r.p.m.:

848 b.hp. or 178 lb. b.m.e.p., specific fuel consumption 0.53 lb.

810 b.hp. or 170 lb. b.m.e.p., specific fuel consumption 0.45 lb.

The fuel used was a California Fighting Grade gasoline with 6.8 cc. of lead (approximate knock rating, 88 Octane-12 Heptane). The pressure in the intake manifold in the foregoing tests was 12 in. Hg. above atmospheric.

Another modified version of this engine, similar to the standard engine, except that it has smaller valves and somewhat different manifolds, gave the following performance:

600 b.hp. or 126 lb. b.m.e.p. at 2400 r.p.m. with 180 deg. fahr. jacket temperature on a specific fuel consumption of 0.49 lb.

At 300 deg. fahr. jacket temperature, 574 b.hp. or 121 lb. b.m.e.p. was obtained at 0.48 lb. specific fuel consumption and 578 b.hp. or 122 lb. b.m.e.p. at 0.53 lb. specific fuel consumption.

The fuel used in these tests was Mid-Continent D.A.G. with 3 cc. of lead (approximately 75 Octane-25 Heptane, knock rating) which was of course unnecessarily good for the tests at 180 deg. fahr. jacket temperature.

The Matériel Division has as yet to find any water-cooled engines that, on a given fuel, will either develop equal unsupercharged mean effective pressure or allow of equal compression ratio without supercharging, as do the Curtiss D-12 and V-1570 engines. Several water-cooled engines have been tested that will develop only 120 lb. b.m.e.p. or a little over on average Mid-Continent D.A.G. and that cannot use compression ratios in excess of 5.4 to 1 on this fuel at ground level.

Air-cooled engines, in general, at equal brake mean effective pressure demand considerably better fuel than the best of the water-cooled engines. In fact, the Air Corps has only one air-cooled engine in service that will give satisfactory full-throttle operation at ground level on Mid-Continent D.A.G. This

engine develops only 220 hp. and a little over 120 lb. b.m.e.p. at 1800 r.p.m.

One 9-cylinder, radial, air-cooled engine, of approximately 1750 cu. in. capacity, develops 130 lb. b.m.e.p. at 1900 r.p.m. with 5.2 compression ratio and a specific fuel consumption of about 0.52 lb. The fuel required is approximately $1^1/_2$ cc. of lead in average Mid-Continent D.A.G. At 8.0 to 1 compression ratio, 144 lb. b.m.e.p. was obtained at 1900 r.p.m. with 0.45 lb. specific fuel consumption. The fuel used was 70 per cent California Fighting Grade gasoline, 30 per cent Motor Benzol, with 6 cc. of lead in the mixture. The anti-knock value of the fuel blend used is not known. However, the California gasoline used has approximate knock ratings of 87 Octane-13 Heptane with 3 cc. of lead and 91 Octane-9 Heptane with 6 cc. of lead. The engine could not be operated at full throttle with the best California gasoline and lead alone; at least not with any lead concentration that the Air Corps considers worthy of test.

One supercharged, air-cooled, 9-cylinder, radial engine, in flight, gives very satisfactory operation on Mid-Continent D.A.G. plus 3 cc. of lead (approximately 75 Octane-25 Heptane knock rating) if the throttle is not opened fully below 5000 ft. altitude. This engine has 5.2 compression ratio and at 2300 r.p.m. develops a pressure at the intake valves of approximately 9.5 in. Hg above atmosphere. On the dynamometer with any fuel, this engine is liable to stick pistons in a few minutes at full throttle and 2300 r.p.m. if the specific fuel consumption be lowered below 0.6 lb. This is no reflection on the engine in question, as it is not designed for ground boosting, but rather as an altitude engine. A change to a piston design with a liberally ribbed interior which results in very considerable piston cooling by the oil in the crankcase (and sharply increases the temperature rise of the oil through the engine) resulted in full throttle dynamometer operation being possible. The engine, equipped with the ribbed pistons, was run for about 12 hours at full throttle, developing 620 b.hp. or 159 lb. b.m.e.p. on a specific fuel consumption of 0.51 lb., the fuel used being a California Fighting Grade gasoline with 6 cc. of lead and an approximate knock rating of 88 Octane-12 Heptane. This performance is of interest on two grounds (*a*) It shows the marked influence of design on fuel requirements, and (*b*) in comparison with the performance obtained from the 8 to 1 compression ratio, 1750 cu. in., 9-cylinder, air-cooled engine referred to above, with a limitation on the fuel used, supercharging offers greater possibilities for increase of output than does high compression alone.

The relation between the maximum output obtainable from the Curtiss V-1570 at 180 deg. fahr. jacket with 8.3 compression ratio and with supercharging, in both cases on approximately the same fuel and working close to the limit of the fuel, is also of interest in showing the advantage of supercharging over high compression for maximum output. This engine also shows that supercharging to high output can be obtained with relatively excellent fuel consumption.

The anti-knock value of the fuel required by an engine can be materially diminished by increasing the fuel consumption over that necessary for "best setting" (specific fuel consumption giving 1 per cent less than maximum power). In general, the policy of the Air Corps is against such a procedure, which, while it produces engines of low specific weight, tends to excessive total fuel consumption and neglect of cylinder design.

In the light of present knowledge the Air Corps proposes to specify a standard fuel and to rate engines in terms of the power they will produce on this fuel at specified maximum allowable fuel consumption at ground level or with an air density at the carburetor air intake equal to that at 5000 ft. altitude, depending upon the duty for which the engine is required. While the rated power will be obtainable at the stated r.p.m., under the conditions given, either increase of specific fuel consumption or of the anti-knock requirement of the fuel will be required when in full climb for most engines, as the resultant reduction of engine speed causes an increased tendency to detonation. Callender, King, and others[8] obtained the figures shown in Table 2, which shows this effect, on a Ricardo E-35 variable-compression engine. While the same fuel was used over the speed range, the usable compression ratio increased with engine speed.

TABLE 2

R.p.m.	Highest useful compression ratio
1200	4.45
1300	4.65
1400	4.90
1500	4.95
1600	5.00

The effect of engine speed in allowing increase of b.m.e.p. with a given fuel will likely be of decided value in the future, as the tendency is for engine speeds to increase.

As has been stated, the design of the cylinder unit has, under any given set of conditions, a very pronounced effect upon the output obtainable from a given fuel. Cooling of the cylinder head and barrel, valve cooling, piston cooling, spark-plug location, and combustion-chamber shape all have their influence. It is the policy of the Air Corps that the maximum of cylinder and engine development shall take place around a standard fuel and that engines shall be made to suit fuels, rather than fuels to suit engines.

Engines will be rated in terms of their performance on standard fuel and at specified fuel consumption, rather than in terms of what may be got out of them with a super-fuel and at excessive fuel consumption. This should tend to improve cylinder design and cooling and put an end to the rather prevalent system of cutting engine weight at the expense of fuel consumption, often with the result that, despite the reduction of engine weight, the total power-plant weight, with fuel for one hour's flight, is increased.

The Heptane-Octane anti-knock ratings of the fuels referred to in this section are, as can be gathered from the discussion on "Determination of Anti-Knock Value," Section 1, at the best only approximate, but represent the best average figures available for the fuels in question and at least mean more than quoting the amount of lead or benzol in D.A.G. or Fighting Grade. The percentages of Heptane and Octane referred to are all by volume, although weight percentages represent a more accurate and desirable method of using this scale.

The case of the liquid-cooled engine with a limitation on the maximum anti-knock value of the fuel available may well be considered. A high temperature of the cooling liquid is desirable, as this diminishes the size of the radiator required and thus the parasitic cooling drag. Practically all work with cooling liquids of high boiling point to date has been done at 300 deg. fahr. but it is possible and probable that the optimum jacket temperature will depend upon the type of engine and the type of airplane it is used in.

Increase of jacket temperature reduces the possible compression ratio and the degree of supercharging. The highest possible compression ratio is desirable as it improves fuel economy and reduces the specific heat rejection to the cylinder walls. A high degree of supercharging is desirable, as it produces the greatest increase of output on a given fuel, at the expense, however, of fuel economy and total heat rejection by the cylinder and piston unit.

It is obvious, therefore, that in any particular engine the opti-

[8] Aeronautical Research Committee (British) Reports and Memoranda No. 1062, H. M. Stationery Office.

mum condition for each installation will depend on a nice balance between radiator size as affected by both jacket temperature and compression ratio, fuel economy as affected by compresson ratio, and output as affected by both compression ratio and degree of supercharge, the latter having the major effect, of course.

In the case of a bomber, where the radiator drag is but a small percentage of the total for the airplane and where the maximum ground power available for take-off is required, it is possible that a jacket temperature of the order of 180 deg. fahr. will be most desirable. In this case, fuel economy is of importance for long range; therefore, the compression ratio will probably be such as to cause no undue sacrifice of range.

In the case of a pursuit airplane, where ground power for take-off is ample and speed and the ability to go to high altitude are of paramount importance, the minimum radiator size and the maximum degree of supercharge will probably be aimed at.

SECTION 4 CORROSION AND ATTACK DUE TO FUEL IN AIRCRAFT ENGINES

Corrosion problems due to fuel are a great deal more severe in aircraft engines than in automobile engines. The steel cylinder barrel almost always used in aircraft engines is largely responsible for this difference, as the cast-iron cylinder of the automobile engine is relatively much less susceptible to corrosion. In addition, many aircraft engines have unlubricated valve gears, with the result that exhaust-valve corrosion and sticking are a great deal more prevalent than in the average automobile engine with its oiled valve gear.

The use of doped fuels containing halogens in many cases results in a considerable amount of corrosion trouble with resultant difficulty in service that is not met with in normal undoped fuels. Corrosion having been a quite general experience with doped fuels, it has become a habit to attribute all such trouble to the use of this type of fuel. It is worthy of note, however, that as bad a case of engine corrosion as any yet experienced by the Matériel Division occurred in an engine that had been operated on 20 per cent benzol mixture, the benzol containing 0.37 per cent sulphur and causing heavy corrosion in the copper-dish test, large, loose flakes of copper sulphide being formed. This fuel blend caused rusting of the cylinder bores and produced a heavy dark-gray deposit in the exhaust ports of the aluminum head.

The corrosion problems occurring in a variety of types of aircraft engines as a result of the use of doped fuels are discussed below.

1 Cylinder Corrosion

The use of tetraethyl lead in conjunction with the theoretical quantity of bromine to convert all the lead present to lead bromide during combustion results in most cases in a slight amount of corrosion in the top part of the cylinder bore with normal cylinder-barrel material such as 0.40 to 0.50 carbon steel. This corrosion, which occurs while the engine is standing, usually extends from $^1/_2$ to 1 in. below the upper limit of piston-ring travel. That any operating difficulty results from this corrosion there is little or no evidence. Similar corrosion appears to occur at times with cast-iron cylinder bores, and yet no trouble is known to occur from this source in automobile engines operated on Ethyl gasoline. The higher the operating temperature of the engine, the the less is this corrosion as a rule. For a considerable period, Ethyl gasoline containing 50 per cent bromine in excess of that necessary to convert all lead present to lead bromide was in use. This excess bromine resulted in a great deal more corrosion than that caused by the use of the present mixture containing only the amount necessary for complete conversion to lead bromide.

Corrosion of the cylinder barrels due to the condensation of aqueous products of combustion which have blown by the piston rings and which condense on the cylinder walls below the pistons during standing causes some trouble. This would seem to be affected by the method of crankcase breathing. One radial, air-cooled engine which breathes through the hollow rear section of the crankshaft into the rear accessory drive housing has caused trouble in this respect, although there is no evidence that such trouble would not occur with undoped fuel. In the matter of crankcase breathing, it is worthy of note that some of the automobile companies make it an invariable practice to place the crankcase breather on the highest point of the case so that the aqueous products of combustion will be expelled on standing, due to the fact that their density is less than that of air. An air bleed into the bottom of the crankcase allowing the entrance of air to replace the blown-by products of combustion which may be discharged to atmosphere by reason of their relatively lower density is worthy of consideration on aircraft engines. This would be similar in effect to the crankcase ventilation systems used in automobile engines.

Corrosion of cylinder bores of cast-iron cylinder automobile engines as a result of condensation of products of combustion of undoped fuels within the combustion chamber is known to occur quite rapidly after shutting down, and in practice causes but little trouble.

In composite aluminum and steel cylinders, the use of tetraethyl lead in the fuel results in a sticky white liquid exuding from the aluminum portions exposed to products of combustion after the engine has stood some time. This liquid, which appears to be mainly an aqueous solution of aluminum bromide, rapidly corrodes carbon-steel cylinder bores, etc., that it may come in contact with if they are dry of oil. As a rule, the presence of oil is sufficient to prevent such corrosion, and in many cases a cylinder will not corrode after removal from an engine until it is washed off with gasoline. It is the practice, therefore, in the Air Corps to oil-slush cylinder unit parts that have been operated on doped fuel, after washing off with gasoline. Attempts have been made to eliminate cylinder corrosion by injecting pressure oil into the induction system while the engine is being shut down, the purpose being to spread an oil film over the inside of the cylinders and over the exhaust-valve stems in the case of unlubricated valve gears. This has been tried with a large air-cooled radial engine with unlubricated valve gear and no direct breathing from the crankcase. The engine has been several times flooded with oil while shutting down and with such quantity of oil as to cause very heavy smoking. No spark-plug fouling has occurred, and corrosion of the tops of the cylinder bores seems to be eliminated. Corrosion of the cylinder bores below the pistons due to condensation of products of combustion within the crankcase has not been eliminated, and rusting of exhaust-valve (cobalt chrome steel) stems occurred in every test, although reduced by the oil treatment.

Limited experience with an inverted 12-cylinder, vee, air-cooled engine with a completely lubricated overhead camshaft valve gear has, to date, shown an entire absence of corrosion due to the use of doped fuel. This is apparently due to the fact that oil drains down the cylinder bores after the engine is shut down. An entire absence of corrosion of valve heads and stems has been observed to date, although the cobalt-chrome valve steel used will corrode fairly rapidly in an unlubricated valve gear.

The Matériel Division is conducting and sponsoring experiment with corrosion-resisting material for cylinder barrels in composite aluminum cylinder construction. Troubles arise in composite aluminum and steel cylinder construction due to the fact that the coefficient of thermal expansion of the aluminum alloys used for the cylinder heads is approximately twice that of the carbon steel normally used for cylinder barrels. The use of corrosion-

resisting material for cylinder barrels at once suggests that the disparity of thermal expansion of the cylinder head and barrel can be, to a large degree, eliminated by the use of the austenitic stainless steels for barrels. The coefficient of expansion of aluminum is only 30 to 40 per cent greater than that of some of the austentic steels and cast irons. This results in the possibility of a great reduction of the differential expansions which cause much trouble in composite cylinder construction. The shrink of the cylinder barrels in the heads can be reduced, with the result that less stress is put in the head, and there is less chance of permanently stretching the head. In addition, the cylinder barrel is less liable to go bell-mouthed at the top of the bore when the shrink of the head on the barrel is relieved as the engine heats up to operating temperature. Piston conditions will be largely improved on account of the reduction of piston clearance allowable. This will improve piston cooling and tend to improve piston bearing conditions, due to reduced rocking of the piston. The reduction of piston rocking and lessening of bell-mouthing of the bore at the head end will further materially improve piston-ring conditions and wear.

The use of cylinder-barrel material with a higher coefficient of thermal expansion than that of normal carbon steel is not a new idea. Experiment with bronze sleeves in aluminum cylinders was carried out in England by Gibson[9] in 1916-18, but was abandoned, owing to excessive wear of the sleeves.

Most of the austenitic nickel-chromium stainless steels, containing 8 to 20 per cent nickel and 12 per cent and over of chromium, have been found to have very high resistance to attack by both condensed aqueous acid products of combustion from doped fuels and to the constituents in the hot gases tending to cause attack. In these respects the austenitic steels have proved superior to the stainless steels containing chromium only and which approximate in coefficient of thermal expansion to the normal carbon steels.

With increase of operating temperature in modern liquid-cooled engines and composite aluminum and steel cylinder construction, the use of cylinder-barrel material with a higher coefficient of expansion than that of the carbon steels would appear to be extremely desirable in order to reduce stresses in the cylinder heads, maintain liquid tightness of the seal between the head, and generally reduce differential expansion between the aluminum portions of the cylinder and the barrel. Thus, the added cost of the corrosion-resistant barrel would appear to be justified upon other grounds than the use of doped fuel. Forged and cast austenitic nickel-chromium steels and austenitic cast-iron containing nickel and copper are being experimented with. One European aircraft-engine constructor reports that attempts some three years ago to use austenitic nickel-chromium stainless steel cylinder barrels resulted in rapid scoring of the barrel, despite the use of castor oil. It is reported that austenitic cast-iron cylinder barrels have already been successfully used in automobile engines, and instead of scoring readily, the material rapidly work-hardens to a hard glaze. Tests carried out for the Matériel Division of austenitic cast-iron and forged-steel barrels for composite aluminum cylinder liquid-cooled engines have to date shown:

(*a*) That the cast iron rapidly develops a hard high glaze. The physical strength is so much lower than that of carbon steel, however, that a barrel that will not fail mechanically cannot be obtained with the space and design limitations of the engine in question.

(*b*) That in the same engine as used for the tests of cast-iron barrels, the forged austenitic steel barrels eliminate the leakage experienced with carbon-steel barrrels at 300 deg. fahr. jacket temperature. They are, however, exceedingly difficult to machine and present problems due to scoring and excessive piston-ring wear.

The importance of differential expansion in the cylinder and piston unit is shown by the vast amount of experiment that has gone on in the automobile-engine field directed to reduction of clearance between the cylinder bores and aluminum pistons. It is noteworthy that the use of aluminum alloy with considerably reduced thermal expansion is making some headway for pistons.

It would appear that austenitic cylinder-barrel material is more readily applicable to liquid-cooled than to air-cooled engines, as the thermal conductivity of such material is only of the order of 30 per cent of that of normal carbon steel. In the case of the air-cooled engine with fins integral with the barrel, the fin thickness and weight would of necessity need to be approximately trebled. The use of fins of higher conductivity attached to the barrel with a sound thermal bond, or the reversion to a cylinder construction, using an integral aluminum cylinder head and finned barrel with a shrunk-in austenitic steel cylinder sleeve, would appear possible. The last-mentioned cylinder construction proved to be unsatisfactory with carbon-steel barrels, due almost entirely to differential expansion. In any case, the cylinder corrosion troubles due to the use of doped fuel in air-cooled engines with lubricated valve gear largely appear to be capable of solution without the use of corrosion-resistant cylinder-barrel material. Many air-cooled engines now in use in the Air Corps are using doped fuel with negligible trouble from cylinder corrosion. In most of the large air-cooled engines in use in the Air Corps, the elimination of doped fuel in favor of any readily commercially available anti-knock fuel would result in overheating and other troubles which would be a great deal more serious than those resulting from the use of dope.

2 Valve Corrosion and Attack

The use of fuels doped with tetraethyl lead will cause, and in many cases has caused, a great deal of operating trouble due to corrosion of exhaust valves. This seems to have two phases: rusting due to condensed acid products of combustion which occurs on the stem and seat of the valve and erosion or attack of the seat and other portions of the valve head by hot products of combustion. The first cause is much the more serious as it causes sticking and often, as a result, rapid wear of the valve guide and stem.

Corrosion can be entirely eliminated by the use of suitable material in unlubricated valve gears. Materials unsuitable for use with unlubricated valve gears can be made to function satisfactorily with lubricated valve gears without great difficulty in the majority of cases. The use of some of the austenitic nickel chromium valve steels appears to be an almost complete answer to corrosion due to fuel doping with tetraethyl lead, even with unlubricated valve gears.

Nitrided Silchrome No. 2 steel is probably the most unsatisfactory valve steel as regards use with Ethyl gasoline that the Matériel Division has as yet had experience with. In an aluminum cylinder, liquid-cooled engine with aluminum-bronze valve seat inserts and completely lubricated valve gear, this material was rapidly attacked on the seats of the valves. The attack was sufficiently rapid to require frequent replacement of the valves. Whether this attack was due to condensed products of combustion or to the hot gases is not known. The same material when used in an air-cooled engine with a partially lubricated valve gear caused a great deal of trouble with valve sticking. This sticking was unknown when undoped fuels were used.

Tungsten steel with analysis within the range used for high-

[9] "Resume of Experimental Work on Air-Cooled Cylinders at the Royal Aircraft Establishment," A. H. Gibson, D.Sc., Internal Combustion Engine Sub-Committee Reports No. 24, H. M. Stationery Office.

speed tools rusts extremely rapidly in unlubricated valve gears when used with Ethyl gasoline. This, however, is hardly of importance, as its use had been almost entirely abandoned on account of its tendency to burn and scale. Tungsten steel was in use in some modern engines in conjunction with internal valve cooling which, by reducing valve temperature, largely prevented scaling. Its use in this case was due to its remarkable wear resistance, but it had to be rapidly abandoned when Ethyl gasoline was used in the engines in question.

Cobalt chrome steel has given a great deal of trouble in unlubricated valve gears with Ethyl gasoline, particularly on the sea coast, and may be regarded as generally unsatisfactory for such use on account of rusting. As far as experience is available, this material appears to be satisfactory for use in lubricated valve gears. Silchrome steel, with and without tungsten (approximately 2 per cent), has proved satisfactory in lubricated valve gears, but rusting of valve seats is at times evident, though no operating trouble has yet been traced to this cause.

All in all, the austenitic nickel-chromium valve steels, some containing tungsten in small quantity, appear to be the best for use with doped fuels. These steels, which have found considerable favor apart from use with doped fuel, have the advantages of remarkable strength at high temperature, are extremely tough at all temperatures if the silicon content is 1 per cent or below, and most of them have high resistance to corrosion and to erosion and scaling by hot products of combustion of doped and undoped fuels. They have the disadvantages of low proportional limit which sometimes results in stretching, cannot be hardened as can cobalt chrome and silchrome steels, and thus do not resist wear as well, and further machining is difficult in the case of hollow-stem aircraft-engine valves. The difficulties of lack of wear resistance can be almost entirely overcome as regards the valve stem by lubrication and reduction of side thrust. Wear on the tip of the stem can be overcome by welding on a hard tip or fitting a hardened plug.

One rather curious effect of the use of Ethyl gasoline in an air-cooled, radial engine has developed in extended endurance with a considerable amount of supercharging. After a large number of hours' running, excessive cylinder temperature and preignition occur, and on removal of the cylinder a deposit is found on the exhaust-valve head in the combustion chamber. On removing this deposit, the cylinder temperature returns to normal and the preignition is absent. The valve steel used contains 7.5 per cent Ni, 12.5 per cent Cr, 2.5 per cent Si, and the deposit is apparently due to lead-bromide attack (see under Exhaust Disposal system), as it is apparently almost entirely magnetic iron oxide. In the later discussion of attack of materials by lead bromide at high temperature, it will be noted that the steel used for those valves is not by any means the best as regards resistance to this attack. Attack does not occcur in the engine in question in normal service, even with a considerable amount of running with supercharging. Therefore, although the trouble is rare, every attention should be given to exhaust-valve cooling in the design of engines for use with doped fuel, particularly in the case of engines with high cylinder temperature and supercharging.

In any engine designed for the use of doped fuel it is desirable that ample valve-gear lubrication be provided. This should be of the circulating oil-splash type. With most of the best of the modern valve steels, ample valve and valve-gear lubrication is an almost complete answer to valve corrosion, as a result of the use of doped fuels.

3 Exhaust Disposal System

Shortly after a general adoption of Ethyl gasoline in service by the Air Corps, numerous cases of airplanes being set on fire by hot particles blown from the exhaust pipes occurred while idling taxying, warming up, or shutting down the engine. In no case did these fires occur in the air, being confined to conditions when the air blast over the wings was sufficiently low to allow the hot particles to settle on the wing surfaces.

Investigation showed that the particles causing the fires were flakes of iron oxide, mainly the magnetic oxide (thus proving that the scale was formed at high temperature). The question of the relative attack of exhaust piping by doped and undoped gasoline and of the relative resistance of a variety of materials to such attack was investigated by the Ethyl Gasoline Corporation. The first method of investigation was to burn doped and undoped gasoline under identical conditions in blow torches, the flame being passed through $1^1/_2$-in. outside diameter by 24-in. long pipes and being sufficiently intense to maintain a zone of 6 in. length at red heat. The results of these tests were as follows:

(*a*) *Mild Steel.* With undoped gasoline: Tube extensively scaled, but scale adherent and none blown out during the 20-hr. test. With Ethyl gasoline: Slightly more scale than with undoped gasoline. Scale decidedly less adherent and some blown out of the discharge end of the pipe during the test. With undoped gasoline plus ethylene dibromide: It was thought possible that the corrosion might be due to the ethylene dibromide in the Ethyl fluid burning to free hydrobromic acid. The amount of ethylene dibromide present in the Ethyl gasoline used in the foregoing test was therefore added to undoped gasoline and a third test was run. This showed corrosion similar to that experienced with undoped gasoline rather than that with Ethyl gasoline.

(*b*) *Monel Metal.* The attack with both undoped and doped gasoline was imperceptible.

(*c*) *Austenitic Nickel-Chromium Steel, 18% Cr, 9% Ni, 0.1% C.* Imperceptible attack with either undoped or doped gasoline.

(*d*) *Mild Steel Calorized.* With undoped gasoline: No perceptible attack. With Ethyl gasoline: No attack in hot zone of tube. Discharge end shows fairly heavy formation of loose, rusty scale, of which about 50 per cent was magnetic.

As the attack did not appear to be due to acid products of combustion, tests were made of the corrosion effects of lead bromide at high temperature. Pieces of a variety of metals about 1 in. sq. were treated with 0.3 gm. of lead bromide and heated to a temperature above the melting point of lead bromide, but below red heat. The heating was continued until all lead bromide was volatilized off, this usually taking about 2 hr. This test is probably too drastic, but the results in general accord with the tests on tubes and experience with various materials in service. The results are as follows:

(*e*) *Mild Steel.* Heavy and very rapid scaling; scale non-adherent.

(*f*) *Mild Steel, Chromium-Plated.* Attack just as heavy and rapid as with mild steel.

(*g*) *Monel Metal.* Very slight attack; deposit produced by attack quite adherent.

(*h*) *Pure Nickel.* Almost imperceptible attack.

(*i*) *Pure Aluminum Sheet.* Fairly heavy attack; deposit adherent. Lead bromide largely reduced to metallic lead.

(*j*) *Nickel-Chromium Steel. 18% Cr, 9% Ni, 0.1% C.* Attack intermediate between Monel metal and mild steel, but relatively slight. Scale less adherent than with Monel metal.

(*k*) *Nickel-Chromium Steel. 18% Cr, 9% Ni, 2½% Si, 0.1% C.* Attack less than in (*j*) and very much localized, producing spots of not very adherent scale, about 0.01 in. thick.

(*l*) *Nickel-Chromium Steel. 17% Cr, 2% Ni, 2.75% Si, 0.2% C.* Attack less than in (*k*), approximately that of Monel metal, but more localized.

(*m*) *Chromium Iron. 13% Cr, 0.1% C.* Attack similar to that of (*j*) but more localized; scale more adherent also.

The Matériel Division has also conducted some crude tests of

lead-bromide resistance at high temperature. Strips of various materials were heated to a bright-red heat and powdered lead bromide was repeatedly dropped on the upper surface and allowed to volatilize off, the treatment being continued for about 10 min., as a rule.

(*n*) *Mild Steel.* Exceedingly rapid attack with the production of heavy, loose, black scale within 5 min. of treating with lead bromide. Mild steel similarly heated without lead bromide showed almost imperceptible scaling.

(*o*) *Nickel-Chromium Steel, 11% Cr, 34% Ni, 4% Si, 0.25% C.* Continued heating for over half an hour produced only a slight, tightly adherent film of black scale.

(*p*) *Cobalt-Chrome Valve Steel, 18% Cr, 1.2% Co, 1.2% Mo, 1.4% C.* Fifteen minutes' heating produced a loose, powdery, non-adherent, reddish-black scale about 0.003 in. thick. This steel, when used in unlubricated valve gears, is heavily attacked whereas it shows almost no attack in lubricated valve gears. The attack by lead bromide is slightly less than that shown by material (*q*). Therefore it would appear that the attack experienced in valve service with cobalt steel is mainly rusting.

(*q*) *Nickel Chromium Valve Steel, 7.5% Ni, 12.5% Cr, 2.5% Si, 0.3% C.* Attack somewhat more pronounced than in the case of (*p*); scale non-adherent. This is a well-known valve steel that gives very satisfactory service for valves and valve seat inserts with heavy lead concentrations, showing neither rusting and, very rarely, scaling. This steel in service, as regards rusting, gives vastly better results than (*p*).

(*r*) *Nickel-Chromium Valve Steel, 19% Ni, 13% Cr, 2% W, 0.40% C.* Better than any steel except (*o*) tried in the Matériel Division tests.

(*s*) *Invar. 33% Ni, Balance Fe.* Attack very slight. Similar to material (*q*).

(*t*) *Heat-Treated Aluminum Bronze, 11% Al, 5% Fe, 5% Ni, 0.1% Mg, Balance Cu.* Rapid attack with the production of non-adherent black scale.

(*u*) *Extruded Aluminum Bronze, 10.5% Aluminum, Balance Copper.* Exceedingly heavy attack with the production of non-adherent black scale which cracks off in cooling. Attack worse than in the case of mild steel under similar treatment.

(*v*) *Cast-Aluminum Alloy, 4% Cu, 2% Ni, 1.5% Mg, Less Than 1.25% Si and Fe, Balance Al.* Slightly pinholed portion of aluminum piston. Heated at below red heat, but to the softening point. Heavy attack and production of relatively large patches of metallic lead. Where the exhaust system does not get sufficiently hot to produce mechanical failure, cast-aluminum exhaust stacks of this alloy have given exceedingly satisfactory service with Ethyl gasoline, no significant attack being experienced.

While the test of heating in the presence of lead bromide as used by both the Matériel Division and the Ethyl Gasoline Corporation is, as previously stated, probably too drastic, it provides approximate data as to the relative resistance of materials to lead bromide at high temperature. The results obtained accord, in general, with attack experienced in various engine parts. The resistance of austenitic nickel-chromium steels and the violent attack of aluminum bronze in particular parallel experiences in hot-running, air-cooled engines as regards valves and valve-seat inserts.

As regards material for exhaust piping, materials (*g*), (*j*), and (*k*) would appear the most promising at the moment, and the final choice will probably depend upon the ease of fabrication and cost, although a variety of considerations enter into the question of the most suitable material apart from corrosion resistance. Complicated and elaborate exhaust disposal systems, such as ring manifolds on radial, air-cooled engines and the extensive exhaust piping used on exhaust-turbine superchargers in particular involve a number of problems other than resistance to temperature and corrosion. Some of the problems involved are discussed below:

(*a*) *Coefficient of Thermal Expansion.* It is not known if the mild steel generally used at present has the most suitable coefficient of expansion. A low coefficient of expansion will, in general, tend to reduce thermal stresses due to temperature differences in the pipe structure itself, particularly at welds, but may cause increased stress as a result of attachment to the engine.

At average operating temperatures of ring manifolds, chromium steel or chromium iron has approximately the same coefficient of expansion as mild steel. Monel metal has approximately 35 per cent greater, and the austenitic nickel-chromium steels, such as materials (*j*) and (*k*), have approximately 65 per cent greater expansion.

(*b*) *Ease of Drawing, Forming, Beating, Etc.* Monel and the austenitic nickel-chromum steels mostly seem to be more difficult to handle in the foregoing respects than dead-soft mild steel, but to present no great difficulties in the fabrication of complicated shapes to those used to working them. The shops at the Matériel Division rather prefer dead-soft annealed austenitic nickel-chromium steel, such as material (*j*), to soft mild steel, as regards forming and beating.

(*c*) *Welding.* In the present state of knowledge, welding would appear to be the simplest method of building up complicated exhaust systems. None of the most suitable materials appears to weld as readily as mild steel.

It is desirable that a material for such use should produce non-spongy welds, that the welds should not be brittle, and that brittleness should not be produced in the metal adjacent to the weld. All the stainless steels discussed appear to be liable to brittleness, either in the weld or in the adjacent metal, unless heat treatment after welding be used. It is also desirable that the welds should have a tensile strength as close as possible to that of the surrounding material. Monel metal appears to braze readily with a spelter consisting of 80 copper 20 zinc which has a melting point of approximately 1600 deg. fahr., which should be more than adequate for any exhaust disposal system worthy of use in aircraft. Even if 80–20 brazing be insufficient, it can readily be reinforced with riveting. None of the stainless steels, either of the straight chromium or the nickel-chromium type, appears to braze well, if at all.

(*d*) *Strength at High Temperature.* Other things being equal, it is desirable that the material should lose as little strength as possible with increase of temperature. From the standpoint of fire hazard in crashes the temperature of the exhaust disposal system at any point should be held below 600 deg. fahr., but this hazard is largely being forgotten or disregarded in the case of radial air-cooled engines. In the case of exhaust-turbine supercharger exhaust piping, local temperatures considerably in excess of 600 deg. fahr. appear at the present to be unavoidable. Therefore, it is desirable that the hot strength of the material used should be as high as possible, in case of accidental overheating, although soft mild steel has to date given little trouble in this respect. The austenitic nickel-chromium steels have the highest strength at elevated temperature of any commercially available material. Monel metal at 1000 deg. fahr. is superior in tensile strength to low-carbon mild steel.

The attack of the exhaust disposal system by the products of combustion of Ethyl gasoline does not, in a suitably designed system, cause relatively greater wastage or attack that is of importance from the viewpoint of life of the equipment. The attack is only of importance from the standpoint of the fire hazard introduced. Any increased rate of wastage of exhaust systems of automobile engines, due to the use of Ethyl gasoline rather than undoped gasoline, would appear in general to be negligible.

Further, the use of corrosion-resistant material for exhaust systems cannot be charged to doped fuel alone, as the commonly used mild steel has been far from satisfactory without the use of

doped fuel, particularly in the case of ring manifolds on large air-cooled engines. Ring manifolds for such engines have been the subject of much investigation in order to reduce burning, corrosion, and scaling, particularly for seaplane use. A variety of stainless steels, very heavy nickel plating of mild steels, and other methods of producing resistance to corrosion and scaling have been worked on. Fortunately, the materials that provide adequate resistance to Ethyl gasoline also are very satisfactory as regards burning, scaling, and salt-water corrosion.

4 Valve-Seat Insert Attack

In air-cooled engines with aluminum-bronze valve seat inserts, mostly of material (*u*), sinking of the exhaust valves into the inserts with fair rapidity is at times experienced with Ethyl gasoline. This occurs either in full-throttle ground testing or in flight, if cooling is inadequate. With some engines, it seems to be a general occurrence under almost any conditions of running and is probably due to inefficient exhaust-valve cooling.

No case of such valve-seat sinking has yet been observed with liquid-cooled engines using cooling-liquid temperatures varying between 180 and 300 deg. fahr. This would indicate that its occurrence is largely a function of valve seat insert and valve temperature.

The tests of aluminum bronze, described in Section 4, subhead 3, show that this material is violently attacked by lead bromide, whereas the austenitic nickel-chromium steels, which have similar coefficients of thermal expansion, are mostly highly resistant. A coefficient of thermal expansion as close as possible to that of aluminum alloy is desirable in order that the inserts shall remain tight in aluminum cylinder heads. Extended full-throttle tests by the Air Corps of materials (*q*) and (*r*) for exhaust-valve seat inserts in an air-cooled engine using gasoline with 6 cc. of tetraethyl lead per gallon showed that both materials were superior to aluminum bronze, such as material (*u*), but that the latter was in reasonably satisfactory condition after 90 hours' running without valve attention. The bronze inserts showed very slight black scaling, whereas the steel inserts were glazed. The valves used in this case were of cobalt-chrome steel, internally cooled, and of approximately 55 Rockwell C hardness on the seats. Similar material to (*r*) eliminated valve-seat sinking in European air-cooled radial engines of 200 hp., operating on Ethyl gasoline in Canada. In this case, valve-seat sinking of the order of $^1/_8$ in. in 50 hours of flying occurred with aluminum-bronze seats and solid cobalt-chrome steel valves. This trouble was apparently due to overheating, caused by faulty installation conditions. Dynamometer tests of this type of engine with inadequate cooling showed that out of a variety of insert materials, including aluminum, bronze and two or three valve steels, the only one immune from sinking was austenitic nickel-chromium steel, and it is understood that since the adoption of this material, no further cases of trouble have occurred.

A rather serious disadvantage of the austenitic nickel-chromium steels for seat inserts is that after considerable service the seating work hardens to a glaze that no ordinary seat cutter will cut, and Carboloy or similar cutting material will probably be required for refacing.

Monel metal would appear to be worthy of trial as a valve-seat insert material. Its resistance to corrosion and lead-bromide attack is high. Its coefficient of thermal expansion (0.15×10^{-5} per deg. cent., 25 to 300 deg. cent.) is between that of bronze and ordinary steel. Ordinary steel inserts can, by suitable precautions, be made to stay tight in an aluminum head. Thus, Monel metal should present no difficulty on this score. The thermal conductivity of Monel metal is 0.06 in c.g.s. units, approximately twice that of the austenitic nickel-chromium steels, which is of some advantage in a seat insert. Whether Monel metal will glaze so that it cannot be recut with ordinary seat cutters will be determined by experiments now in progress.

5 Valve Gear

In general, the use of Ethyl gasoline tends to somewhat increase the corrosion of valve-gear parts, such as valve springs, camshafts, rockers, etc. Such corrosion, due to the condensation of products of combustion within the valve-gear housing, is by no means unknown even with undoped fuel, heavy pitting of valve springs and camshafts being not uncommon. The greater acidity of the products of combustion of Ethyl gasoline shows a tendency to increase such rusting and pitting. The remedies for this type of corrosion are: (1) The elimination of pockets which hold condensed products of combustion within the valve-gear housings; (2) removing the fumes from the valve housing as rapidly as possible; (3) oiling all parts of the gear as liberally as possible; and (4) avoiding such overcooling of the valve-gear housing as tends to increase condensation. A further remedy applicable to upright vee or in-line engines, the valve gears of which cannot in general be very liberally oiled owing to leakage of oil down the intake and exhaust valve stems, appears to be some oiling scheme, such as reservoirs which would slowly discharge oil accumulated during running over the valve gear, as the engine cools after shutting down.

In the case of inverted vee engines with overhead camshaft valve gears, the limited experience available suggests that valve-gear corrosion is negligible as the valve gears can be very heavily sprayed with oil without danger of an excess passing to the valve ports via the valve guides.

6 Spark Plugs

The use of Ethyl gasoline has certain effects on spark plugs which do not occur with undoped fuels. Lead deposits, probably largely lead bromide, occur on the insulator and on the inside of the shell. These deposits, in the case of high-class mica plugs while disturbing in appearance, appear to cause little or no trouble. Furthermore, lead deposits almost invariably result in an entire lack of carbon deposits and, on the whole, are much less troublesome than the latter.

Test of 12-cylinder vee engines with half the cylinders running on Ethyl gasoline (3 cc. lead) and half on benzol mixture showed that electrode burning was less with Ethyl gasoline than with benzol mixture. Tests of Ethyl gasoline containing only the theoretical amount of bromine to convert all lead present to lead bromide and of gasoline with similar lead concentration (3 cc.) and 50 per cent excess bromine showed no appreciable difference in electrode attack, although less deposit on the insulators with the higher bromine concentration. Tests with 75 per cent of the theoretical bromine necessary to convert the lead present to lead bromide and a lead concentration of 6 cc. showed heavy slag-like deposits on the spark plug cores and no advantage in reduction of engine corrosion.

While in continuous or endurance running, Ethyl gasoline appears to reduce burning of the spark-plug electrodes, in general flight service, with the engine usually in operation for less than 10 per cent of the time, plug electrodes seem to waste more than they do on undoped fuel. This would appear to be due to the condensation of acid aqueous products of combustion, and prevention is being sought in the use of electrode materials other than the standard manganese nickel.

The foregoing remarks as to the effects of Ethyl gasoline on spark plugs apply only to mica plugs. The Air Corps has only a limited experience with the use of Ethyl gasoline and porcelain spark plugs. This experience, however, showed tendencies to electrode attack and fluxing of the insulators.

The foregoing conclusions with regard to the use of mica plugs apply only to lead concentrations up to 6 cc.

7 Corrosion, Conclusions

The foregoing discussion of the corrosion difficulties of doped fuel in aircraft engines may appear to indicate that the difficulties are serious and hardly worth attempting to overcome. However, it is the viewpoint of the Matériel Division that doped fuel of some type is here to stay for high-performance military aircraft engines and that the difficulties attending the use of such fuel can be overcome, as they have been in automobile engines. Further, some engines now in service in the Air Corps use doped fuel with negligible operating difficulty. Therefore, the present policy of the Air Corps is to require that new engines developed for its use shall be capable of operation on gasoline containing 6 cc. of tetraethyl lead per gallon with little or no more trouble than given by an undoped fuel of similar anti-knock value.

SECTION 5 VAPOR LOCK

The Matériel Division has kept in close touch with the work of the Fuels Research Sub-Committee of the A.P.I. and S.A.E. on vapor lock of aviation gasolines. Some work on the measurement of gasoline temperatures at various points of fuel systems in flight is being carried out by the Matériel Division for the committee.

Some vapor-lock trouble is experienced with the present military fuel systems and average gasoline obtained to Air Corps Specification 2-40F. This is largely due to the fuel pumps having to lift gasoline from the tanks rather than being drowned, although some trouble is experienced even with gravity systems.

The Bureau of Standards, which is carrying out the work for the aforementioned committee, has found that the "corrected" A.S.T.M. 10 per cent point is very closely related to the gasoline temperature at which vapor locking will occur at sea-level pressure. Examination by the Matériel Division of a large number of samples of gasolines giving reasonably satisfactory service in the Air Corps and meeting Specification 2-40F showed that the average corrected 10 per cent point was 65 deg. cent. Therefore the Air Corps, in its recently issued specification for Aviation Fighting Grade Anti-knock gasoline, has eliminated the 5 per cent point specified in Federal Specifications and has substituted a corrected 10 per cent point with temperatures limits of 62 to 72 deg. cent. This is pending further investigation of the vapor-lock problem and improvement in fuel systems.

Vapor locking in aircraft is apparently almost entirely confined to the fuel system (pipes, pump, etc.) and almost never occurs in the carburetors. Therefore improvement in fuel systems, and in particular the use of drowned fuel pumps, should allow the use of gasolines with lower 10 per cent points.

From the supply standpoint, lowering of the 10 per cent point is desirable, as some of the stabilized Mid-continent casing-head gasolines with lead added make excellent fuels for hot-running engines. Furthermore, lowering of the 10 per cent point will facilitate starting. Until fuel systems are improved, however, any lowering of the 10 per cent point cannot be considered by the Air Corps. Even with improved fuel systems, it is possible that it will prove undesirable to lower the 10 per cent point, as such lowering may produce excessive boiling of the fuel in the tanks of supercharged airplanes which climb to 30,000 ft. altitude with extreme rapidity. The fuel in the tanks under these conditions will, at altitude, be practically at ground-level temperature and therefore more likely to boil, unless steps are taken to cool it off with increasing altitude.

Attempts by the Matériel Division to use fuel with a corrected 10 per cent point of 43 deg. cent. in a pursuit airplane having a drowned pump and a cockpit temperature of 45 deg. cent. resulted in pronounced vapor-locking trouble and the engine cutting out repeatedly at 7000 to 8000 ft. altitude.

SECTION 6 LUBRICATING OIL FOR AIRCRAFT ENGINES

In the past, the ideas of the Air Corps as to the necessary qualities of lubricating oils were largely the result of the wide use of the Liberty engine and oil tests conducted with it. The advent of engines with much higher specific performance and in particular higher cylinder temperature, which need heavy oils even in winter, has led to rather drastic revisions of specifications. The study of the requirements of oil for winter use and for engines with higher cylinder-wall temperature resulted in the issue of Air Corps Specification 3556, which is a tentative specification for oil for service-test purposes directed to studying and securing better oils for the modern equipment now coming into service. Considerable study of oil for high-performance engines is now under way, and more is contemplated.

The ideal properties of a lubricating oil for aircraft-engine purposes appear to be as follows:

(*a*) Maximum oiliness, particularly at high film temperature
(*b*) Low viscosity coefficient
(*c*) Minimum tendency to decompose and gum in use
(*d*) Low pour point for winter use.

Oiliness is of pronounced importance in aircraft-engine lubrication, and the Matériel Division has almost no information regarding this property of engine lubricating oils except that castor oil possesses it to a higher degree than any other oil yet tested. It is possible with castor oil to operate engines that will stick pistons on mineral oil in a few minutes' running at full throttle. No information on the oiliness of different mineral oils is available to the Matériel Division, although it is reported that differences have been observed in this respect in engine tests of otherwise almost similar oils. Some oil refiners state that very complete dewaxing reduces oiliness, but it would seem that determination of this property under engine conditions is an exceedingly difficult problem.

A low viscosity coefficient is desirable as this results in an oil that not merely thins out less at high temperature but thickens up less at low temperatures. In a given oil, provided an adequate supply is reaching the surfaces to be lubricated, the ability to maintain lubrication appears to be approximately proportional to the absolute viscosity. Thus in engines operating at high cylinder temperature, it is desirable that the decrease of viscosity with increase of film temperature should be as low as possible. Viscosities of oils for aircraft use are measured at 100 and 210 deg. fahr., whereas the temperature of the oil film in air-cooled cylinders is at times in excess of 350 deg. fahr. Thus the importance of a low viscosity coefficient is obvious. The importance of low viscosity coefficient, as regards its influence upon engine friction when starting in cold weather, has been thoroughly investigated by Messrs. Irwin and Frank[10] of the Matériel Division, and engine friction has been shown to be almost entirely a function of the viscosity coefficient in oils of similar viscosity at 210 deg. fahr.

Provided oil is reaching the surfaces to be lubricated in sufficient quantity and is of adequate viscosity, the most common cause of failure in engines with high cylinder temperature is produced by carbonaceous deposits in the piston-ring grooves, which ultimately result in the rings sticking, with resultant blow-by and piston failure.

The Power Plant Branch of the Matériel Division has made some preliminary efforts to investigate the phenomena of piston-ring sticking. As ring sticking in a single-cylinder, aircraft engine usually results in piston failure and, further, the regulation of

[10] "Relationship of Laboratory Tests and Cold-Weather Motor Operation of Various Oils," E. R. Irwin and G. W. Frank, Matériel Division Report, Serial No. 2933.

temperature conditions with such equipment is decidedly difficult, it was decided to use a Series 30 Ethyl Gasoline Corporation Knock-Test engine. In order to accelerate the test, a cast-iron piston was used in conjunction with a jacket temperature of 350 deg. fahr. After considerable experiment the following procedure was evolved: The engine was run at 900 r.p.m. at full throttle with a standardized fuel at approximately maximum-knock mixture strength, the spark advance was regulated to produce a reading of 525 deg. fahr. ±25 deg. on the Matériel Division type 3 temperature plug, and the cylinder-head temperature was held within a limit of 20 deg. fahr.; 800 cc. of the oil to be tested was charged into a washed and dried crankcase. A piston and ring assembly, the weight of which had been obtained to within 10 mg., was installed and the engine was started up, and after reaching standard temperature conditions, it was held under these conditions for 7 hours. At the conclusion of the 7-hour period, the piston was removed, a second carefully weighed piston and ring assembly was installed, and the test was continued under similar conditions for a second 7 hours. At the conclusion of the 14 hours of testing, the oil was carefully evacuated from the crankcase and held for chemical and physical examination. Each piston, after removal, was washed off with casing-head gasoline, dried and weighed, the carbon and other deposit removed from the top of the crown, and the assembly again weighed. The piston assembly was then enclosed in a bomb, containing carbon solvent, and the temperature raised to approximately 300 deg. fahr. for about 2 hours. The piston rings were then removed and the entire piston and rings scraped free of all deposits. The rings were then replaced, the piston reweighed, and the weight compared with the original weight when installed. In cases of very excessive deposit in the top ring groove, this deposit was, in some cases, removed and weighed.

Table 3 gives the original physical properties of the oils used in the tests.

TABLE 3

Oil No.	Saybolt viscosity at 210°F., sec.	Saybolt viscosity at 100°F., sec.	Conradson carbon residue
1	100	1140	0.3
2	100	1200	1.6
3	102	1140	1.3
4	100	2400	0.6
5	121	1650	0.3

Table 4 gives the properties of the oils after draining from the crankcase at the conclusion of the tests.

TABLE 4

Oil No.	Dilution, per cent	Saybolt viscosity at 210°F., sec.	Saybolt viscosity at 100°F., sec.	Conradson carbon residue	Precipitation number
1	7.0	95	1160	0.9	0.05
2	0.2	133	2220	2.7	0.08
3	0.7	148	2290	2.8	0.08
4	2.0	119	3110	2.0	0.50
5	1.0	146	2400	1.3	0.06

Table 5 gives the results of oxidation tests on four of the oils used.

TABLE 5

	Saybolt viscosity at 210°F., sec.			Conradson carbon residue			Precipitation number		
		Oxidized			Oxidized			Oxidized	
Oil No.	Unused	7 hr.	17 hr.	Unused	7 hr.	17 hr.	Unused	7 hr.	17 hr.
1	100	116	149	0.3	0.8	1.9	0	0.0	0.0
2	100	118	281	1.6	3.0	6.5	0	0.2	7.0
3	102	128	226	1.3	2.6	5.2	0	0.0	0.0
4	100	137	...	0.6	2.5	...	0	4.0	...

Oxidation test consists of bubbling 0.5 cu. ft. of air per hour through a test tube containing 40 cc. of the oil under test, the tube being immersed in an oil bath at 420 deg. fahr., the temperature of the oil being 390 to 400 deg. fahr.

Table 6 gives the weights of total deposit and of total deposit less the weight of deposit on the crown.

TABLE 6

Oil No. 1	
First 7-hr. period:	
Top ring completely free at conclusion of test.	
Total carbon deposit on piston	2.13 gm.
Total carbon deposit less that on top of crown	1.34 gm.
Second 7-hr. period:	
Top ring completely stuck for 360 deg. of circumference at conclusion of test.	
Total carbon deposit on piston	2.59 gm.
Total carbon deposit less that on top of crown	1.94 gm.
Oil No. 2	
First 7-hr. period:	
Top ring completely free in groove on completion of test.	
Total carbon deposit on piston	1.27 gm.
Total carbon deposit less that on top of crown	0.53 gm.
Second 7-hr. period:	
Top ring completely free in groove on completion of test.	
Total carbon deposit on piston	0.98 gm.
Total carbon deposit less that on top of crown	0.82 gm.
Oil No. 3	
First 7-hr. period:	
Top ring completely free in groove.	
Total carbon deposit on piston	1.16 gm.
Total carbon deposit less that on top of crown	0.68 gm.
Second 7-hr. period:	
Top ring sticky in groove, but could be pried loose for all but 45 deg. of circumference.	
Total carbon deposit on piston	1.20 gm.
Total carbon deposit less that on top of crown	0.93 gm.
Oil No. 4	
First 7-hr. period:	
Top ring free in groove, but 0.40 gm. of deposit in groove behind ring.	
Total carbon deposit on piston	2.20 gm.
Total carbon deposit less that on top of crown	1.89 gm.
Second 7-hr. period:	
Top ring completely stuck for 360 deg. of circumference.	
Total carbon deposit on piston	3.12 gm.
Total carbon deposit less that on top of crown	2.82 gm.
Oil No. 5	
First 7-hr. period:	
Top ring quite free in groove at conclusion of test.	
Total carbon deposit on piston	1.89 gm.
Total carbon deposit less that on top of crown	1.36 gm.
Second 7-hr. period:	
Top ring completely stuck in groove for 360 deg. of circumference.	
Total carbon deposit on piston	1.85 gm.
Total carbon deposit less that on top of crown	1.61 gm.

From the results shown in Table 6, the total deposit less the deposit on the top of the crown appears to bear a very close relationship to the relative ring-sticking tendencies of the oils tested.

Neither the original properties of the oils, their condition after test, nor their resistance to oxidation appear to be a guide to ring-sticking properties, although the oxidation test would seem to give an indication, in the case of extremely bad oils.

Castor oil, remarkable for its oiliness factor, is probably by far the worst oil used in aircraft engines as regards ring-sticking tendencies.[11]

Two outstanding results are shown by these tests: (*a*) That the Conradson carbon value of the unused or the used oil is no guide to its ring-sticking and carbon-depositing tendency at high cylinder temperature; (*b*) that the naphthenic-base oils do not give softer deposits than the paraffin-base oils as far as these tests go. Oil No. 4 is a very largely naphthenic oil and gave the hardest deposits in the ring grooves of any oil tested.

Oils Nos. 2 and 3 are both Pennsylvania products. No. 3 is said to be a blend of bright stock and neutral, whereas No. 2 is said to be completely vacuum-distilled. It is notable that oil No. 2 is the only one which did not cause a stuck top ring in the last 7 hours of the 14-hour test period.

[11] "The Lubrication of Aircraft Engines," F. A. Foord, Proceedings of Royal Aeronautical Society, 1929.

The test procedure used may be too drastic and too accelerated, but nevertheless it is felt that it bears a closer relationship to the use of oils in modern aircraft engines with high cylinder temperatures than do most of the published tests at jacket temperatures of the order of 180 deg. fahr. A large program of similar tests on widely differing oils with both cast-iron and aluminum pistons and a similar test procedure to the foregoing is planned in the future.

Low pour point is necessary for oils for winter use, particularly in the case of the heavier oils required, even in winter, by engines of high cylinder temperature. This property of oils was investigated by the Matériel Division[10] in conjunction with the viscosity coefficient as regards the influence of both upon engine-starting friction and pumpability.

The Matériel Division recently conducted cold-weather starting tests on Liberty engines with the three oils shown in Table 7.

TABLE 7

Oil No.	Pour point, °F.	Saybolt viscosity at 210°F., sec.	Saybolt viscosity at 100°F., sec.	Viscosity coefficient
6.....	25	122	1500	12.30
7.....	5	124	2280	18.40
8.....	−15	113	1440	12.75

The oils had been thoroughly circulated through the engines before starting the tests, and the airplanes were allowed to stand out all night to get thoroughly cold. The air temperature at the time of the tests was 23 deg. fahr.

With oil No. 6, the engine started on the fifth attempt, and oil pressure was obtained in 3 min. from starting. The temperature of the oil into the engine rose to 41 deg. fahr. after 30 min. idling and to 50 deg. fahr. after 45 min.

With oil No. 7, the engine was extremely stiff and needed breaking loose by hand, and ten attempts were required to start it. After a start was obtained, the oil temperature and pressure reactions were similar to those of oil No. 6

With oil No. 8, a start was obtained on the fourth attempt, and oil pressure was obtained in 1 min. The oil temperature rose to 41 deg. fahr. in 20 min. idling and to 50 deg. fahr. in 30 min.

While considerable further winter testing of oils with widely different viscosity coefficients and pour points is required, nevertheless the foregoing tests are indicative of the importance of viscosity coefficient and pour point upon engine-starting friction and oil pumpability at low temperatures. As regards the work on piston-ring-sticking tendencies of oils, only the engine testing is due to the Power Plant Branch, all physical and chemical testing of the oils used being under the supervision of E. R. Irwin, Materials Branch of the Matériel Division.

SECTION 7 CARBON REMOVERS

The experience of the Matériel Division in its tests of oils for piston-ring sticking has led to the conclusion that carbon solvents are, or shortly will be, available which have an exceedingly powerful effect in removing or loosening carbon deposits behind piston rings, on the top of piston crowns and from the interior of combustion chambers. As the reliability of aircraft engines increases, and consequently the period between overhauls lengthens, the use of such carbon solvents will probably be of value. The Matériel Division has been watching the results of the use of carbon remover in a large fleet of automobiles and trucks. In this fleet, carbon remover has been in use for three years. Cylinder heads are commonly not removed in 20,000 miles of running and over, gummed intake valves can be eliminated without lifting cylinder heads, and stuck piston rings, which were frequently found on overhaul, are, with regular use of carbon remover, virtually unknown.

If the regular use of carbon solvent in aircraft engines with high cylinder temperature will serve to reduce or eliminate piston-ring sticking, its use will be more than justified. Carbon deposits in the combustion chambers of aircraft engines are usually not troublesome. However, at times they cause difficulty. In the endurance test referred to under "Valve Seat Insert Attack," heavy, hard deposits had collected at the tops of the cylinder bores, and as a result the top lands of the pistons were reduced in diameter by as much as 0.060 in. in places after 90 hours' full throttle running. Such deposits could almost certainly be eliminated by the use of carbon solvent, and their regular removal should, in addition to improving piston conditions, tend to smother full-throttle running and better maintenance of full power.

The Matériel Division is finding carbon solvent useful in removing the extensive carbon deposits that occur inside the crowns of liberally ribbed pistons of hot-running engines. These deposits interfere with piston cooling and are almost impossible to remove by scraping on overhaul, but can be readily removed even after only a 1-hour treatment with carbon remover without heating. For this purpose the Matériel Division is selecting a relatively non-toxic carbon remover owing to the inevitable exposure to fumes and skin absorption.

SECTION 8 DETONATION

It is with much trepidation that even mention is made of what detonation is or what causes it. However, the available evidence resulting from the researches of a number of workers on the fundamental aspects of the subject indicates that the cause of detonation is preoxidation of the charge during the suction and compression strokes before the ignition spark occurs. Non-knocking fuels and knocking base fuels with lead added have been shown to be materially less oxidized at the instant before the spark passes than are knocking fuels. This leads to the conclusion that, if the fuel could be injected into the cylinder late in the compression stroke, such oxidation would be materially diminished, and high compression, supercharging, and high cylinder-wall temperatures would probably be possible with fuels that knock exceedingly readily in the normal carbureted engine. The injection would of course have to be so arranged that sufficient time was allowed before the occurrence of the spark to produce complete mixing and evaporation of the injected fuel.

Note

All references to D.A.G. or Fighting Grade throughout this paper refer to straight-run gasolines, unless otherwise specified.

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Discussion

W. F. Parish.[12] Many data in the history of the development of industrial machinery will contribute largely toward immediate development of the aircraft engine, which seems to be reaching the limit of its performance with the present available fuels and mineral lubricants. Any machine, in the early stages of its development, requires a period of experimentation that will result in its continuous beneficial evolution. But this often is carried to a point at which the machine gets out of balance with the lub-

[12] Consulting Lubricating Engineer, New York, N. Y. Mem. A.S.M.E.

ricant used. As machines are developed, their speed is increased, they are increased in size, or in the power developed, or in that required per unit of weight. Their bearing pressures and the temperatures at which they operate are also increased on account of their weights and speeds and the mode in which the machine is built. Under such evolution, the machine may continue to develop to a point at which the lubricants available no longer meet the severe operating conditions demanded. The operating troubles are then such that the machine is no longer of commercial value. Unless the factors which have thrown it out of balance are corrected, the machine is of no further practical use, and some other form of equipment will supplant it.

Evolution of New Machinery

The value of a new machine will always be measured in terms of its more perfect and continuous operation. In effect, this means that it must be lubricated more effectively over long periods without difficulty. The trend of evolution has been instanced many times in the past, one being that of the early steam engines which were designed for crankcase-bath lubrication. In this particular type of engine the conditions of operation were such that the oils put into the crankcase emulsified, decomposed, oxidized, and the lighter ends distilled off. This left a heavy residue which failed as a lubricant because oil conductors would become filled with solids and the bearings continually would melt because of lack of oil and on account of interrupted oil films. The difficulties of maintaining such engines in operation proved insurmountable, even when an attempt was made to provide lubrication by introducing oil and water combinations. Such mixtures, when at all suitable in proper proportions, would soon be thrown out of equilibrium by the evaporation of the water. The cycle of gradual destruction would then again begin. All of such types of engines are now obsolete, because they could not be lubricated properly and therefore were not commercial.

Textile Spindle as an Example

The original textile spindle later had a lubricating bath which was introduced to secure more perfect lubrication. The available lubricants were oils that oxidized very rapidly and caused gumming when confined in a bath, which resulted in stoppage of the spindles. The apparatus for the application of the lubricant unbalanced the lubricant and the requirements of the machine. These factors could not be brought back into equilibrium until new lubricants were found that would not oxidize. When the desired perfect balance was finally secured, the principle of bath lubrication, using a mineral lubricant and bearings of suitable design, has greatly advanced the art of building machinery.

Inherent Difficulties Stated

The inherent difficulties with the present type of internal-combustion automotive gasoline engine, including both aircraft and marine types, are that portions of unburned fuel come into contact with the lubricant to some variable degree. Other portions of the lubricant are thrown into contact with the heated surfaces at such high temperatures that they reach the cracking point of the lighter fractions of the oil, resulting in the deposition of carbon residues. These two conditions bring about a gradual change in the lubricant. The method in vogue for the last 10 years, in an attempt to solve this apparently unbalanced condition, has been to increase the viscosity of the oil. As the development of the engines has proceeded along lines of increasing compression, speed, and power, the engines have become hotter; therefore it has been thought that thicker lubricants were required as a consequence. Little doubt exists that much of the additional heat in these engines is furnished by the increased fluid friction of the heavy lubricants. We probably are fostering a vicious cycle in having hot engines that are then lubricated with a heavier oil which, in turn, will make engines still hotter. In this valuable paper we have heard that thick, high-viscosity oils used in the engines under certain conditions are stated to become still thicker. This is due to cracking and the elimination of the lighter fractions of the lubricant. The final result is that the oil holds while in circulation a gradually increasing amount of material of the nature of carbon residue and the gums from partially burned oil. The addition of this matter eventually modifies the surface tension of the oil film. The film then ruptures, and abrasion of the surfaces follows. Gums, which act as binders of the residues and dirt that gets into the system, form behind the rings and put them out of commision. A condition such as this shows clearly that the limits of the lubricant have been reached. It shows also that the development of the engine has been carried to a point at which the available lubricants as then applied are no longer able to withstand the severe conditions.

Remedy Lies in Thinner Lubricants

If the aircraft-engine development is to proceed beyond the present point, it is obvious that we must secure cooler operating engines, and much can be done by using the thinner lubricants and changing the entire method of oil application. New forms of apparatus for cooling, filtering, and possibly rectifying the oil must be developed, and possibly also new methods of film maintenance at such points as work under the most severe conditions. No other way seems to be left open if we are to proceed with engine development. Improvements in methods of lubricant application have already been shown to be the practical remedy for the development of today's successful steam turbine, Diesel engine and many types of production machinery. These machines are, as a result, perfectly balanced, and thus are never heard of by the engineering societies as constituting an engineering problem.

Mr. Heron. As regards military aircraft engines, there is little evidence that dilution of the lubricating oil is of consequence. If excessive priming, washing the oil off the cylinder walls when endeavoring to start, can be described as dilution, then it does exist, but the dilution experienced with automobile engines does not enter the aircraft-engine lubrication problem.

The Air Corps for the present does not intend to allow the use of lubricating oils of higher viscosity than 125 sec. Saybolt at 210 deg. fahr. and 1810 sec. maximum at 100 deg. fahr., and is in agreement with Mr. Parish that the continual tendency to make hotter engines and raise oil viscosities is a vicious cycle.

The Air Corps believes, in the light of present knowledge, that if an engine needs a heavier oil than Grade 120, Specification 3556, to secure satisfactory operation, either the engine or the installation should be so modified as to produce satisfactory operation on that oil.

Encouragement by the Air Corps of the use of heavier oil would doubtless result eventually in the adoption of 200-sec. oil (at 210 deg. fahr.) of low viscosity coefficient.

The Matériel Division has no evidence to support Mr. Parish's statement that the heavier oils cause engines to run hotter. Heavier oils admittedly increase friction, but experience does not show that they increase operating temperature of the cylinder unit; in fact, rather the reverse.

While piston-ring sticking due to carbon deposits occurs in service operation of multicylinder engines in the Air Corps, this is not, as Mr. Parish states, due to the presence of considerable gum in the body of the oil in the tanks. Oil is at present completely drained from the lubrication system and replaced with fresh oil every 10 hours. Examination of the used oil does not show more than slight change; nevertheless, in hot-running the oil has decomposed and coked behind the piston rings even with such frequent change of oil.

The remedy for lubrication troubles lies very largely in design, although the best available oils must, of course, be used. Maintaining the oil fed to the engine at a suitable temperature will secure adequate bearing lubrication and prevent film breakdown there. The case of the film or partial film on the cylinder wall is another matter, however, as the final temperature of the oil on a cylinder wall at a temperature of 350 deg. fahr. would not appear to be materially affected by the temperature at which it is fed into the engine. If anything, the film temperature would appear to be reduced by increased oil-entrance temperature, as under such conditions a increased supply of oil usually reaches the cylinder walls and pistons, and thus oil cools them.

While much can be done in improvement of lubrication by better film maintenance and design to secure cooler running pistons, etc., nevertheless it is hardly possible to agree with Mr. Parish that hot-running engines must be abandoned on account of oiling difficulties. To abandon the high-temperature engine would mean giving up air-cooled and liquid-cooled engines with high cooling-liquid temperatures, and the progress of development does not point thus. The Air Corps has recently run two types of 525 hp. 9-cylinder air-cooled radial engine (both of 6 in. cylinder bore) for 146 hours without piston-ring sticking or lubrication trouble. Marked increase of specific output in both of these engines would probably produce lubrication trouble and ring sticking, but increased oil supply, improved cylinder and piston cooling, and possibly frequent removal of carbon deposits with carbon solvent would probably result in satisfactory operation with increase of output.

So far as reduction of carbon deposits is concerned, the use of oil of lower viscosity does not appear to be a remedy. Using the ring-sticking test method described in the paper, with oils of the same type, but one of 75 sec. Saybolt viscosity at 210 deg. fahr. and the other of 150 sec. at 210 deg. fahr., little difference in tendency to deposit carbon was noted.

Mr. Parish cites the Diesel engine among other types of machinery in which the lubrication problems have been satisfactorily solved. Robert Sulzer[13] has shown that in a two-cycle Sulzer Marine Diesel engine of 600 mm. bore, the top piston ring comes in contact with a portion of the cylinder liner which is at a temperature of 480 deg. fahr., and found the edges of the piston crown (liquid-cooled piston) to reach 530 deg. fahr. The piston-ring temperatures and the mean temperature of the cylinder liner were, however, very considerably lower than those occurring in modern high-temperature aircraft engine. Liner temperatures of 480 deg. fahr. at points where lubrication must be maintained do not occur in modern air-cooled aircraft engines which are worthy of consideration for military use and efficiently installed. That lubrication can be maintained under such conditions for periods of service vastly longer than those required of any aircraft engine would indicate that high temperature alone does not render efficient lubrication impossible.

With regard to filtering and reclamation of aircraft-engine lubricating oil, it is of interest to note that in the series of piston-ring sticking tests now being conducted by the Matériel Division, a reclaimed and acid-treated Mid-Continent oil has proved as good as any oil tested. This oil was reclaimed by heating, followed by filtration in a commercial reclaimer, and was then treated with sulphuric acid in the laboratory. The effect of the acid treatment was to reduce markedly the carbon deposits occurring in comparison with that produced by the reclaimed oil without acid treatment.

The worst oil by far as yet tested is "Nujol." This inferiority may be due to insufficient viscosity to maintain ring sealing and resultant gas blow-by, causing rapid gum and coke formation.

[13] "Temperature Variation and Heat Stresses in Diesel Engines," *Engineering* (London), April 2, 1926, and Proceedings of Institution of Naval Architects.

GRAHAM EDGAR.[14] Several points in Mr. Heron's interesting and valuable paper appear to the writer to warrant comment:

(1) With regard to the methods of testing fuels for anti-knock value, the writer believes Mr. Heron's conclusions absolutely sound. Once it is admitted (and there seems no doubt whatever of the actual facts) that the conditions under which a test is carried out may alter the relative anti-knock value of two fuels, it becomes clear that a rough idea must be obtained of the relative anti-knock value of various fuels *under the conditions of practical operation*, and then a set of conditions must be chosen on the test engine to duplicate as accurately as possible, the *average* values obtained under operating conditions. This is true for automobile fuels as well as for aircraft fuels. Fortunately for the automobile-fuel situation, motor gasolines seem much less affected by test conditions than aircraft fuels, so that it is a simpler problem to test them.

As experimental data on multicylinder engines are accumulated, it will be possible to specify more intelligently the ideal conditions of operation; but Mr. Heron's specifications represent at least an intelligent approach to this difficult problem,.

(2) Mr. Heron's data on the performance of high-compression and supercharged engines certainly illustrate the vast improvements which are possible with fuels of very high anti-knock value, and undoubtedly justify the stand taken by the Matériel Division that it will adopt the best possible base gasoline of wide availability and will further improve such fuel with the maximum permissible amount of anti-knock.

(3) With regard to the difficulties experienced by the Matériel Division with the use of Ethyl fluid in certain aircraft engines, the writer agrees with Mr. Heron that it would not appear to be a very difficult problem to eliminate these difficulties entirely, or to reduce them to negligible factors, by investigations by the aircraft-engine manufacturers on suitable materials for valves, valve seats, exhaust systems, etc. When Ethyl gasoline was first introduced for automotive use, some difficulties were met with certain types of valve material under conditions of severe operation, but with improvements in valve materials, together with improvements in the composition of Ethyl fluid, complaints alleging any injurious effect of Ethyl gasoline on any part of the automobile engine have become most exceptional. Under the widely different conditions of aircraft operation, and with different materials of construction, it is not surprising that new problems have arisen.

It is fortunate, also, that the answer to the problems arising from the use of Ethyl fluid in aircraft fuels appears to lie in the use of materials of construction and details of design which are advantageous for use with any fuel, at least in the great majority of cases.

MR. HERON. *Paragraph* (1). It has to be admitted that the accumulation of the relative performance of widely varying types of fuel in multicylinder engines is proving a slow process.

The Matériel Division is considering attempting, as an alternative, to carry out some tests on a highly supercharged air-cooled single-cylinder engine, the cylinder to be used being that of a modern super-performance radial engine.

Paragraph (2). The engine performance quoted in the paper has since been exceeded on two types of supercharged engines, and while none of the figures given could probably be maintained at present for endurance running, it is almost certain that such performance will be possible for lengthy periods in the next year or two.

Paragraph (3). Steady progress is being made with the solution of operating difficulties attending the use of Ethyl gasoline in the

[14] Ethyl Gasoline Corporation, Yonkers, N. Y.

Air Corps. Exhaust-disposal-system corrosion appears to be eliminated completely; nickel-chromium steel with 18 per cent Cr, 9 per cent Ni, and 0.1 per cent C appears to be the best solution. Cracking at the welds is being experienced with Monel metal exhaust rings. However, cylinder corrosion has yet to be entirely eliminated, the austenitic steel barrel presenting as yet many difficulties.

H. K. Cummings.[15] Mr. Heron is to be commended for his recognition that the knock problem in aircraft engines is not the same as that in motor-vehicle engines. The desire to obtain precise, or closely reproducible, measurements of one of the causes or effects of detonation too often obscures the real issue—measurement of that aspect of detonation which is objectionable. Measurements of detonation of aircraft and of automobile fuels may well require quite different apparatus and technique. The use of automobile fuel has an acquired dislike for the noise accompanying detonation, wholly apart from any damage which may be ascribed to the detonation. In the case of airplane fuels, overheating and mechanical shock are the objectionable features, because of their effect on the engine. With this in mind, it is apparent that in rating the detonation of fuels the mixture ratio should be adjusted to that giving the maximum intensity of the objectionable feature of the detonation reaction, rather than arbitrarily adjusting the carburetor for maximum power or, say, maximum sound intensity in the case of aircraft fuels.

While some form of temperature measurement would appear to be indicated as the most logical means for estimating the detonation of aircraft fuels, a comprehensive investigation should first be undertaken with the object of determining what type of surface shows the closest correlation to operation conditions. It is possible that an indirectly cooled surface may not even qualitatively reflect the condition of the directly cooled surface. The Bureau of Standards found cylinder-head-wall temperatures in a detonation-test engine to be practically unaffected by mild autoignition which it is believed originated from the indirectly cooled exhaust-valve surface. In this case therefore, temperature measurements on a directly cooled surface failed to distinguish the onset of a dangerous condition of an indirectly cooled surface. The opposite condition may possibly occur also.

Referring to the section on lubricating oils, it may be worth noting that Sir William Hardy believes wax desirable because of its indirect influence in promoting oiliness; that is, making it possible for the component causing oiliness to be selectively absorbed.

In connection with the carbon-deposition tests, it would be of interest to know whether the same two piston assemblies were used in each set of tests. Oil flow past the piston is one of the major factors influencing the rate of carbon deposition. One piston may wear the cylinder differently than another; hence the order in which different assemblies were used may be of importance, as well as the number of different assemblies used.

While the results shown in Table 6 do not appear to support the Conradson test as an indication of the carbon deposit, it should be remembered that the oil flow past the piston is very largely a function of the viscosity of the oil on the cylinder walls. Oil No. 4 undoubtedly was at a disadvantage from this viewpoint, and the conclusion regarding the relative carbon depositing tendency of the napthene-base oil might be modified if oils having equal viscosity at 350 deg. fahr. were used. The observation that the naphthene-base oil gave hard carbon deposits is in accord with results obtained by W. H. Herschel in engine tests at the Bureau of Standards some years ago.

Mr. Heron. Mr. Cummings has tersely stated the difference between the effects of detonation in automobile and aircraft engines; i.e., that in the former it is mostly a matter of offense to the ear and in the latter mainly a matter of overheating.

As he states, when making ratings, the mixture ratio should be adjusted so as to produce the maximum intensity of the objectionable feature; namely, sound intensity for automobile engines and overheating for aircraft engines. With the use of the temperature plug it has been found that mixture ratio giving maximum plug temperature coincides with that producing maximum sound intensity and also with that producing a maximum bouncing-pin reading. This is somewhat at variance with the general assumption that maximum knock-mixture strength is also that giving maximum power. While maximum power-mixture strength, or rather the mixture ratio giving 1 per cent less than maximum power, as a rule, in an aircraft engine, gives maximum cylinder temperature, it does not, however, produce maximum exhaust-valve temperature. Maximum exhaust-valve temperature is, as a rule, produced by a mixture ratio considerably leaner than that giving maximum power. As has been stated in reply to Dr. Becker, the temperature-plug readings in general follow those of the cylinder head in the Series 30 Ethyl Gasoline Engine; thus, the temperature plug would appear to measure effects at variance with the conditions producing exhaust valve overheating in aircraft engines.

It is agreed that a comprehensive investigation is needed of the best method of temperature rating of fuels as regards sensitivity and relation to operating conditions in multicylinder engines.

In general, in aircraft engines the overheating of pistons, exhaust valves, and spark plugs, which are all indirectly cooled, is of much greater importance than that of the directly cooled surfaces of the cylinder.

With reference to Sir William Hardy's views on wax promoting oiliness in a lubricating oil, the Matériel Division has made some preliminary tests with the Stephens Consistometer and finds that the addition of wax raises the temperature at which film breakdown occurs.

In the tests of ring-sticking and carbon-depositing properties, the same two piston and ring assemblies were used in all tests and were used in the same order in each test. Two pistons were used in order to check the progressive decomposition of the oil. Furthermore, in the case of oil No. 4, the use of a single piston for the 14-hour test period resulted in the top piston ring being so severely stuck that it could be removed only by heating in carbon remover for two days, followed by heating the ring and piston crown with an oxyacetylene torch. Mr. Cummings states that the rate of oil flow past the piston is one of the major factors causing carbon deposition. In the ring-sticking test method used by the Matériel Division increase of oil supply to the piston not only reduces carbon deposits in the ring grooves, but also on the inside and top of the piston crown. In high-temperature multicylinder aircraft engines the best-known method of avoiding ring sticking with a given oil and without design change is to increase the rate of oil passage by the rings.

Mr. Cummings remarks that oil No. 4 was at a disadvantage in the comparative tests shown in Table 6 on account of the fact that its viscosity at cylinder-wall temperature is lower than that of the other oils tested. This is possibly true, but the test was devised to investigate the ring-sticking qualities of typical aircraft-engine oils and a naphthenic oil having the same viscosity at 350 deg. fahr. or thereabouts, as the other oils tested would hardly be of much use for aircraft purposes, if such an oil could be procured.

Further testing under identical conditions has failed to show much connection between viscosity of oils of a given type and their tendency to stick rings and deposit carbon.

The test method used does not always give reproducible results,

[15] Chief, Automotive Power Plants Section, Department of Commerce, Bureau of Standards, Washington, D. C.

and further work on it to improve the reproducibility is necessary. The results in general, however, check up with observations as to ring sticking, carbon deposit, etc., experienced with the same oils in hot-running aircraft engines.

A. E. BECKER.[16] This paper contains such a wealth of valuable information that it is impossible to do justice to it in a discussion based on a hasty reading. The paper requires careful study and analysis to appreciate the full extent of the constructive work which Mr. Heron is doing on the aviation-fuel problem.

Mr. Heron's knock-rating work at high temperatures is distinctly of the pioneer type. Through his courtesy we also have had an opportunity to make some comparative measurements by the bouncing-pin and temperature-plug methods (described in the paper) in our laboratory. These measurements were made at 350 deg. fahr. on an engine similar to the Ethyl Gasoline Corporation's Series 30 Knock-Testing Engine and with the same fuels given in Table 1 of Mr. Heron's paper. For purposes of comparison, the data are given (see Table 8).

TABLE 8

Fuel	Engine speed	Throttle opening, deg.	Compression pressure, lb. per sq. in.	Spark advance, deg.	Cc. lead to equal 50 per cent benzol By bouncing pin	By temperature plug	Plug temperature, °F.
P.P. No. 10 (Mid-Continent D.A.G.)	600	10	99	20	1.7	1.3	510
	600	11½	109	20	1.9	1.8	555
	835	11	84	32	1.1	0.4	568
	835	14	105	32	2.2	0.8	613
P.P. No. 104 (California Fighting Grade)	600	10½	102	20	1.5	0.5	544
	600	13	118	20	1.2	0.9	589
	835	12	91	32	0.7	0.1	578
	835	15	113	32	1.7	0.6	645

It will be noted that these data check those of Table 1 in a general way. From the two sets of data it is obvious that the bouncing-pin method always results in more tetraethyl lead being required in one sample to match a second sample than is required when the temperature method of comparison is used. If the samples can be taken as typical of all fuels, it follows that the use of the bouncing-pin method will always result in being on the safe side when meeting a given anti-knock standard. It is suggested that a more delicate potentiometer over the temperature range met during testing might result in better differentiation between samples by the plug method. A shortcoming of this latter method is the tendency for carbon to deposit on the face of the plug in a manner similar to the description in the paper of the depositing of carbon on the exhaust valves. Just as the removing of this carbon from the exhaust valves causes a lowering of cylinder temperature and the prevention of preignition, so also the removing of the carbon from the face of the temperature plug results in the plug recording somewhat higher temperatures.

The writer can also verify what Mr. Heron says in regard to heptane-iso-octane equivalents. Two fuels which have the same heptane-iso-octane equivalents when compared at 210 deg. fahr. are not necessarily either equivalent to each other or to the same heptane-iso-octane mixture at 350 deg. fahr. On the other hand knock ratings made in our laboratory with heptane-ethyl alcohol, heptane-benzene, and heptane-iso-octane blends correlate perfectly with h.u.c.r.'s obtained by Mr. Boyd for similar blends on his variable-compression engine. Likewise a fair degree of correlation has been obtained between our knock rating scale and those of the Texas Company, Vacuum Oil Company, Atlantic Refining Company, and Standard Oil Company of New York, measurements in all cases having been made at 212 deg. fahr., although the test methods and engines were not always the same.

Extensive work similar to that just described must be done at high temperatures before it can be determined just which method will most nearly result in rating fuels in practically the same order as they are now being rated at 212 deg. fahr. Further, much additional work with multicylinder engines, as proposed by Mr. Heron, must be carried out to enable a conclusion to be made regarding airplane-fuel-knock engine-testing conditions which will correspond to average airplane-engine operating conditions. In arriving at a solution of the problem, variable-compression work, similar to that which Mr. Boyd has been doing and as suggested by Mr. Heron, should be carried out at these higher temperatures with different fuels.

It is stated in the paper that maximum power, minimum fuel consumption, and minimum cylinder temperatures are obtained simultaneously when sufficient anti-knock compound has been added to the base fuel. This statement is at variance with the generally accepted notion that minimum fuel consumption always occurs at a much leaner mixture than that which results in maximum power.

The idea advanced in the paper that engines should be built to conform to a standard fuel is good. However, this should not be promoted to the extent that it would prevent the use of any large supply of good anti-knock fuel which might at some future time become available as a result of new processes or otherwise. The writer believes that the author of the paper kept this possibility in mind when he made the statement about building engines to suit fuels rather than fuels to suit engines.

Mr. Heron states the case in regard to vapor lock very well when he says the lowering of the 10 per cent point is desirable. Every effort should be made to so design fuel systems and carburetors that fuels of lower 10 per cent point can be used, since it will result in both larger supplies of fuel being available and in better anti-knock characteristics.

The author is certainly to be congratulated upon the comprehensiveness of the work which he has done on a problem which is extremely difficult. The writer feels sure that the many suggestions he has made will lead to fruitful results during the next few years.

MR. HERON. The data and remarks submitted by Dr. Becker are of decided interest. Examination of the engine test results show that, in comparison with those shown in Table 1, the average divergence in cubic centimeters of lead and in percentage of lead content is decidedly less for the temperature-plug determinations than it is for the bouncing-pin results.

Dr. Becker's argument in favor of the bouncing pin is of course sound so long as only gasoline containing lead is being purchased and is being tested against a standardized aromatic blend. It would seem that comparative tests of aromatic blends with lead added against California and cracked gasolines both containing lead by bouncing-pin and temperature-plug methods should be made.

The accumulation of carbon upon the temperature plug and its effect in raising its temperature applies also to the cylinder, piston, and exhaust valve when deposits accumulate on these portions of the engine. In any knock-test method of comparing against a standard, it would seem that the effects of such deposits should affect relative ratings much less than do engine speed and knock intensity.

Rather than use a potentiometer for recording the temperature of the plug in place of a millivoltmeter, it would appear that a plug design or test conditions showing greater sensitivity should be aimed at. The visual indication of temperature changes on a millivoltmeter when switching from fuel to fuel or when the engine condition is unsteady and unsuitable for carrying out knock tests

[16] Fuel and Lubrication Section, Research Laboratories, Standard Oil Development Company, Elizabeth, N. J.

is helpful to the operator and is far less trouble than the continual balancing of a potentiometer.

The Matériel Division has no particular ax to grind in favor of the temperature plug and is at present using the knock meter and bouncing pin on the Series 30 Ethyl Gasoline Engine and finds that this combination is much the most rapid and convenient method for rating fuels of which it has any experience.

It may be said for the temperature method:

(*a*) That is measures the tendency of a fuel to heat indirectly

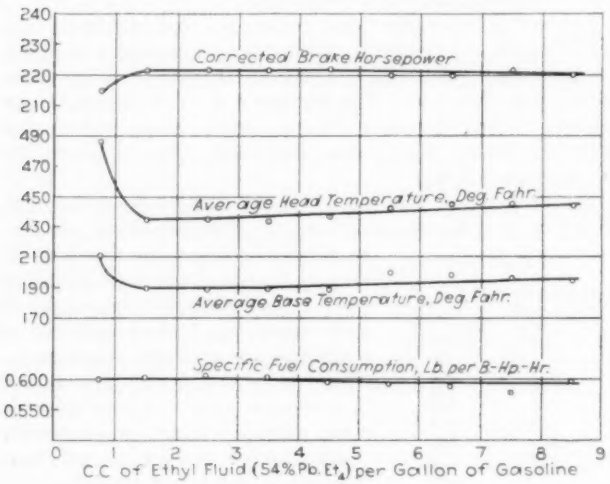


FIG. 4

cooled surfaces within the cylinder unit. In the case of an aircraft engine it is the overheating of indirectly cooled parts of the cylinder unit, such as pistons, spark plugs, and exhaust valves, resulting from the use of fuel too low in anti-knock value which causes failure. Incidentally, the relation of the temperature-plug readings to the readings of two other sets of thermocouples in the cylinder head of the Ethyl Gasoline Series 30 Engine has been checked. One couple was almost in the exhaust-valve seat and the other right below the bouncing pin. Under condition of constant spark advance and throttle opening, the relation between the readings of cylinder-head temperature and plug temperature was approximately constant. However, change of spark advance changed the relation between plug and head temperature; i.e., at different spark advances and constant plug temperature the cylinder-head temperature varied somewhat.

(*b*) That it is exceedingly simple and very reliable.

(*c*) That it provides a means of measuring at least the relative heating effect of a fuel at engine speeds in excess of those at which the bouncing pin will operate satisfactorily.

(*d*) At 900 r.p.m., its sensitivity to small lead or benzol concentration changes is at least equal to that of the bouncing pin at any engine speed yet used by the Matériel Division on the Series 30 Ethyl Gasoline Engine.

The Matériel Division has rated a number of straight-run California gasolines in terms of lead requirement to equal a standard California gasoline plus 3 cc. of lead. Ratings have been made at 900 r.p.m. and 350 deg. fahr. jacket temperature with the temperature plug and these checked against ratings made at 600 r.p.m. and 300 deg. fahr. jacket temperature with the bouncing pin. The agreement is surprisingly close and mostly within the limits of reproducability with either of the test methods.

With reference to carrying out variable compression engine tests at different jacket temperatures, it would appear desirable for aircraft fuels that:

(*a*) The engine used should have a cylinder closely approximating that of the most advanced type of liquid-cooled aircraft engine.

(*b*) That the tests should be carried at an engine speed of the order of that of modern aircraft engines at full power; i.e., 2000 r.p.m. or over.

In lieu of a variable compression engine, the use of a modern aircraft-engine cylinder with a variable amount of supercharging might well be considered. There is good ground for this suggestion as the trend with modern aircraft engines is rather to use up a given improvement in fuel by supercharging than by any considerable increase of compression ratio.

With regard to obtaining minimum fuel consumption, maximum power, and minimum cylinder temperature simultaneously when sufficient anti-knock compound has been added to the fuel under test, the minimum fuel consumption referred to in the paper is the fuel consumption giving 1 per cent less than maximum power (best setting), and once sufficient anti-knock compound has been added to the base fuel, this consumption is not reduced by further additions of anti-knock compound. That "best setting" consumption was referred to, was so stated in the paper—possibly not clearly, however.

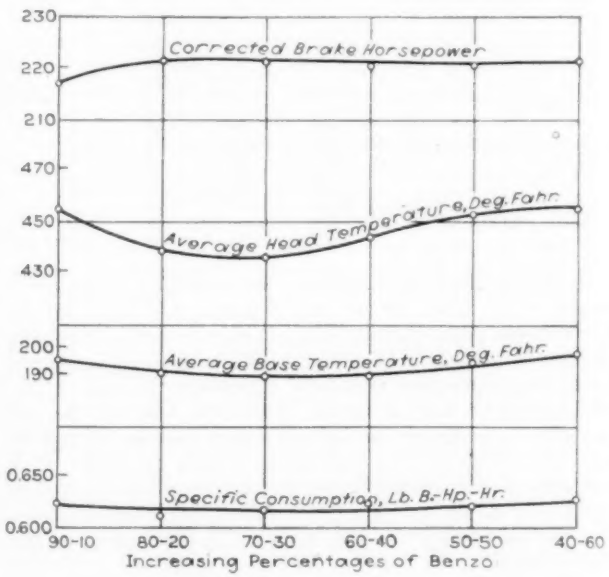


FIG. 5

Typical results of the effects of variation of lead and motor benzol concentration in average Mid-Continent D.A.G. upon power and cylinder temperature at "full rich" (not best setting) carburetor setting are shown in Figs. 4 and 5. These tests were carried out with a Wright J-5, 9-cylinder, air-cooled engine of 790 cu. in. cylinder capacity at 1800 r.p.m. and with 6.5 to 1 compression ratio. No particularly good data are available in support of the statement that the "best setting" consumption falls as the anti-knock concentration is progressively increased from the point where the fuel is definitely lacking in sufficient anti-knock to that necessary to secure maximum power with minimum fuel consumption.

When the fuel is deficient in anti-knock value, to be able to run at full throttle the air-to-fuel ratio has to be deceased to secure elimination of detonation. As the anti-knock value is increased, the mixture ratio for full-throttle running can be increased and the fuel consmption thus decreased. It would appear obvious that, as the fuel consumption for full power without detonation is decreased, the consumption for 1 per cent less than full power will be similarly decreased.

While it is Air Corps policy that engines shall be built to suit standard fuels rather than to supply each engine type with an appropriate fuel, nevertheless, as Dr. Becker suggests, the fuel standard will be raised as often as developments make this possible, and engine output will of course keep step with the fuel improvement.

While lowering of the 10 per cent point is of course desirable, the Air Corps is at present having such trouble with vapor lock as to render it desirable to specify a Reid vapor-pressure requirement on all gasoline for its use. The substituting of a Reid vapor-pressure test for the lower temperature limit of the 10 per cent evaporated point is necessitated by the fact that some gasoline deliveries apparently contain propane. For the present, pending improvement in fuel systems, it appears that a Reid vapor pressure of 6.5 lb. per sq. in. at 100 deg. fahr. cannot be exceeded.

ROBERT E. WILSON.[17] It is very fortunate for both engine manufacturers and users and those who supply its fuel and lubricants that an able and unbiased body like the Air Service is effectively prosecuting such important lines of research. The writer concurs heartily in Mr. Heron's statement that engines should be built to give good performance on certain standard fuels, and also his conclusion that in the light of present knowledge we must use tetraethyl lead in substantial quantities if we are to get real super-compression fuels. Making the standard fuel equivalent in anti-knock to 3 cc. of lead in California Fighting Grade gasoline certainly gives such a fuel—one which will make possible such performance and economy that engine manufacturers will have an adequate incentive to use metals which will withstand any danger of corrosion by Ethyl fluid or its products of combustion. Automobile engines give no trouble from this source, and there is no reason why aviation engines cannot be made equally satisfactory.

The Army Air Service has fostered two other things which will be of great assistance in making super-compression fuels generally available at a reasonable price. One is that the use of high-temperature liquid cooling by products such as glycol should greatly increase the ease of avoiding knocking with straight hydrocarbon fuels, most of which tend to lose their anti-knock value rather rapidly at the very high temperatures encountered in air-cooled engines. It must be remembered that even a very small local hot spot in an engine (for example, the exhaust valve) will greatly increase the tendency of a given fuel to knock.

The other helpful factor is the recommendation of a stability test instead of an inaccurate and over-strict copper-dish test for gum. The existence of the latter specification has made it practically impossible to use any cracked product in aviation fuels, in spite of the fact that cracked products generally have better anti-knock properties, and can be refined to a point where they give no actual trouble from gum. Our own company developed such a stability test several years ago, and has found the results obtained therewith a reliable guide to the behavior of a fuel in storage and service.

MR. HERON. While, as Dr. Wilson suggests, the avoidance of local hot spots is easier with high-temperature liquid-cooled engines than with air-cooled engines, nevertheless the best of the air-cooled engines have exhaust-valve cooling that is at least good as the best of the available water-cooled engines. As far as the existing data go, they would suggest that the glycol-cooled engine will give greater mean effective pressure on a given fuel than will the air-cooled engine. However, the comparison is not very exact, as it is based on an air-cooled engine with 195 cu. in. cylinders, having very excellent exhaust-valve cooling and running at 1900 r.p.m., and liquid-cooled engines with 96 and 131 cu. in. cylinders running at 2300 and 2400 r.p.m., both having less efficient exhaust-valve cooling than the air-cooled engine.

Dr. Wilson's views on cracked fuels for aviation and the accelerated aging test for gum are of decided interest, coming as they do from such a wealth of experience. The Air Corps' experience of cracked gasolines is as yet slight. In addition to the flight tests described in the paper, it is now receiving a fair volume of gum-stable cracked undoped D.A.G. to its Specification No. 2-40-F and has had no trouble to date with this fuel.

The lead response of cracked gasolines tested by the Air Corps has so far proved disappointing, despite the very high anti-knock value of some of these gasolines in the undoped condition.

The Air Corps wishes to encourage the use of cracked fuels of high anti-knock value, as it is only by their use on a fairly wide scale that their suitability for aircraft use can be determined.

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AER-52-31

Aluminum Alloys for Aircraft-Engine Piston and Cylinder Heads[1]

By A. J. LYON,[2] DAYTON, OHIO

The paper contains results of a great amount of valuable research on aluminum alloys for aircraft-engine piston and cylinder heads that has not heretofore been covered, and the findings should be valuable to all aircraft-engine designers. Two alloys have heretofore been recommended, but the Materiel Division at Wright Field has developed a series of new alloys, and the paper describes the method of manufacture, the effect of numerous alloying constituents on their properties, the strength at elevated temperature, their stability, and the methods of heat treatment, with comparison with the present standard alloys.

ALUMINUM-ALLOY PISTONS for automobile engines were first used in Belgium and France as early as 1912. Shortly after the World War, pistons were made in the United States for automobile engines by the permanent-mold process which involves casting in metal molds with removable iron cores. The alloys used for this purpose were mostly of the 10 per cent copper-aluminum type. With the development of heat-treating processes, small amounts of magnesium and iron were made to the foregoing alloy, in which form it is known as S.A.E. No. 34, or Lynite 122. There are also compositions containing nickel, manganese, and various other alloying constituents, the most important being the aluminum-copper-magnesium-nickel alloy "Magnalite."

The complicated designs and size of many aircraft-engine pistons and practically all cylinder heads make it impractical to use the permanent-mold process for the casting of these parts. Dry- or green-sand castings are generally used for air-cooled cylinder heads, while pistons are made either in green-sand molds or in semi-permanent molds, the latter being a metal mold with a sand core. Aluminum alloys cast in sand have a lower strength and hardness than when cast by the permanent-mold process, so that materials satisfactory for use in automobile-engine pistons are in many cases not suitable for aircraft engines. The U. S. Army Air Corps has directed considerable attention to the study of properties of piston and air-cooled cylinder-head materials, and it specifies the two following alloys in the heat-treated condition for this purpose:[3]

Alloy Lynite 122, or S.A.E. No. 34—Copper, 10 per cent; magnesium, 0.25 to 0.50 per cent; iron, 1.25 to 1.50 per cent; aluminum plus impurities, remainder.

Alloy Y, or Magnalite—Copper, 4 per cent; nickel, 2 per cent; magnesium, 1.50 per cent; aluminum plus impurities, remainder.

Neither of these materials is satisfactory in all respects. The aluminum-copper-magnesium alloy is especially difficult to cast sound in the form of the heavy sections that often are encountered in air-cooled cylinder heads, and the aluminum-copper-iron-magnesium alloy is unsatisfactory on account of its relatively high specific gravity, low strength when cast in heavy sections, and the softening of the alloy due to exposure to the operating temperatures of aircraft engines. The Matériel Division at Wright Field has developed a series of alloys which have a more desirable combination of properties than these alloys listed, and this paper describes the method of manufacture, effect of numerous alloying constituents on their properties, strength at elevated temperature, their stability, and methods of heat treatment, with comparison with the present standard alloys.

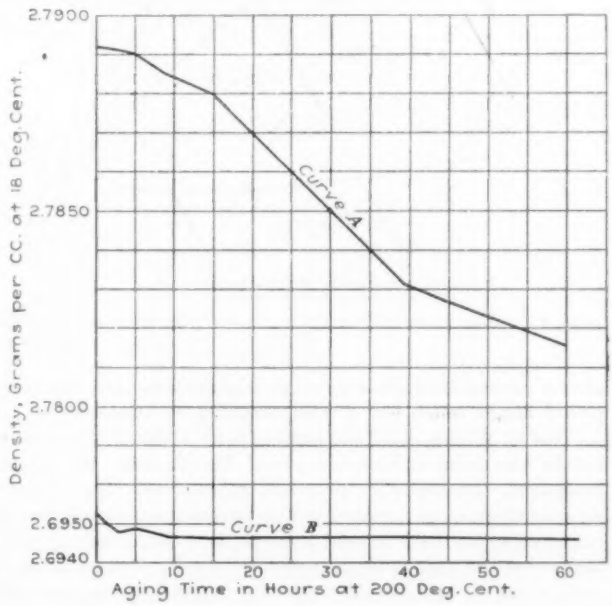


FIG. 1 EFFECT OF AGING AT 200 DEG. CENT. ON DENSITY OF COPPER-ALUMINUM ALLOY AND ON MAGNESIUM-SILICON-ALUMINUM ALLOY
(Curve *A*, 4.5 per cent copper-aluminum alloy, quenched at 500 deg. cent.; curve *B*, 0.58 per cent magnesium, 0.4 per cent silicon and aluminum alloy, quenched at 500 deg. cent.)

THE HIGH-STRENGTH ALUMINUM ALLOYS

The foregoing alloys can be classified as high-strength aluminum-casting alloys on a basis of tensile strength and hardness. The strength is in the range of 30,000 to 40,000 lb., and the

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[1] The investigations for this work have been made at Government expense and for Government purposes in the Matériel Division of the Air Corps, War Department.

[2] First Lieutenant, Matériel Division, Air Corps, Wright Field.

[3] "Materials for Aircraft Parts Subjected to High Temperatures," by J. B. Johnson, a paper read before a meeting of The American Society of Mechanical Engineers, December 5, 1928.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Brinell hardness is in excess of 100. The ductility is lower than the alloys commercially used in the general run of sand castings. Alloys of this later class have tensile strengths ranging from 18,000 to 25,000 lb. per sq. in., and their hardness is less than 90. The high-strength aluminum-casting alloys must be subjected to a heat-treating process which involves soaking at an elevated temperature, quenching, and aging.

Either copper or magnesium, with sometimes both constituents, is used in all the high-strength aluminum alloys. Copper forms a compound $CuAl_2$ which has a maximum solid solubility in aluminum of approximately 5 per cent copper at the eutectic melting point (540 deg. cent.). At 500 deg. cent. the solid solubility is reduced to 4.5 per cent copper, and at 250 deg. cent. it is less than 3 per cent. Magnesium combines with the silicon, always present in aluminum as an impurity, to form the compound Mg_2Si which has a solid solubility in aluminum of 1.6 per cent at 580 deg. cent., the melting point of the eutectic formed by the aluminum and the compound Mg_2Si. Like the copper compound, the solubility decreases rapidly with temperature. By quenching from a relatively high temperature, the hardening constituents are held in solid solution, and tests made immediately after the quenching operation show improvement in hardness, strength, and ductility over the material in the cast condition. Upon aging the quenched alloys, either at atmospheric temperatures or at higher temperatures, the compounds $CuAl_2$ and Mg_2Si act alike and are precipitated in the form of finely divided particles which are accompanied by still further increase in strength and hardness and by a decrease in elongation. In the commercial heat-treated aluminum-casting alloys, copper is used either alone as a hardening constituent or with magnesium. When both elements are present in the alloy, the solubility of the copper compound is reduced at both high and low temperature; likewise, the solubility of magnesium silicide is less in the presence of copper than it is in pure aluminum. In magnesium silicide alloys, magnesium, in addition to the amount required to unite with the silicon to form the compound Mg_2Si, reduces the solubility at both high and low temperatures, and in the Cu-Al alloys excess copper has the same effect. There are certain other alloying constituents which affect the solid solubility of these hardening compounds in aluminum, the most important of which, from a commercial standpoint, is nickel, which has little effect on the hardening effect of magnesium silicide, but increases the solubility of the copper compound both at high and low temperatures, and as a result reduces its hardening effect. Gayler and Preston ("The Age Hardening of Some Aluminum Alloys," Journal of the Institute of Metals, Vol. 41) studied the effect of the precipitation of the compounds $CuAl_2$ and Mg_2Si upon the hardness and density of several alloys containing copper and magnesium. The graphs in Fig. 1, taken from the data published by the foregoing authors, indicate that alloys which are hardened by the precipitation of magnesium silicide (curve *B*) undergo a less volume of change than those hardened by the precipitation of the copper compound (curve *A*). This is of importance from the standpoint of stability of dimensions of parts exposed to elevated temperatures in use. Curve *B* for the magnesium silicide alloy shows no decrease in density after five hours at a temperature of 250 deg. cent., while curve *A* for the aluminum-copper alloy shows that the density of volume changes during a period of fifty hours' exposure to the same temperature. It would follow that parts cast in the copper alloy should be annealed for prolonged periods in order to insure stability of dimensions, while an alloy which is hardened by the precipitation of magnesium silicide can be subjected for a much shorter period and thereby rendered immune to further "growth." The charts shown in Fig. 2 have been taken from the work of Bingham and Haughton on the age hardening of aluminum-copper-nickel-magnesium alloys, "Alloying Constituents and Age Hardening of Some Ternary and Quaternary Alloys." The Journal of the Institute of Metals, No. 2, 1926, Vol. 36, p. 144, shows that 2 per cent nickel in the alloy containing 4 per cent copper completely eliminates the precipitation of the compound $CuAl_2$, while the 10 per cent copper magnesium-aluminum alloy owes its increased hardness and strength upon aging to the precipitation of both the compounds Mg_2Si and $CuAl_2$.

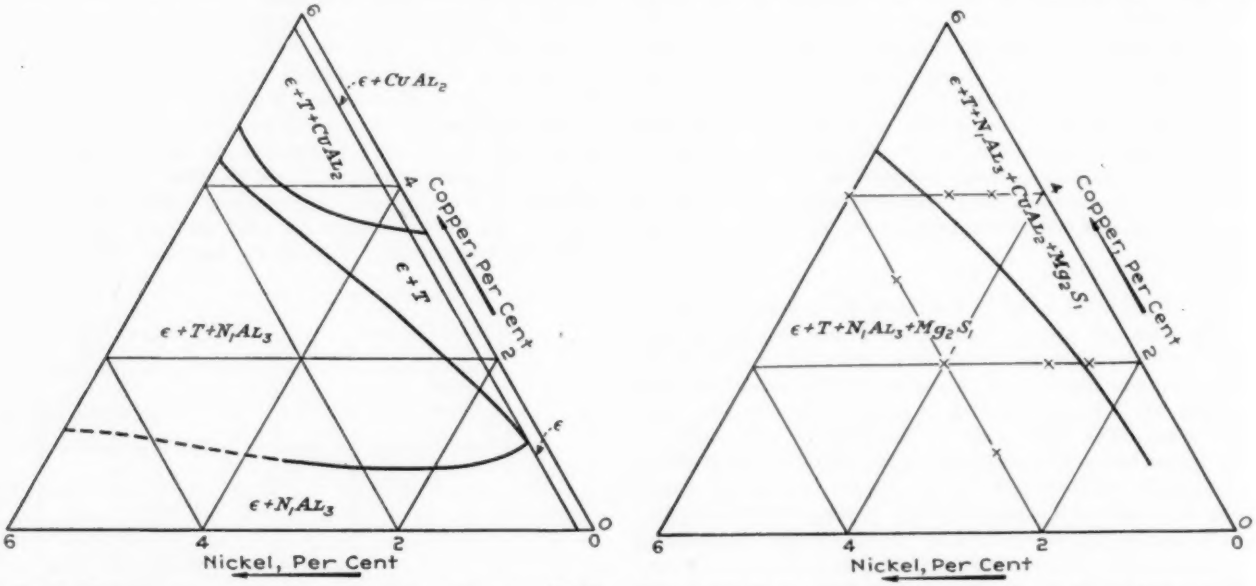


FIG. 2 CONSTITUTION AT 500 DEG. CENT. (FOR *A*) AND AT 200 DEG. CENT. (FOR *B*) OF A COPPER-NICKEL-ALUMINUM ALLOY CONTAINING 1 PER CENT MAGNESIUM

EFFECT OF SMALL AMOUNTS OF NICKEL ON THE PROPERTIES OF THE ALUMINUM-COPPER MAGNESIUM ALLOYS

A high-strength alloy developed by the Material Division for parts requiring a good combination of strength and elongation

has been described by Daniels, Lyon, and Johnson in the Transactions of the American Institute of Mining and Metallurgical Engineers, 1925, Vol. 71, p. 864. This alloy theoretically contains copper magnesium and silicon in the proportions that produce the maximum hardening and strengthening effect upon heat treatment. Test specimens cast in green sand, conforming to the dimensions of the American Society for Testing Materials, quenched at 510 deg. cent. and aged at 150 deg. cent. for 24 hours, gave the following physical properties:

Tensile strength, lb. per sq. in.	32,300
Elongation, per cent in 2 in.	3.2
Brinell hardness, 500 kg., 10 mm.	74

The composition of the foregoing alloy was: Cu 3.0, Mg 0.5, Si 0.3, Fe 1.2, Al remainder. Increasing the silicon content to 0.6 per cent to allow for silicon in the iron compounds, the following average results were obtained:

Tensile strength, lb. per sq. in.	41,370
Elongation, per cent in 2 in.	0.5
Brinell hardness, 500 kg., 10 mm.	101

It was concluded that an alloy of this type could be used for pistons and cylinder heads with the advantages of a lower specific gravity than the 10 per cent copper-magnesium alloy and with better casting properties than the 4 per cent copper and 2 per cent nickel-magnesium alloy. Small amounts of nickel were added for the purpose of determining the effect of this element upon the physical properties. The following results were obtained on standard test specimens soaked at 525 deg. cent. for 5 hours, quenched in cold water, and aged at 150 deg. cent. for 24 hours:

Nickel,[1] per cent	Tensile strength, lb. per sq. in.	Elongation, per cent in 2 in.	Brinell hardness
Nil	42,690	0.5	105
0.25	42,590	0.5	109
0.50	42,450	0.5	111

[1] Cu 3.0, Fe 1.2, Mg 0.5, Si 0.6, Al remainder.

The physical properties are not affected by small additions of nickel where the alloys are quenched from a temperature below 530 deg. cent., but as little as 0.5 per cent nickel is shown in Fig. 3 to have a pronounced effect upon the strength and hardness of the heat-treated alloys when quenched from higher temperatures. It was also observed that these alloys could be heated to a higher temperature without danger of "burning" than those of the same composition except for the nickel content. Another possible advantage of the nickel is to lessen "growth" by preventing the precipitation of the compound $CuAl_2$. The charts of Fig. 2 show that 0.5 per cent nickel does not quite place the alloy containing 3 per cent copper in the field where there is no precipitation of the copper compound when aged or annealed at 200 deg. cent. However, hardness tests made on forged alloys with the nickel content ranging from 0.2 to 2.0 per cent by Bingham ("The Constitution and Age-Hardening of Some Ternary and Quaternary Alloys," Journal of the Institute of Metals, 1926, Vol. 36), show that from the practical standpoint 0.4 per cent nickel suppresses the age hardening of the copper-aluminum alloys. It was found in this investigation that 0.5 per cent nickel is sufficient to lessen "growth" caused by the precipitation of the compound $CuAl_2$, but not enough to appreciably affect the casting properties of the alloy, as described under another heading.

Iron and Silicon in the Aluminum-Copper-Nickel-Magnesium Alloys

Iron and silicon occur as impurities in all grades of aluminum ingot used in the manufacture of aluminum alloys. Where grade 1 primary aluminum is used, the iron and silicon from this source do not exceed 0.5 per cent. However, in the remelting of the alloys in connection with the manufacture of casting, additional iron is picked up from ladles, stirring rods, and melting pots. There is a conflict of opinion on the beneficial effects of added iron in excess of that which occurs in the form of an impurity, but it is generally conceded that it has a beneficial effect on the strength of alloys at elevated temperatures and in the casting of parts that are subject to cracking during solidification. In the high-strength aluminum alloys, containing copper and magnesium, neither iron nor silicon appreciably affects the solid solubility of the hardening constituents

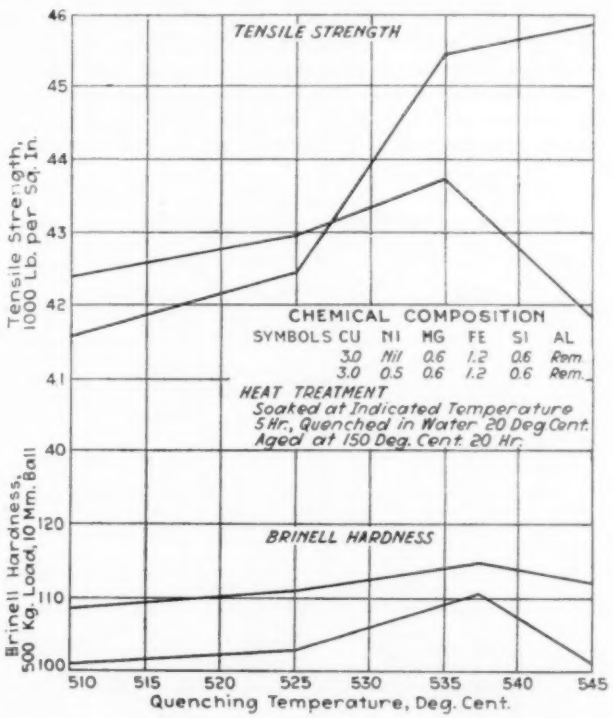


Fig. 3 Effect of Nickel on the Strength and Hardness of Aluminum-Copper-Magnesium Alloys

($CuAl_2$ and Mg_2Si) and therefore does not affect the age-hardening of these alloys in the same manner as does nickel or excess magnesium. Iron forms the compound $FeAl_3$ with aluminum which has a limited solubility and forms a eutectic with aluminum at about 1.7 per cent iron. Silicon is soluble in aluminum in the solid state up to approximately 1 per cent. Iron in the presence of silicon forms a number of compounds and series of solid solutions which appear in the microstructure of aluminum alloys in various crystal forms.

Gwyer and Phillips ("The Constitution of Alloys of Aluminum With Silicon and Iron," Journal of the Institute of Metals, 1927, Vol. 38) have described three constituents that are present in aluminum-iron-silicon alloys as cast and annealed at 500 deg. cent.: $FeAl_3$, which occurs in the as-cast alloys in the form of elongated masses of needles and plates; Beta, as a constituent which resembles Chinese script and which is a solid solution of iron and silicon; *X* constituent, which occurs in the form of needles and contains various proportions of iron and silicon in solid solutions. The latter two constituents are formed as a result of a reaction between the compound $FeAl_3$ silicon and aluminum during the freezing of the alloys and to a certain extent by prolonged annealing of the solid alloys at high tempera-

TABLE 1 PHYSICAL PROPERTIES OF ALUMINUM-COPPER-NICKEL-MAGNESIUM ALLOYS WITH AND WITHOUT ADDED IRON

Quenching temperature,[1] deg. cent.	Tensile strength, lb. per sq. in. Fe 0.34[2]	Tensile strength, lb. per sq. in. 1.65[3]	Elongation, per cent in 2 in. 0.34	Elongation, per cent in 2 in. 1.65	Brinell hardness 500 kg., 10 mm. 0.34	Brinell hardness 500 kg., 10 mm. 1.65
As-cast	19,910	26,720	1.5	3.0	57	66
510	36,810	45,690	0.5	0.7	116	110
516	41,740	44,770	0.5	0.5	115	106
538	45,000	...	1.0	...	119	...
549	34,300	44,800	0.5	1.0	113	105
560	11,740	48,450	0.5	1.3	105	111

[1] Sand cast; aged 16 hr. at 150 deg. cent.
[2] Cu 3.0, Ni 0.52, Si 0.49, Mg 0.56.
[3] Cu 3.1, Ni 0.55, Si 0.52, Mg 0.54.

tures. E. H. Dix, of the Aluminum Company of America, also has furnished valuable data on the occurrence of iron and silicon in aluminum, and in general his findings are in agreement with those of the English investigators. Geyer, Phillips, and Mann, ("Alloy of Aluminum With Copper, Silicon, and Iron," Journal of the Institute of Metals, 1928, Vol. 40, p. 346) also have shown the presence of a quaternary eutectic formed by $CuAl_2$, the constituents Beta and X, primary silicon, and aluminum.

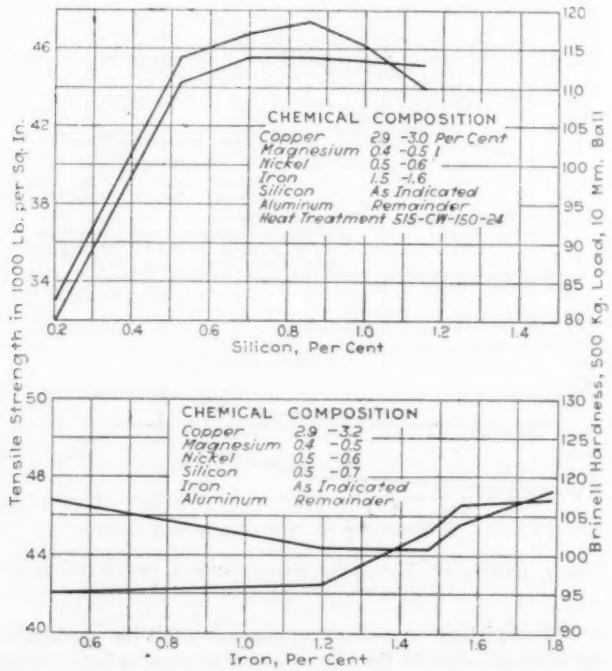


FIG. 4 EFFECT OF IRON AND SILICON ON THE TENSILE STRENGTH AND HARDNESS OF ALUMINUM-COPPER-MAGNESIUM ALLOY, QUENCHED FROM A TEMPERATURE OF 525 DEG. CENT.

The melting point of this eutectic is approximately 520 deg. cent. and is important in connection with the heat treatment of all high-strength aluminum alloys containing silicon, iron, and copper.

Archer and Jeffries ("New Developments in High-Strength Aluminum Alloys," Transactions of the American Institute of Mining and Metallurgical Engineers, 1925, Vol. 71, p. 839) have shown the effect of iron in a wrought aluminum-copper alloy to be harmful, but with the addition of approximately 1 per cent silicon to a 4 per cent copper-aluminum alloy containing 1 per cent iron, the tensile strength was increased 35 per cent without greatly affecting the elongation.

A study of the effect of iron and silicon in the cast aluminum-copper-nickel-magnesium alloys was made in connection with this investigation, and the results are shown in Figs. 4 and 5 and in Table 1. The best combination of strength and hardness of this type of alloy is developed with 0.6 to 0.8 per cent silicon, with an iron content of approximately 1.5 per cent. The results given in Table 1 are for two alloys of the same composition except for the iron content and show that the use of iron in this type of alloy is desirable.

In alloys containing 1.5 per cent iron the temperature to which the alloys can be heated without "burning" and rendering the material unfit for use is rapidly reduced with increasing silicon, as shown by the graphs in Fig. 5 and the results given in Table 2.

From the results of the foregoing experiments two composi-

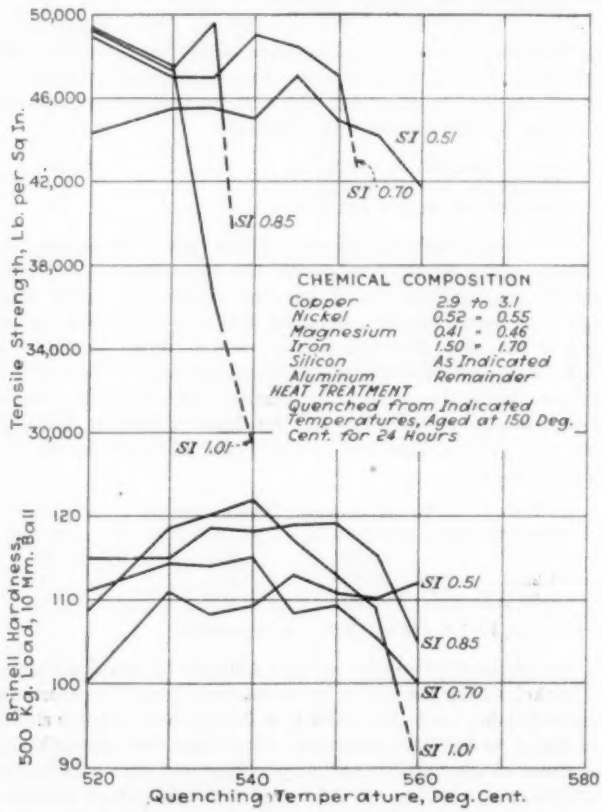


FIG. 5 TENSILE STRENGTH AND HARDNESS OF 3 PER CENT COPPER-MAGNESIUM-NICKEL-ALUMINUM ALLOYS, SHOWING THE EFFECT OF SILICON ON THEIR HEAT TREATMENT

tions were selected representing the upper and lower practical limits for silicon in the aluminum-copper-nickel-magnesium alloys for use in pistons and cylinder heads. For the purpose of identification, these alloys have been given the symbols X and Z, and their chemical compositions follow:

	Cu	Ni	Fe	Si	Mg	Al
Alloy X	3.0	0.5	1.5	0.6	0.5	Rem.
Alloy Z	3.0	0.5	1.5	1.0	0.5	

TABLE 2 TENSILE STRENGTH AND BRINELL HARDNESS OF ALUMINUM-COPPER-NICKEL-MAGNESIUM ALLOYS, SAND-CAST AND HEAT-TREATED

Quenching temperature[1], deg. cent.	Tensile strength, lb. per sq. in. Si, % 0.53[2]	0.70	0.86	1.01	Brinell hardness 500 kg., 10 mm. 0.53	0.70	0.86	1.01
515	41,020	49,270	49,370	49,000	100	111	115	108
526	45,610	46,900	4,710	46,180	111	114	114	118
532	46,600	47,030	49,760	37,280	108	114	118	120
538	46,080	49,400	28,340	20,000	109	115	117	122
543	47,310	48,570	21,300	16,400	113	108	119	117
549	45,060	47,570	20,330	19,950	111	109	119	113
555	44,120	19,540	19,520	19,360	110	105	115	110
560	41,740	12,440	5,287	3,992	112	97	106	89

[1] Aged 24 hr. at 150 deg. cent.
[2] Cu 2.90–3.08, Fe 1.47–1.73, Ni 0.52–0.55, Mg 0.41–0.48, Al remainder.

Effect of Copper

For the purpose of determing the limits of the copper in this type of an alloy, the copper content was varied in the range 3 to 5 per cent with the following results for sand-cast specimens quenched from 525 deg. cent. and aged for sixteen hours at 150 deg. cent.:

Copper[1] per cent	Tensile strength, lb. per sq. in.	Brinell hardness, 500 kg., 10 mm.
3	46,970	116
4	43,740	113
5	44,340	122

[1] Ni 0.5, Mg 0.5, Fe 1.8, Si 0.5, Al remainder.

The foregoing results show that the physical properties at atmospheric temperatures are not appreciably affected by the variations in the copper content. The heat treatment and the properties resulting therefrom of the 5 per cent copper alloy are affected by iron, silicon, and nickel in the same manner as the 3 per cent copper alloy, as shown in Fig. 6. An alloy of the following compositions was chosen from this series for the purpose of investigating its properties:

Alloy symbol	Chemical composition
W	Cu 5.0 per cent
	Ni 0.5 per cent
	Mg 0.5 per cent
	Fe 1.5 per cent
	Si 0.6 per cent
	Al Rem.

Effect of Magnesium

Two heats were made of exactly the same composition and containing iron, nickel, copper, and silicon in amounts corresponding to the specifications of experimental alloy X. Magnesium (0.5 per cent) was added to one of the heats, while none was added to the other. Standard cast-to-size test specimens were cast in green sand, then heat-treated. One lot of specimens from the foregoing composition was tested after aging at room temperature for 24 hr. and another lot after aging at 150 deg. cent. for the same length of time. The results given in Table 3 show that the specimens aged at room temperature are of the same order as those obtained by Hyman (Journal of the Institute of Metals, 1925, Vol. 34), who recommends an alloy of similar composition to alloy X for general purposes where combinations of high strength and ductility are required in castings. The results given in Table 3 for the specimens aged at 150 deg. cent. show a tensile strength 20 per cent higher than ordinarily obtained on Y alloy and Lynite 122 and show conclusively that the hardening and strengthening are due almost entirely to the precipitation of the compound magnesium-silicide. Small variations in the magnesium content can be expected in the mixing of the alloys, and the effect on the strength and hardness is shown in the following tabulation:

Magnesium,[1] per cent	Tensile strength, lb. per sq. in.	Brinell hardness, 500 kg., 10 mm.
0.4	47,220	107
0.5	46,715	108
0.6	45,880	109

[1] Cu 3.0, Ni 0.55, Fe 1.6, Si 0.99, Al remainder.

The foregoing results were obtained on sand-cast specimens quenched from 515 deg. cent. and aged for 24 hours at 150 deg. cent.

Method of Alloying

Aluminum ingot for the manufacture of aluminum alloys may be obtained commercially in the following grades that conform to U. S. Army Specifications:

Grade	Chemical composition Al	Cu	Fe-Si	Mn and other impurities
Special	99.5	0.1	0.4	....
Grade A	99.0	...	0.8	0.25
Grade B	98.0	...	1.5	0.50

Grade A aluminum as used in the mixing of the high-strength alloys usually contains approximately 0.5 per cent of iron and 0.3 per cent or less of silicon. The relationship of the iron and silicon contents of alloys X, Z, and W has been shown to be of great importance, so in order to control the composition accurately, an intermediate alloy containing both elements is used in mixing the alloys. The compositions of these hardeners are as follows:

Alloy	Fe	Si	Mg	Al
X and W	8.0–10.0	4.0–6.0	5.0–6.0	Rem.
Z	8.0–10.0	6.0–8.0	5.0–6.0	Rem.

TABLE 3 PHYSICAL PROPERTIES OF ALUMINUM-COPPER-NICKEL-ALUMINUM ALLOYS, WITH AND WITHOUT MAGNESIUM

Condition	Tensile strength, lb. per sq. in. Mg = nil[1]	Mg = 0.5[1]	Elongation, per cent in 2 in. Mg = nil	Mg = 0.5	Brinell hardness, 500 kg., 10 mm. Mg = nil	Mg = 0.5
As-cast	22,563	27,813	5.8	1.7	51	74
HT-A	27,470	39,546	7.2	3.3	58	99
HT-B	28,727	40,373	8.7	3.5	58	96
HT-C	26,933	51,733	7.2	1.5	58	120
HT-D	28,043	51,846	8.2	1.5	57	118

[1] Cu = 3.1, Ni = 0.6, Fe = 1.7, Si = 0.6, Al remainder.
HT-A, 540 deg. cent., 5 hours' quench in cold water, aged 24 hours at 20 deg. cent.
HT-B, 540 deg. cent., 24 hours' quench in cold water, aged 24 hours at 20 deg. cent.
HT-C, 540 deg. cent., 5 hours' quench in cold water, aged 24 hours at 150 deg. cent.
HT-D, 540 deg. cent., 24 hours' quench in cold water, aged 24 hours at 150 deg. cent.

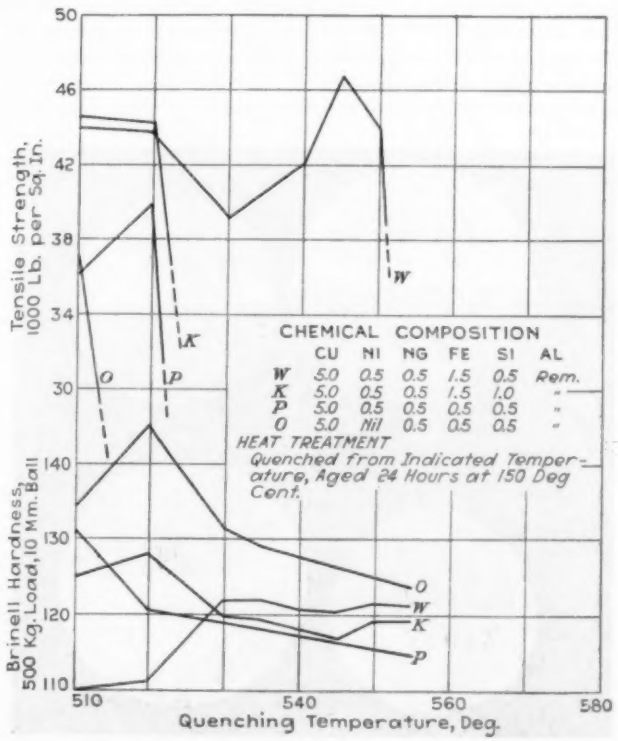


Fig. 6 Tensile Strength and Hardness of 5 Per Cent Copper-Magnesium-Aluminum Alloys and the Effect of Iron, Silicon, and Nickel on Their Heat Treatment

These hardeners can be mixed in crucible-type furnaces and the composition of the alloying elements be controlled within the foregoing limits. The variation in the final alloy is less than 0.1 per cent. The copper and nickel are introduced in the alloys in the form of a Cu-Ni-Al hardener containing the following proportions of copper and nickel:

Alloys X, Z, and W	Copper, 30 to 40 per cent Nickel, 5 to 8 per cent

In connection with the manufacture of Y alloy it has been stated that it is desirable to introduce the magnesium directly into the final alloy rather than to add it by means of a magnesium-rich alloy of aluminum, but in the manufacture of alloys, X Z, and W there was no difference in the casting properties of the alloys made by the direct addition of magnesium and those made by introducing the nickel with the iron and silicon. The iron-silicon-magnesium intermediate alloy breaks readily under the hammer for the purpose of weighing and when cast into iron molds is free from segregation. It has a relatively low melting point and readily goes into solution in aluminum at a temperature of 800 deg. cent. Experiments with several compositions of nickel, copper, magnesium, and aluminum proved that the addition of magnesium in this manner was unsatisfactory. Nickel with magnesium produces intermediate alloys that go into solution very slowly, and excessive dross is formed upon the surface of the molten metal.

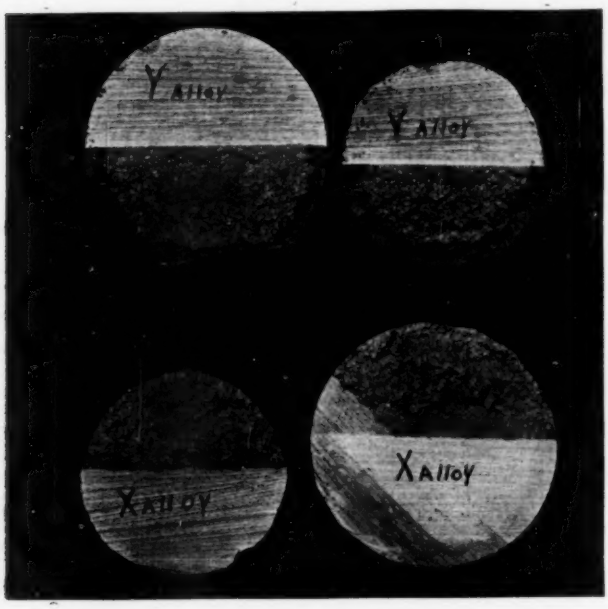


FIG. 7 PHOTOGRAPH SHOWING POROUS RIM NEAR SURFACE OF Y-ALLOY CASTINGS AND THE ABSENCE OF THIS DEFECT IN ALLOY X

CASTING PROPERTIES

The soundness of castings of alloy W is superior to the other two experimental alloys X and Z, but all are greatly superior to Y alloy. Cylinders 3 in. in diameter and 6 in. in length cast in green sand, as shown in Fig. 7, were poured at 675 deg. cent. from the following alloys:

	Composition					
Alloy symbol	Cu	Ni	Mg	Si	Fe	Al
Z	3.0	0.5	0.7	1.0	1.5	Rem.
X	3.0	0.5	0.5	0.7	1.5	Rem.
W	3.0	0.5	0.5	0.7	1.5	Rem.
122	10.0	Nil	0.3	0.2	1.3	Rem.
LCD	2.5	Nil	0.7	0.6	1.5	Rem.
Y	4.0	2.0	1.5	0.3	0.4	Rem.

From the surface of these cylinders, $^1/_8$ in. was machined, and the cylinder was light-etched in a solution of sodium hydroxide, and others were fractured and the appearances were compared. The surface of the cylinder cast in Y alloy contained numerous gas cavities of considerable depth and size. The alloy LCD was the best of the series, while alloys Z, X, W, and 122 were about of the same order, except that "pinholing" was more pronounced on the 122 cylinder. Fractures of specimens from alloy Y contained a porous, discolored rim approximately a $^1/_4$ in. deep, which contained numerous cavities. Alloys X, Z, and W show this defect to a much less degree, and it can be eliminated by the fluxing of the metal; and alloys 122 and LCD, which do not contain nickel, were free from it. Fig. 8 shows a comparison of the fractures of alloys Y and X. Chemical samples taken from the surface of the cylinders referred to and which were cast in Y alloy show the normal amount of nickel and magnesium, but the copper was 1 per cent less than the average sample, showing that segregation occurs in heavy cast sections.

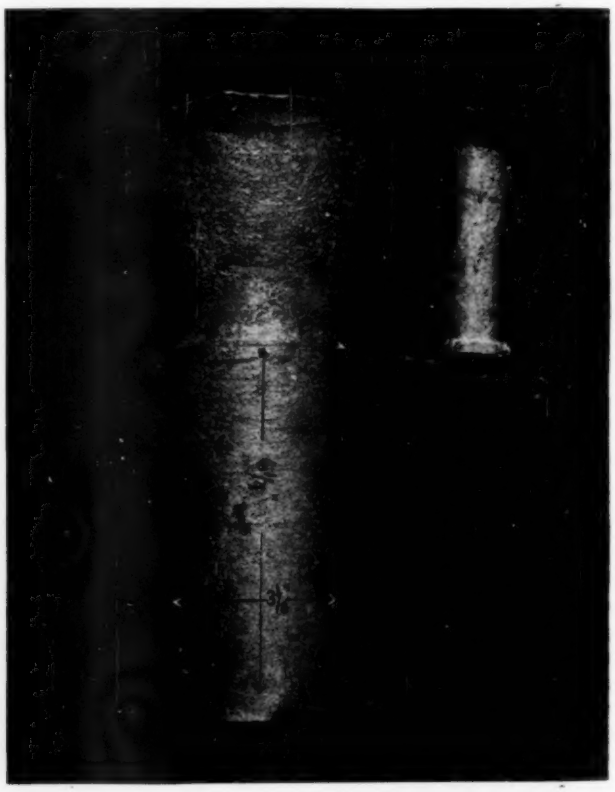


FIG. 8 PHOTOGRAPH SHOWING METHOD OF CASTING CYLINDER USED FOR COMPARING CASTING PROPERTIES (SEE FIG. 7) AND FOR GROWTH MEASUREMENTS (SEE FIG. 14)

A further comparison was obtained between the casting characteristics of alloys Z, X, W, and Y by substituting the former alloys in a piston of complicated design that had given trouble in the foundry and machine shop because of gas holes, cracks, and distortion. These alloys produced sound castings when cast with the same method of gating and chilling as Y alloy, except that alloy X required slightly heavier risers as the solidification shrinkage is higher and the pouring heads and risers "pipe" to considerable depth. Increasing the copper

TABLE 4 STABILITY OF PHYSICAL PROPERTIES OF PISTON ALLOYS WHEN REHEATED

Heat treatment	Tensile strength lb. per sq. in.			Elongation, per cent in 2 in.			Brinell hardness 500 kg., 10 mm.		
	X alloy	Y alloy	Lynite 122	X	Y	122	X	Y	122
Quenched	32,280	33,660	34,280	5.5	1.0	1.0	74	98	94
Quenched and aged	48,200	39,410	37,910	0.5	0.5	0.5	115	115	119
Reheated at 260° C., 1 hr	44,140	33,210	34,800	0.5	0.5	0.5	107	100	99
Reheated at 260° C., 5 hr	38,120	35,920	28,960	0.5	0.5	0.5	100	92	89
Reheated at 260° C., 25 hr	32,530	29,810	25,300	2.0	0.5	0.5	74	83	77

Composition as mixed	Cu	Fe	Si	Ni	Mg	Al
X	3.4	0.75	0.6	0.7	0.7	Dif.
Y	4.0	0.5	0.3	2.0	1.5	Dif.
Lynite 122	10.0	1.5	0.3	Nil.	0.25	Dif.

content of alloy X from 3.0 per cent to 3.5 per cent and the nickel content from 0.5 per cent to 0.7 per cent and decreasing the iron produced a marked improvement without affecting the physical properties, and castings molded in exactly the same manner as for Y alloy showed no evidence of "shrinks" or "draws." All castings in the experimental alloys were sound, as was shown by cutting the pistons into sections and by the use of X-ray analysis. The rejections due to imperfect castings alone averaged about 50 per cent in Y alloy for this particular piston, while in the experimental alloys the rejections due to foundry defects in the experimental alloys were practically nil.

Heat Treatment

The heat treatment of alloys X, Z, and W is similar to the process for Y alloy and Lynite 122 with the exception of the temperature to which the materials can be heated for quenching. The soaking period, or the time the material must be held at the quenching temperature in order to effect solution of the hardening constituents, is comparatively short, and no appreciable gain in the physical properties is obtained for periods over five hours, as is shown in Table 3. The results given in Table 2 and the graphs shown in Figs. 5 and 6 indicate that the maximum properties of the alloys containing 0.6 and 0.8 per cent silicon can be developed over a considerable range of temperature, which is a distinct advantage in commercial heat-treating practice. The maximum temperature to which Y alloy can be heated without "burning" is 530 deg. cent., and small variations in temperature have a very pronounced effect upon the physical properties, as shown by the following results:

Quenching temperature, deg. cent.	Tensile strength and Brinell hardness of alloys, lb. per sq. in.			
	122	Y	W[1]	X[1]
505	36,900–116	34,200–100	42,380–117	47,150–111
515	39,100–134	37,900–106	43,900–111	49,270–115
525	37,800–124	44,350–111	39,160–124	46,900–114
538	22,300–112	20,000–110	42,860–121	49,400–115
549	23,600–106	20,100–109	43,300–122	47,570–109
560	21,600–101	18,000–102	21,150–102	12,440– 97

[1] Silicon 0.70; maximum quenching temperature.

The aging or temperature to which the alloys must be reheated after quenching to develop the maximum hardness and strength can be accomplished at temperatures ranging from 150 to 260 deg. cent. Aging temperatures higher than 150 deg. cent. require a much shorter time to produce the maximum hardness and strength, as shown in the following tabulation for the alloy:[4]

Aging temperature, deg. cent.	Aging time, hr.	Tensile strength, lb. per sq. in.	Elongation, per cent in 2 in.	Brinell hardness
150	2	37,510	3.7	89
150	8	39,420	1.0	95
150	16	44,800	1.0	105
150	24	48,960	0.8	113
150	48	47,150	0.5	113
205	1/2	44,520	0.7	103
205	1	45,220	0.7	110
190	5	47,930	1.0	111
260	1/2	44,540	0.5	108

Aging treatments of 16 and 24 hours at 150 deg. cent. were used in the experimental work in connection with the development of these alloys unless specified otherwise. For commercial practice, however, an aging treatment of 5 hours at 190 deg. cent. is recommended for pistons. The quenching temperature should be in the range of 510 to 530 deg. cent., as shown.

In many of the present types of aircraft engines it is the practice to shrink the aluminum alloy head upon a steel cylinder barrel at a temperature of approximately 375 deg. cent. The following physical results obtained on sand-cast test specimens quenched at from 500 to 515 deg. cent. and reheated to 375 deg. cent. illustrate the superiority of X-type alloy for this purpose:

Alloy	Reheating temperature, deg. cent.	Time at temperature, hr.	Tensile strength, lb. per sq. in.	Elongation, per cent in 2 in.	Brinell hardness
X[1]	375	1/2	27,200	2.8	61
Y	375	1/2	30,042	0.8	78
122	375	1/2	22,100	0.9	61
Z[2]	375	*1/2	30,080	2.5	71

[1] Si 0.5 per cent. [2] Si 1.0 per cent.

Comparing the foregoing results with the chart shown in Fig. 13 and with Table 4, it may be seen that the alloys are in their stable condition and that little change should occur in the foregoing properties upon reheating to the operating temperatures of aircraft-engine cylinder heads for prolonged periods of time. These results further show that cylinder heads that are subjected to the shrinking operation during assembly should not be artificially aged. A further advantage of alloy X is that it can be straightened after the quenching operation in the same manner as castings made from the high-strength alloy commercially known as Lynite 195, while the ductility of alloys Y and Lynite 122 is too low in the as-quenched condition to correct for any distortion that occurs during the quenching operation.

Tensile and Hardness Tests

In all cast metals and alloys the rate of solidification has a pronounced effect upon the mechanical properties, but in alloys that tend to segregate when cast in heavy sections the strength drops off more rapidly than with alloys that do not. For the purpose of comparing the properties of alloys used for sand castings it is desirable to know the effect that mass and thickness of section have upon the physical properties, as the rate of cooling and of solidification of the molten metal decreases as the size of the casting increases. In Figs. 9 and 10 the tensile strength and the hardness of the experimental alloys, X, Z, and W are compared with the two standard alloys now used by the Army Air Corps (Lynite 122 and Y alloy). These graphs were prepared from results obtained on test specimens cast in green sand ranging from 1/2 in. in diameter to 1.5 in. in diameter over the reduced section. The methods of gating were the same in all cases and the pouring temperatures were held constant at 675 deg. cent. A series of specimens, with the exception of the Lynite 122 specimens, were soaked at 510 deg. cent. for 5 hours, quenched in water, and artifically aged for 24 hours at 150 deg. cent., and then tested. The Lynite 122 specimens were soaked

[4] Chemical composition: Cu 3.08, Ni 0.55, Fe 1.57, Si 0.69, Mg 0.54, Al remainder. Sand-cast standard test specimens were quenched from 520 deg. cent.

at 500 deg. cent. to avoid burning. Another series were cast and allowed to stand for 30 days at room temperatures and then tested.

The superiority of all experimental alloys when cast in heavy sections such as are encountered in pistons and cylinder heads is readily shown. In particular, it may be seen by comparing the experimental alloys with Lynite 122 and Y alloy that the $^1/_2$-in. sections show a difference in strength of about 10,000 lb. per sq. in., or 25 per cent, in favor of the former, while for the $1^1/_2$-in. specimen the tensile strengths of X, W, and Z alloys are 75 per cent higher than Lynite 122 and Y alloy.

For the specimens cast and aged for 30 days at atmospheric temperatures, the graphs in Fig. 9 show that alloys X and Z

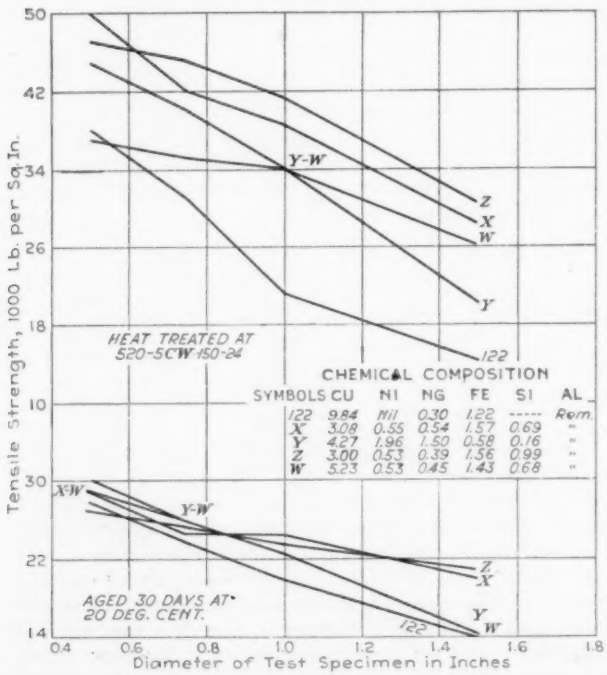


Fig. 9 Tensile Strength of Aluminum Piston and Cylinder-Head Alloys, Showing the Effect of Section Thickness

retain their strength in the larger sections better than the other three alloys, and it will be observed that there is a sharp break in the curve for Y alloy at a point corresponding to the 1-in. specimens. This is explained by the segregation of the copper and the formation of the rim of porous metal as shown in the photograph of Fig. 8. This condition is pronounced in the $1^1/_2$-in. specimens, and (as noted under another heading) is characteristic of heavy sections cast in all of the Al-Cu-Ni-Mg alloys. It is due to the occurrence of nickel and magnesium together, and lowering the magnesium content alone of Y alloy does not eliminate this porous rim of metal near the surface of all heavy sections.

The hardness results given in Fig. 10 show that the heat-treated experimental alloys are superior to the high-copper alloy, and in addition reveal an interesting phenomenon in the specimens cast from Y alloy. The hardness of the heat-treated specimens increases with increasing thickness and is explained by the manner in which this alloy solidifies. The equilibrium relations of the constituents in this type of alloy are very complex and involve the decomposition of constituents that separate from the partly solid alloy into other constituents at a lower temperature of the still partly solid alloy. Bingham and Haughton ("Constitution of Some Alloys of Aluminum with Copper and Nickel," Journal of the Institute of Metals, 1923, Vol. 29, p. 94) describes a constituent Tau (see Fig. 2) as a ternary compound or solid solution, containing copper, nickel, and aluminum, which is formed by a reaction between the compound $NiAl_3$ (solid) and the remaining liquid at a temperature of approximately 585 deg. cent. This constituent is very hard and has the property of dissolving both the compounds $NiAl_3$ and $CuAl_2$. The rate of solidification, therefore, would be expected to have a bearing on the amount of this constituent formed and on the final hardness of the alloy. It should be borne in mind that in dealing with cast alloys equilbrium conditions are more closely approximated under conditions of very slow cooling. From the practical standpoint this partly explains the variations of physical results that are often obtained on sand castings of Y alloy. The author further is of the opinion as a result of work on the stability of the properties of this alloy that $NiAl_3$, as well as Mg_2Si, plays an important part in the age-hardening of this type of alloy. There is evidence that mag-

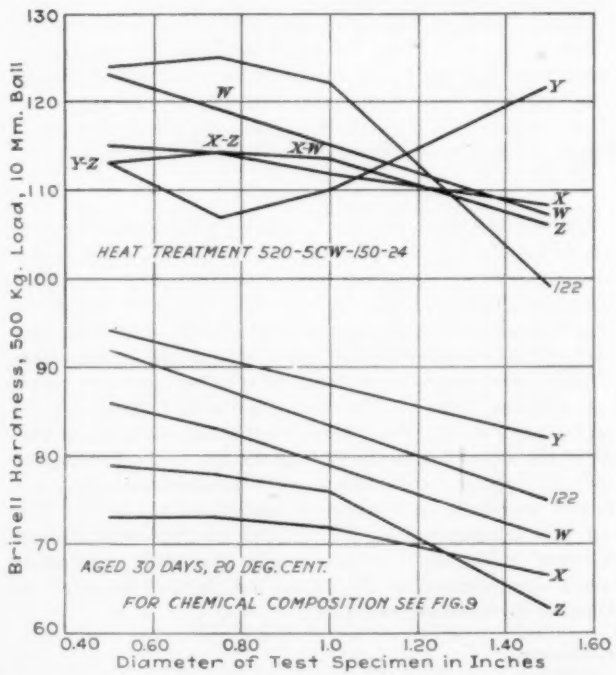


Fig. 10 Brinell Hardness of Aluminum Piston and Cylinder-Head Alloys, Showing the Effect of Section Thickness

nesium affects the solid solubility of the former constituent, and in an alloy of Y composition it is the principal age-hardening element rather than Mg_2Si.

Tension Test and Hardness Tests at High Temperatures

Tension tests were made on the experimental alloys X, Z, and W, and for the purpose of comparison, specimens cast from alloy Y and Lynite 122 were tested under the same conditions. The methods employed were the same as those used in the tests previously published for the two latter alloys by J. B. Johnson ("Materials for Aircraft Parts Subjected to High Temperatures," presented at the Annual Meeting, New York, N. Y., December 5 to 8, 1927, of The American Society of Mechanical Engineers.) These tests are of the "short-time type" as described by R. L. Templin and C. Braglio ("Temperatures Affect Aluminum Alloys," *The Foundry*, January 1, 1929). Templin and Braglio results for Y alloy and Lynite 122 in the "as-cast" condition are in agreement with the results obtained by the Air Corps and show

that Y alloy is slightly superior to Lynite 122 in the temperature range of 200 to 250 deg. cent. Templin and Braglio do not show results for Lynite 122 in the heat-treated condition, but Johnson has shown that the same superiority exists in the quenched and aged material. The results of the present tests shown in Fig. 11 indicate that at 260 deg. cent. and above alloys Y and X have the same tensile strength, but that the latter alloy has the advantage of a higher strength below 260 deg. cent. The strengths of alloys Z and W are, for practical purposes, of the same order.

The curves in Fig. 11 show that increasing amounts of either silicon and copper adversely affect the strength at high temperatures. Comparing alloys X and Z, it may be seen that the addition of less than $^1/_2$ per cent silicon causes a reduction in strength of approximately 7000 lb. per sq. in. at 600 deg. cent., or on a percentage basis a reduction of over 30 per cent. Copper, like silicon, in this type of alloy improves the casting properties and also is beneficial from the standpoint of machinability. Its effect upon the tensile strength at high temperature is less than that of silicon, and for this reason the composition of the Al-Cu-Ni-Fe-Mg alloy that gives the best combination of strength ductility at atmospheric and high temperatures, together with good casting properties, is as follows, the percentages corresponding closely to the composition previously given for alloy X:

Cu	3.4 to 3.8 per cent
Fe	0.75 to 1.75 per cent
Si	0.6 to 0.8 per cent
Mg	0.5 to 0.7 per cent
Ni	0.6 to 0.8 per cent
Al	Remainder

Stability of Properties

"Short-time" tensile and hardness tests at temperatures in the range of from 200 to 320 deg. cent. do not represent the actual strength and hardness of the aluminum alloys that age-harden after quenching. Continued exposure to temperatures in the foregoing range causes a decrease in strength and hardness proportional to the time of exposure within certain limits. In order to obtain the actual properties at these temperatures of an aluminum alloy containing copper or magnesium in the heat-treated condition it would be necessary to soak for long periods before testing. The results of the tensile and hardness tests at high temperatures, shown in Figs. 11 and 12, represent accurately the comparative strength of the alloys, but their actual strength in the stable condition at 260 deg. cent. would be from 15 to 30 per cent lower, depending upon the alloy. The graphs in Fig. 13 show the results of hardness tests upon disks $^1/_2$ in. in thickness and 6 in. in diameter cast against a chill in green sand, heat-treated and aged, and then reheated to 260 deg. cent. and tested for hardness at atmospheric temperature at intervals of time ranging from 1 to 120 hours. After each period of heating at 260 deg. cent., the specimens were quenched in order to eliminate the aging during cooling from the reheating temperature to atmospheric temperature. The tensile and hardness tests given in Table 4 were made on standard cast-to-size specimens heat-treated and aged to maximum strength and hardness and then reheated at 260 deg. cent. in the same manner as the disks. The results of the hardness tests show that Y alloy has a higher stable hardness than the experimental alloys or Lynite 122. The strength of alloy X is practically the same as Y alloy, and both are superior to Lynite 122, while X alloy has the advantage of higher ductility in the stable condition.

In the temperature range of from 320 to 375 deg. cent. the Al-Cu and Al-Cu-Ni-Mg alloys reach a stable condition after short periods of exposure. The high-temperature tests in Figs. 11 and 12 represent quite closely the actual strength of the alloys exposed to these temperatures for long periods of time. Experiments on a 5 per cent copper aluminum alloy showed that quenching and aging for 1 hour at temperatures in the range of from 315 to 375 deg. cent. placed the alloy in a stable condition and that reheating at lower temperatures produced no further change in the physical properties. Under the heading of "Heat Treatment" it has been shown that alloy X has a good combination of strength and ductility when heated to 375 deg. cent. after quenching and that the properties of the alloy in this condition are stable, as the age-hardening of this alloy is governed by the

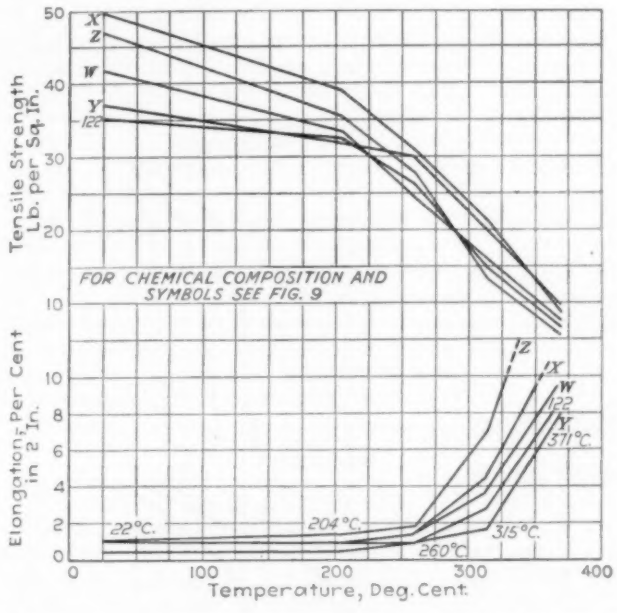


Fig. 11 Tensile Tests of Cylinder Head and Piston Alloys at High Temperatures (Sand Cast and Heat Treated)

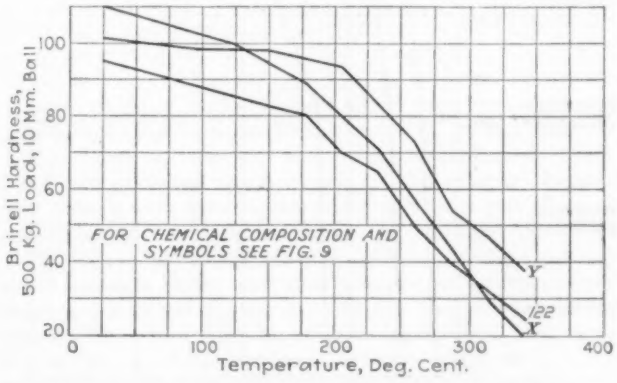


Fig. 12 Brinell Hardness of Piston Alloys at High Temperatures (Sand Cast and Heat Treated)

same principle as for the 5 per cent Cu-Al alloy. As stated elsewhere, the superior ductility of X alloy over the standard alloys, coupled with its property of retaining its tensile strength at high temperature, indicates that X alloy can be substituted for either of the standard alloys in cylinder heads to considerable advantage.

Hardness tests on pistons taken from various types of engines which had been run from 10 to 100 hours have shown that the hardness of the heads is approximately the same as the stable hardness values shown in Fig. 13 and Table 4 for alloys Y and Lynite 122. The skirts and piston-pin bosses of these pistons suffered no reduction in hardness and indicated that the operating

temperature was not high enough to cause softening during the ordinary life of a piston. Although at the writing of this report there are no test results available on pistons cast in X alloy that have been run in an engine, the laboratory tests indicate that this alloy can be used in pistons as well as for cylinder heads, and when compared to Lynite 122 it has approximately the same hardness in the stable condition, with the advantage of higher strength at both atmospheric and high temperatures and a lower specific gravity.

Specific Gravity

The specific gravities of alloys X and Z are of the same order

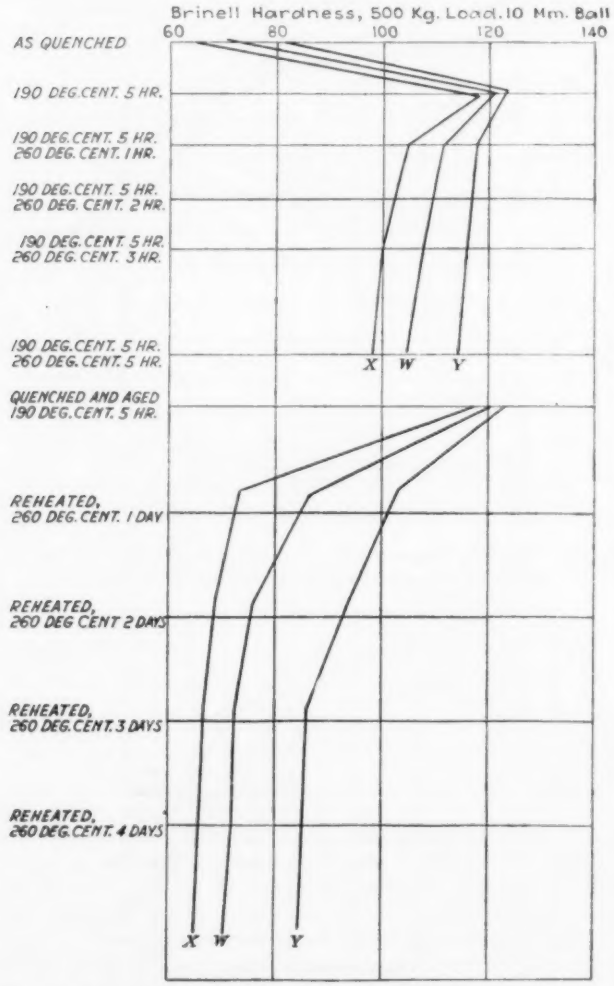


Fig. 13 Brinell Hardness of Piston Alloys at Atmospheric Temperatures (Sand Cast, Heat Treated, and Aged)

as the Y alloy and when compared to Lynite 122 afford a means of an appreciable saving in weight. The following shows a comparison of specific gravities:

Symbol	Specific gravity, grams per cc.	Percentage
122	2.91	100.0
Y	2.73	93.5
W	2.77	95.3
X	2.72	93.6
Z	2.73	93.5

The results listed were obtained on sand-cast specimens cut ¾ in. in diameter from the enlarged ends of standard tensile specimens and weighted in air and in water to the nearest 0.001 gram. The method of casting and pouring temperatures was the same, so that the results may be considered comparable. A saving of 6 per cent in weight can be obtained by the use of X alloy in place of Lynite 122, which amounts to a saving of 8 or 9 lb. in the cylinder heads alone of a conventional 400-hp. air-cooled engine.

Thermal Expansion

Coefficients of expansion measurements on this series of alloys have not been made, but a rough estimate can be made from comparison with similar alloys and from existing data showing the effect of alloying constituents. The coefficient of expansion for pure aluminum at 200 to 300 deg. cent. is 27.5×10^{-6} per deg. cent. (See Circular of the Bureau of Standards No. 346, p. 57.) The addition of copper up to 5 per cent has little effect upon the linear expansion at 300 deg. cent., but additions of silicon to the low-copper aluminum alloy appreciably reduce it.

The average coefficient of linear expansion in the range of from 20 to 300 deg. cent. for alloys containing copper in amounts less than 5 per cent, together with magnesium, silicon, and manganese, ranges from 25.0 to 27.0×10^{-6}. Available data show that alloys with 10 per cent or more of copper have a slightly greater coefficient of expansion than alloys of the low-copper or Y-alloy type.

For design purposes the coefficient of expansion for alloy X should be same as for alloy Y and Lynite 122, approximately 26.0×10^{-6} per degree cent.

Thermal Conductivity

The use of aluminum alloys in pistons and cylinder heads is advantageous from the standpoint of thermal conductivity as well as a means of saving weight. The following values taken from the eleventh report of the Alloy Research Committee, the Institution of Mechanical Engineers, shows the relation between the conductivity of pure aluminum and of cast iron:

Aluminum	0.50 to 0.52 cal. per cm. per sec.
Cast iron	0.09 to 0.11 cal. per cm. per sec.

The thermal conductivity of the 5 to 8 per cent copper-aluminum alloys sand cast and annealed has been recently shown by Griffiths and Schofield ("Conductivity of Aluminum Alloys and Bronzes," Journal of the Institute of Metals, 1928, Vol. 39) to be 82 per cent of that of the pure metal. Additions of copper lower the conductivity still further, and these authors give a value of 73 per cent of the pure metal for a 12 per cent copper aluminum alloy. Likewise, the addition of such elements as nickel, iron, or magnesium lowers the conductivity of the low-copper aluminum alloys. Griffiths and Schofield give a value of 73 per cent of the conductivity of pure aluminum for Y alloy, which corresponds to 0.40 cal. per cm. per sec., which is in accordance with the results previously obtained on this alloy and given in the eleventh report of the Alloy Research Committee.

An aluminum alloy containing 5 per cent copper, 2 per cent iron, and 0.5 per cent magnesium and one containing 9 per cent copper and 2 per cent nickel were found to have the same conductivity as Y alloy; so for practical purposes the thermal conductivity of alloy X should be about the same.

It should be borne in mind, however, that all the published data are for the as-cast or annealed alloys, and the effect of quenching from elevated temperatures such as employed in the heat treatment of the high-strength aluminum alloys would be to reduce the thermal conductivity, and the subsequent annealing of them would tend to restore the conductivity of the alloy to that of the as-cast and annealed condition.

Stability of Dimensions

The commercial casting alloys of aluminum increase permanently in size when in use. This permanent increase in dimensions in termed "growth," and its magnitude varies with the alloy and the temperature to which it is exposed in service. Leslie Aitchison (*Metal Industry*, May 16, 1924) attributed "growth" of pistons to stress set up during the solidification and cooling of the casting and mentions that after machining no growth takes place. He recommends annealing at 400 deg. cent. before the final machining operations as a means of dissipating the stresses. Zay Jeffries ("Journal of the Society of Automotive Engineers," 1920, Vol. 7, p. 295) has stated that all the aluminum-copper alloys suffer a change in volume when heated to 300 deg. cent., no matter how the alloys are cooled in ordinary method of production, but that this volume change can be made permanent at room temperature.

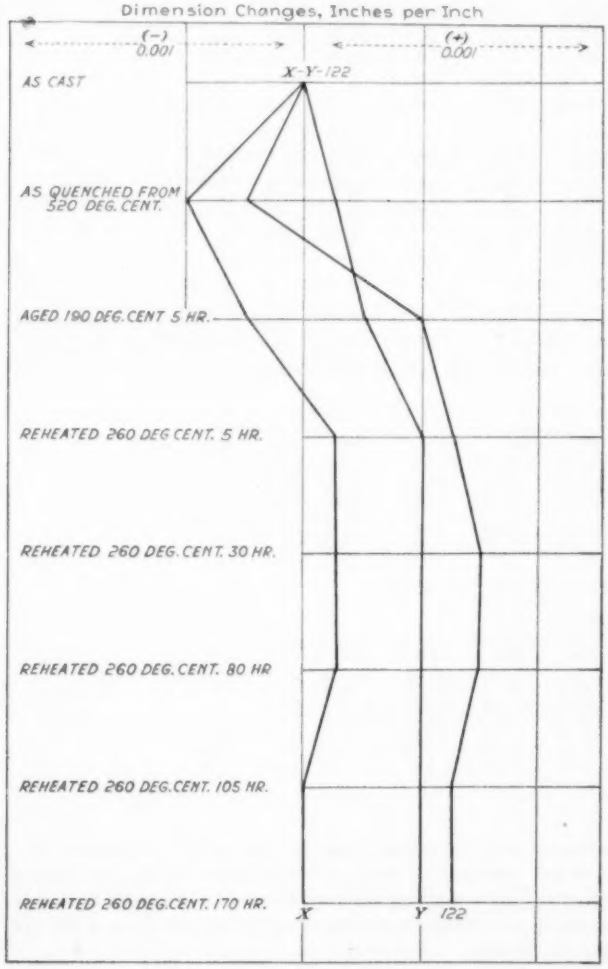


Fig. 14 Graph Showing Stability of Dimensions of Piston Alloys (Sand Cast and Heat Treated)

The laboratories of the Matériel Division have made growth measurements on pistons approximately 6 in. in diameter cast from Lynite 122, quenched and aged for 16 hours at 150 deg. cent., increased 0.007 in. after reheating to 260 deg. cent. for 5 hours. Pistons cast in Y alloy showed little or no growth, and in no instance has it been found to exceed 0.005 in. per inch. The chart in Fig. 14 shows the results of a series of careful measurments on a solid cylinder 6 in. in length and 3 in. in diameter, cast in alloys X, Y, and Lynite 122. It is shown that X alloy and Lynite 122 suffer a slight contraction when quenched from elevated temperatures. Y alloy shows a slight increase in volume which may be attributed to the natural aging that occurs during the period of three hours that the specimens were allowed to stand at atmospheric temperature in order to obtain equilibrium conditions of temperature.

After artificial aging or annealing at 190 deg. cent. for 5 hours, it is shown that all three of these alloys increase approximately 0.002 in. per inch after reheating to 260 deg. cent. for 5 hours. Longer periods at this temperature produced no further change in alloys X and Y, but Lynite 122 showed a slight increase. This is in accordance with the density changes shown in the graphs of Fig. 1. It was further shown that continued exposure to a temperature of 260 deg. cent. caused the alloys first to grow and then to shrink very slightly, which also is in accordance with the findings of the National Physical Laboratory (England). It is apparent that the growth of the aluminum casting alloys is very closely allied with their age-hardening characteristics and is due to the precipitation of the hardening constituents. From the practical standpoint, alloys aged to maximum hardness suffer little or no change in dimensions upon reheating, and short exposures to the upper limits of the age-hardening temperature range (150 to 260 deg. cent.) are more effective in producing stability of dimensions than are longer periods at lower temperatures.

Discussion

E. H. Dix, Jr.[5] In discussing this paper it will be necessary to confine remarks primarily to a comparison of alloys of the high-copper (122) type with those of the Y-alloy type of the older composition range, because the alloys discussed by the author are so new that the writer has not had sufficient experience with them to justify comments.

The Y alloy was introduced in this country as a material for pistons and cylinder heads largely because it seemed to retain its strength and hardness at elevated temperatures better than most other aluminum alloys. This characteristic is also possessed by alloys of the 122 type, but the fact has not received as much publicity, and as a result the Y alloy has come to be considered as the outstanding aluminum alloy for high temperature applications. Elevated temperature tests made at Wright Field[6] and published results by Mr. R. L. Templin[7] indicate that there is very little to choose between these two alloys as regards strength at temperatures of aircraft-engine operation.

The tests so far referred to are of the short-time type in which the specimen is pulled soon after temperature conditions have reached a practical equilibrium. In the case of heat-treated alloys subjected to high-temperature applications, such tests do not tell the complete story. A truer picture may be obtained by heating the alloys to the highest temperature to which they will be subjected during processing and engine operation, for a long period of time prior to making the high-temperature tests. Such results comparing the 122 and Y alloys have been obtained by Mr. D. A. Paul, working with Mr. Templin at the Aluminum Research Laboratories. Unfortunately the tests were made on sand-cast bars of 122 and forged bars of Y alloy, which of course makes the comparison less favorable for 122 alloy than if the results had been available

[5] Metallurgist, Aluminum Research Laboratories, Aluminum Company of America, Pittsburgh, Pa.

[6] J. B. Johnson, "Materials for Aircraft Parts Subjected to High Temperatures," A.S.M.E., Dec., 1927.

[7] R. L. Templin, C. Braglio, and K. Marsh, "Mechanical Properties of Aluminum Casting Alloys at Elevated Temperatures," A.S.M.E., May, 1928.

on Y alloy in the cast form also. The results given herewith were obtained on bars which had been annealed for 48 hours at 650 deg. fahr. prior to testing. This temperature is probably the maximum which would ever be reached by cylinder heads or pistons in aircraft engines, but a similar effect would be produced by annealing for a much longer time at a somewhat lower temperature. The tensile and yield strengths for 122 alloy at 500 deg. fahr. and above are superior to of Y those alloy.

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The author and the Wright Field organization are to be congratulated for the full realization, which is evident from a reading of the paper, of the importance of the casting qualities of a new alloy to be used for cylinder heads. Before a cylinder head may be used it must be cast, and not matter how desirable an alloy may be from other viewpoints, if it cannot be handled in the foundry, it is useless. The author has carried his investigation through the preliminary laboratory stage. The next

No. 122 Cast, Annealed[a]

Property	Temperature, deg. fahr. 75	200	300	400	500	600	700
Tensile, strength, lb. per sq. in.	24,000	22,000	19,700	16,700	13,200	9,600	5,800
Per cent of room temperature	Str.	91.7	82.1	69.6	55.0	40.0	24.2
Yield point, lb. per sq. in.	12,000	11,000	10,000	8,800	7,000	5,400	4,000
Per cent of room temperature	Str.	91.7	83.3	73.3	58.3	45.0	33.3
Elongation in 2 in., per cent	2.0	2.0	3.0	5.5	9.5	18.0	30.0
Per cent reduction of area	2.0	2.0	2.0	4.0	6.5	9.0	13.0
Modulus $\times 10^{-6}$, lb. per sq. in.	10.6	10.4	10.2	9.4	6.8	5.5	4.1
Y Forged, Annealed[a]							
Tensile strength, lb. per sq. in.	27,500	27,000	26,000	21,500	11,500	7,750	5,200
Per cent of room temperature	Str.	98.2	94.6	78.2	41.8	28.2	18.9
Yield point, lb. per sq. in.	9,000	8,700	8,400	7,750	5,800	4,000	1,700
Per cent of room temperature	Str.	96.7	93.3	86.1	64.4	44.4	18.9
Elongation in 2 in., per cent	19.0	19.5	21.0	32.0	58.5	78.0	92.0
Per cent reduction of area	34.0	35.5	39.0	57.0	78.0	88.0	91.0
Modulus $\times 10^{-6}$, lb. per sq. in.	11.0	10.5	10.0	9.6	8.7	5.3	4.3

[a] Annealed 48 hr. at 650 deg. fahr.

The following results on 122 alloy in the sand-cast and heat-treated condition and in the sand-cast and stabilized condition may also be of interest:

No. 122 Cast, Heat-Treated[b]

Property	Temperature, deg. fahr. 75	200	300	400	500	600	700	800
Tensile strength, lb. per sq. in.	40,700	37,700	35,700	38,300	26,300	10,800	4,800	3,000
Per cent of room temperature	Str.	92.6	87.7	94.1	64.6	26.5	11.8	7.4
Yield point, lb. per sq. in.	35,000	32,300	28,800	31,000	21,300	8,000	3,000	..
Per cent of room temperature	Str.	92.3	82.3	88.6	60.9	22.7	8.6	..
Elongation in 2 in., per cent	0.67	1.00	1.00	1.00	2.50	15.0	35.0	70.0
Per cent reduction of area	0	0	0	0.50	2.30	14.8	43.0	80.3
Modulus $\times 10^{-6}$, lb. per sq. in.	10.4	10.3	10.1	9.2	7.1	4.4	2.8	..

[b] Solution treatment of 12 hr. at 960 deg. fahr. and quenched in water at 180 deg. fahr.

No. 122 Cast, Stabilized[c]

Properties	Temperature, deg. fahr. 72	200	300	400	500	600	700	800
Tensile strength, lb. per sq. in.	28,100	27,500	26,300	24,000	20,000	14,000	7,000	3,800
Per cent of room temperature	Str.	97.9	93.6	85.4	71.2	49.8	24.9	13.6
Yield point, lb. per sq. in.	20,375	20,000	19,000	16,500	13,500	10,000	4,900	..
Per cent of room temperature	Str.	98.2	93.3	80.9	66.3	49.1	24.1	..
Elongation in 2 in., per cent	1.00	1.00	1.00	2.0	3.0	6.0	17.0	56.0
Per cent reduction of area	0.76	0.50	0.35	0.75	1.17	6.89	24.6	54.6
Modulus $\times 10^{-6}$, lb. per sq. in.	10.8	10.6	10.3	9.8	8.7	7.0	4.8	..

(c) Stabilized 3 hr. at 600 deg. fahr.

It is the writer's opinion that the requirements for a satisfactory piston alloy are somewhat different from those of a cylinder. Growth, expansivity, and extreme lightness are not quite so important in cylinders as in pistons. Casting characteristics and mechanical property requirements are also sufficiently different for the two applications, so that it is possible to obtain an alloy suitable for pistons which may not be entirely satisfactory for cylinder heads and vice versa. In this connection, a new piston alloy which is just emerging from the development stage and is being used by some of the automobile, truck, and aircraft companies should be mentioned. This alloy has 15 per cent less expansivity than the alloy of the 122 type, and because of other characteristics may be fitted with 20 per cent less clearance.

The specific gravity as given by the author for X metal appears to be rather low for the preferred composition. The specific gravity of pure aluminum is 2.70, so that it is difficult to understand how an alloy containing as much as 3.6 per cent copper, 1.25 per cent iron, and 0.7 per cent nickel could have a specific gravity as low as 2.72 when there is only 0.7 per cent silicon and 0.6 per cent magnesium to compensate for the heavier added necessary step of course must be an actual production run. It is not sufficient to make a few cylinder heads, but rather, before the alloy may be accurately evaluated, it must be run in production in comparison with other known alloys on a difficult type of casting.

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a better alloy for cylinder heads and pistons which has resulted in this paper is a sign of the times. There is no organization in the country so well able to see the future needs of the aircraft engine industry as the Matériel Division at Wright Field. It is to be hoped that the excellent work of the author and others of the organization will continue toward this end.

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The 132 alloy referred to in Mr. Dix's discussion has been developed to overcome these difficulties. It contains about 14 per cent silicon, with smaller amounts of nickel, copper, and magnesium. Its specific gravity is 2.71, only slightly greater than that of pure aluminum, and its expansivity about 80 per cent that of 10 per cent copper alloy or of Y alloy. In addition to this, 132 alloy machines far more easily than the 20 per cent silicon alloy and may be heat-treated so as to produce hardness from 100 to 130 Brinell. It is particularly desirable for aircraft pistons, both on account of its low expansion and its low gravity, which not only reduces reciprocating weight and inertia forces, but which permits an additional weight saving on account of the smaller counterbalances required. The material is being used for pistons by a number of leading aircraft, bus, and automobile-engine manufacturers.

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Elongation in 2 in., per cent	19.0	19.5	21.0	32.0	58.5	78.0	92.0
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AER-52-32

Fire-Prevention Problems

By C. G. McCORD,[1] PHILADELPHIA, PA.

This paper gives a few general principles and points out the trend of investigations that have been made along the lines of fire prevention and the design of fire extinguishers. Prevention in the case of aircraft is a relative term as there are no thoroughly satisfactory fire extinguishers, although the improvements are encouraging as more thought is being given to the subject.

While much work is in progress toward the development of aircraft fire-extinguishing systems in Government departments and by the manufacturers of that type of equipment, and although the need is manifest, there is at present no basis for a hard and fast regulation regarding the nature of equipment to be prescribed for installation in military or commercial airplanes.

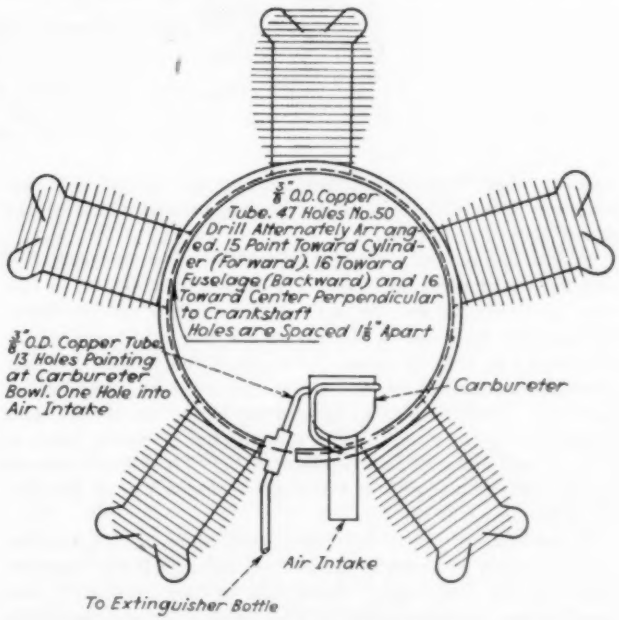
MOST FIRES in conventional-type airplanes occur in the engine compartment, and main reliance is placed on apparatus designed to reach that space, portable equipment usually being provided to take care of the other parts of the plane. Protection of multiengined planes, where the engines are mounted in the open or in nacelles, is somewhat more complicated, but the general rules are the same as for engines mounted in the nose of the fuselage. The cause and the cure for fires in aircraft must receive consideration together, and in an effort to cover the ground some grouping or classification of fires is in order.

Fires in the air are mainly attributable to broken or leaky fuel systems or oil systems aided by hot engine surfaces including exhaust stacks, electrical discharges, friction or metal impact sparks, open flame from so-called backfires (probably very little from normal exhaust flames, which are usually shrouded in nitrogen, carbon dioxide, and water vapor), or by hot carbon or metal particles from exhaust stacks. In connection with this last-named cause it has been reported that black iron or steel stacks, worked on by erosion and products of combustion of doped fuel containing lead compounds and bromine, will throw off hot scale particles that are a menace where gasoline fumes or even dry doped fabric may be within range.

Some of the causes named are instantaneous and do not continue; others persist and may tend to reignite the fire after the first flames are extinguished. Lightning has been blamed for the destruction of some planes in flight, but as a direct cause of fire it may be disregarded, although bonding of metal parts to avoid possible brush discharges of collected atmospheric electricity has become a standard practice and is probably needed.

Rupture of fuel tanks is a possible cause of fires in the air as well as on the ground. A rupture has occurred in a Navy plane when a partly filled tank was subjected to the hammer of the fuel when the plane was pulled sharply out of a steep dive. In addition to military combat planes many commercial types are now in use, capable of acrobatics imposing high stresses, and the principle being demonstrated, due precautions must be observed in all types.

One cause of reignition of airplane fires on the ground which has been observed during the naval aircraft factory tests and which should be the cause of concern is gasoline released by breaking of fuel lines or tanks and which runs out on the ground. Flames traveling down the dripping stream will often set a puddle of the gasoline afire. Even if the flames are extinguished in the plane the fire will retrace its path up the drip, probably after the extinguishers have been exhausted. There is no means for determining the attitude of the plane after landing, which adds to the difficulty of combating this phase of the danger.



FIRE-EXTINGUISHER MANIFOLD INSTALLATION FOR RADIAL ENGINE

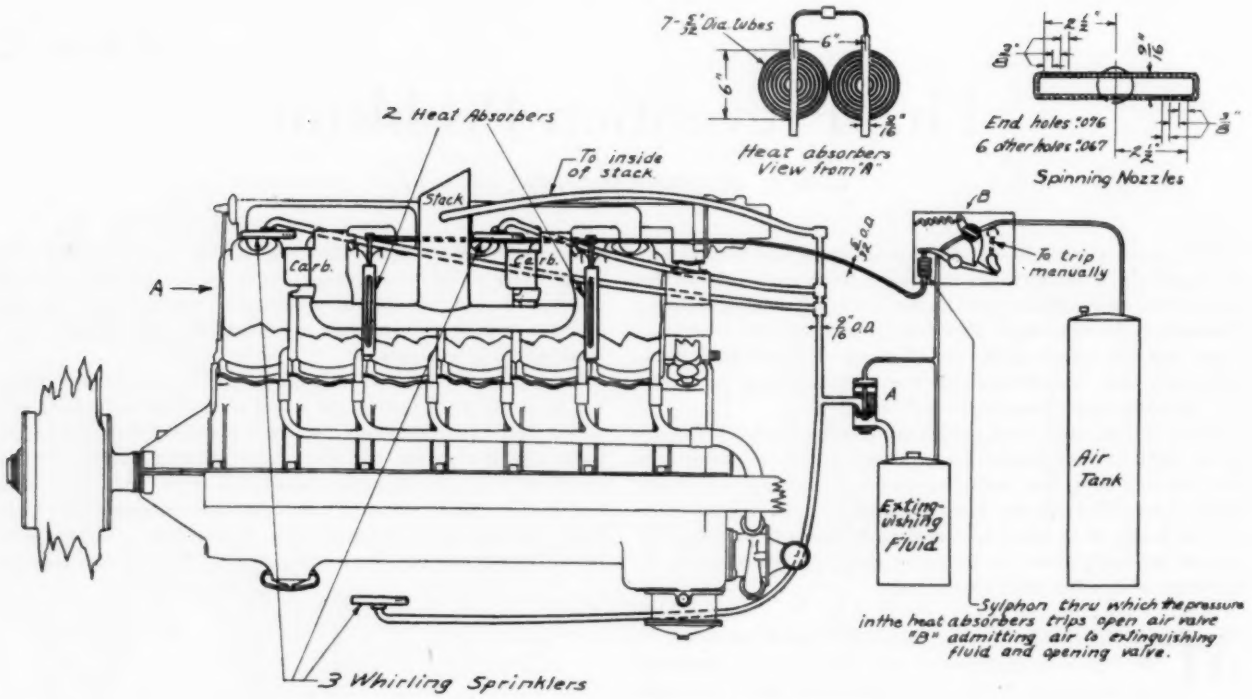
To prevent carburetor back fires causing trouble the air intakes may be screened, with a reduction in brake horsepower as high as 4 per cent, using the Davy safety lamp principle and triple screens properly spaced. However, screened or not, the air intakes should be led well outside the engine cowling. This is not always simple in a seaplane where spray may strike the air horn, but no deviation from the rule should be permitted where it is possible to extend the intakes or air horns and drain the bends and pockets.

Magnetos are usually flame proof to the extent that they could be run safely in an atmosphere of explosive gasoline-air mixture. Broken high-tension lines should be guarded by attention to supporting and location as well as to strength of the materials. When the fire occurs outside the engine compart-

[1] Lieutenant Commander, U.S.N.; Superintendent of Tests, Naval Aircraft Factory.

Presented at the Fourth National Aeronautic Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Dayton, Ohio, May 19 to 22, 1930.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.



INSTALLATION OF AUTOMATIC SAFETY FIRE DEVICE

ment or other closed compartment it appears impossible to provide any form of built-in extinguisher apparatus which will be effective.

On the ground consideration should be given to crash fires and hangar or servicing fires. The causes to which crash fires may be attributed are the same as those in the air just listed. Hangar fires and fires in aircraft on the line, when engines are not being run, may be caused by smoking, tool sparks, oily rags or waste accumulations, and are spread by paint, dope, oil, or gasoline. These fires differ from those in the air in that there is less chance for strong drafts, and the use of water, foam, or of powdered material may be resorted to. These fires are more apt to destroy property and less apt to cause loss of life than those occurring in the air.

Fireproofing of fabric by doping with ammonium phosphate or boric acid is no longer in general use, although it was a requirement several years ago. Ventilation of the interior of wing panels where explosive gases may be formed during wing doping is, however, common practice.

Short stacks on the exhaust of an engine, especially if finned for heat dissipation, are better, from the fire-hazard standpoint, than collector rings or long exhaust stacks of the manifold type. A temperature study of a Liberty engine showing that while the manifold type of stack may run as high as 1380 deg. fahr. at its end (full throttle, lean mixture, retarded spark), the short finned stacks on the same engine ran below 475 deg. fahr. The comfort of personnel in the matter of exhaust gases and noise, however, is a factor in the design, and in most modern types of plane the exhaust is collected and led away from the engine, an advantage for night flying, and an aid to comfort, although to some extent a menace in a crash.

Gaseous fuels in lighter-than-air craft seem to be dangerous in proportion to their density as a fire risk. Those heavier than air tend to drop to the lower part of the ship, from a leak, whereas those lighter than air rise to parts where there is no danger. As between liquid and gaseous fuels from the fire-hazard standpoint, the chance of quickly releasing the storage tanks from the ship in emergency would recommend the former.

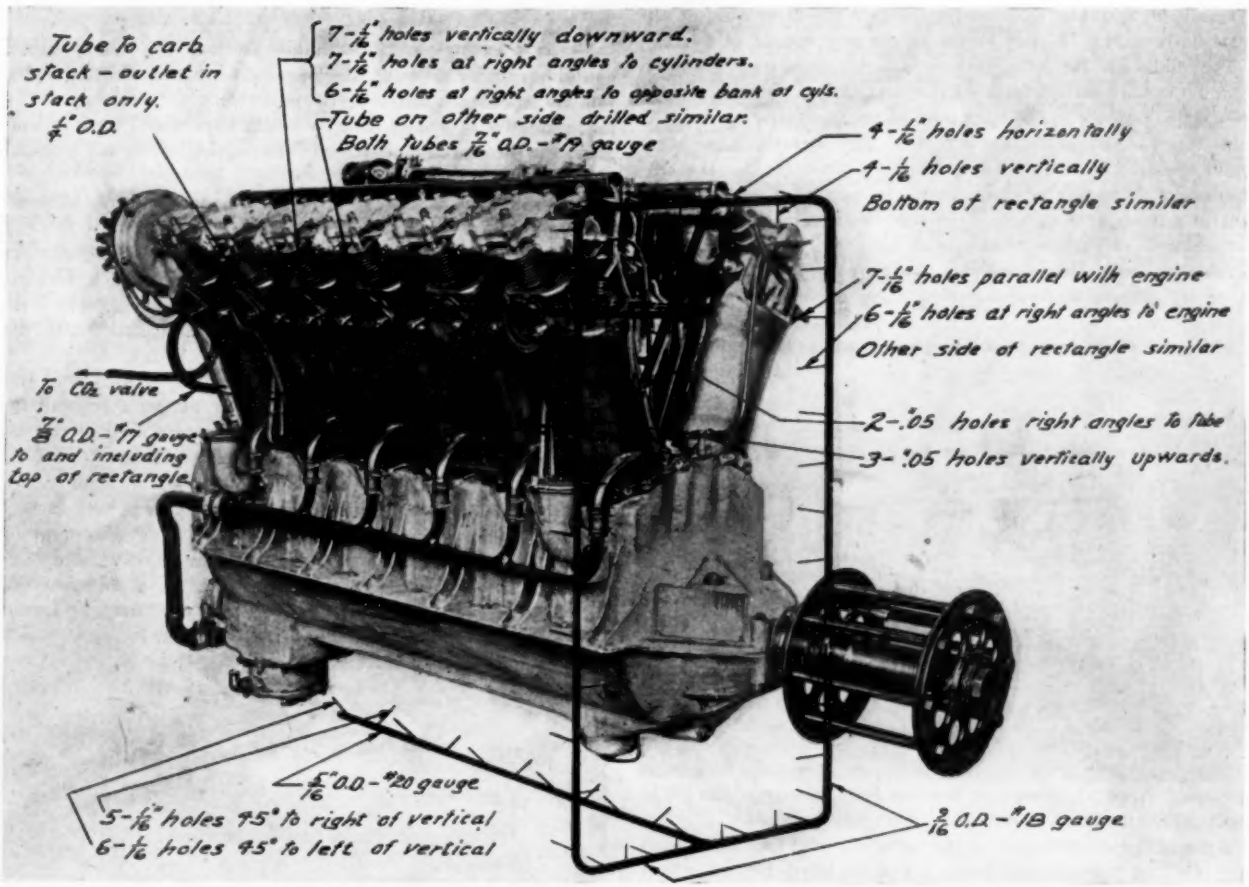
Placing pipes in jackets containing fire-extinguishing fluid and the introduction of carbon tetrachloride into the gasoline are suggested methods of protection of those vital parts. The former method is open to criticism on account of the danger of the outer jacket being quite as liable to failure as the fuel lines, and probably being drained before the fire occurs. Also, such a system would be heavy. The latter method is of little practical value except in local spots, as it was determined by test in an open container that dilution amounting to 71.5 per cent of CCl_4 in gasoline was required to make a non-inflammable mixture.

In pressure-type gasoline systems it has been suggested that CO_2 pressure be used instead of air pressure. This would add an item of weight, and seems hardly worth the cost as the protection afforded is slight. Whether forcing the gas into the gasoline under pressure as in charging sparkling beverages would have an effect on the detonating characteristics of the fuel is another matter and aside from the discussion.

A general rule in the air is to cut off the fuel supply first, close the shutters and sideslip the plane to keep the flames away from wings and cockpit, and if possible to blow out the fire, meanwhile operating the extinguisher, and then before landing or crashing to cut the ignition. This gives reasonable assurance of the lines and carburetor bowl being empty or at least partly so.

The running engine may serve to blow out the fire or to impact the fire-extinguishing medium against the fire, which chances offset the argument that it fans the fire. Unless wood or some similar substance which burns without definitely volatilizing is present, the strong draft is a cure for rather than an aid to combustion.

Imperfect electrical connections and worn insulation are sources of sparks. The preventive, of course, is rigid inspection and proper closing of connection or fuse boxes. Exposed surfaces on water-cooled engines except exhaust stacks are normally



FIRE-EXTINGUISHER MANIFOLD ON WATER-COOLED ENGINE

below 212 deg. fahr. Cylinder barrels of air-cooled engines may run from 300 deg. to 500 deg. fahr., but these are exposed surfaces and unlikely to be a source of danger as installed.

Lubricating or crude oils are probably more dangerous in contact with a hot surface than gasoline, which evaporates before combustion occurs when in contact with surfaces. Latent heat of vaporization also enters into this. For heated surfaces 720 deg. fahr. seems a safe temperature, the ignition temperature of lubricating oil and doped fabric being around 750 deg. fahr. Although with many liquid fuels an inflammable gas is given off when there is contact with a hot surface at even lower temperatures, this may dissipate too rapidly to ignite.

A tight unpainted firewall is necessary to separate the engine compartment from the rest of the plane when engines are mounted in the fuselage. Use of wood, fabric, or paint in an engine compartment adds to fire risk, as all materials of this nature tend to be fanned to a glow by a strong air blast, whereas volatile materials under the same condition have the fire blown out and its spread limited by such a blast.

Battery ignition is probably more hazardous than magneto ignition as the magneto stops when the engine does. The leads are shorter and less liable to breakage from the same cause that would rupture a fuel line, and the magneto itself is easily flameproofed. Static sparks are generated when gasoline is being strained through chamois skin, a very common practice, unless the metal parts such as buckets, tanks, and hose nozzles are grounded. Fuel storage tanks can be protected from fire best by keeping the tanks always full, floating the fuel over water, and having due regard to small leaks in the tanks and connections. This system is employed on seaplane tenders and aircraft carriers in the Navy.

EXTINGUISHING MEDIUMS

Mediums for fire extinguishing may be divided into (*a*) those in a liquid state at ordinary temperatures but forming gases or vapors when heated, (*b*) those which at ordinary temperatures are in a gaseous state, (*c*) solids in a powder form which produce gases when heated, and (*d*) foam.

As applied to aircraft fires in which gasoline or oil may be expected to be liberated, the principle of use of the various classes of extinguishing mediums is the same. Oxygen is excluded from the burning surfaces by the formation of a surrounding blanket or cloud of gas. The change in state or expansion involving a matter of latent heat cools the surrounding surfaces and the fire is extinguished. The problem so far is simple, but the danger of recurrence of the fire is not eliminated. If there is fabric or wood present in the region of the fire, there is danger of recurrence from incandescent sparks remaining in charred material after the flames are out. Wetting the materials is desirable.

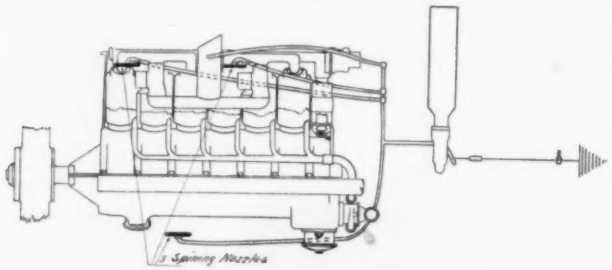
WATER

The cheapest and easiest to handle of the liquid mediums, of course, is water. Due to the formation of steam it is useful when applied to a hot fire, but comparatively useless where the fire is less intense. Its use is desirable where there is no gasoline or oil in the fire, a condition seldom found, but it serves to spread gasoline or oil when present. The application of water

is mainly in sprinkler systems for hangars where wetting of all parts surrounding the fire serves to prevent spread of flames.

The materials for extinguishing aircraft fires now in most common use are carbon tetrachloride and carbon dioxide. Both are easy to procure in the open market and are fairly cheap. A substitute for carbon dioxide is methyl bromide, which has been found effective and while not so readily obtainable has the advantage of using lighter equipment. In general it may be said that in aircraft at the present time the choice is limited to carbon tetrachloride or carbon dioxide, and neither of these is entirely satisfactory from all points of view, nor have the experiments been an entire success wherein a combination of the two mediums is employed. The solution is not in sight at present.

The gases which seem most suitable for use as fire extinguishers are constituents of engine exhaust gases. It seems logical to



Water-Cooled Engine Manifold

provide some apparatus for compressing these gases in the plane and storing them as commercial gases are stored for the same purpose. No work along this line has been attempted, and it is mentioned merely as a possible future development.

A few of the shortcomings of the materials tried may be pointed out: Carbon tetrachloride forms a gas, on being heated, which smothers life as well as fire, if highly concentrated. It is reputed to cause an exciting effect on persons subjected to its atmosphere for breathing, and to make the person so affected uncertain in judgment at a time when quick action and use of all faculties may be most needed. Phosgene is also a constituent of this gas, and the Bureau of Mines states that a concentration as low as 25 parts in a million, on 30 to 60 min. exposure, may be fatal. The average phosgene from eight fires on record was 34 parts in a million, only two of the eight fires showing less than 25 parts.

Pure carbon-tetrachloride vapor is lethal in concentrations as low as 2.4 to 4 per cent, and methyl bromide is dangerous within the same range with 30 to 60 min. exposure, and its rapid vaporization has a bad effect if applied to the bare skin, probably due to a combination of drying and latent heat. The products of decomposition under heat of methyl bromide are little known.

Carbon dioxide in higher concentration shuts off oxygen from the lungs and its effects are too well recognized to merit discussion.

Foam

A foam composed of carbon-dioxide bubbles covering a liquid surface has the advantage of blanketing the surface very well, spreading rapidly, and in still air it will remain on the surface until danger of reignition is past. The bubbles are usually formed in licorice or some similar substance. In a slipstream or strong air blast, foam is of no use as it is blown away as fast as produced, and is not to be considered as acceptable for aircraft use except in hangars.

Foam appears to have the advantage of holding and slowly releasing the carbon dioxide with which it is charged, and for that reason is comparatively safe in a poorly ventilated space.

Methods of Operation

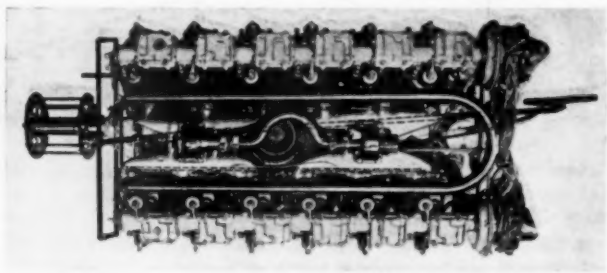
Gaseous extinguishing mediums are held in heavy-walled containers under pressure, and released to the distributing manifold by opening a valve or by puncturing a metal cap. The gas is rapidly expended, and no means is provided for a second charge, when the first charge has been expended, in any apparatus so far submitted.

Carbon dioxide being liquid under storage conditions of pressure must have the outlet of the container at the bottom or it must be provided with a siphon tube extending to the container bottom to secure the best results. The weight of a carbon-dioxide container to hold a 5-lb. charge is about 10 to 11.7 lb. empty. They are made of seamless steel and conform to the I.C.C. regulations for shipping containers.

Liquid mediums are contained within thin-walled shells and ejected by pumps, air or gas pressure, or by pressure built up by explosion of a powder charge. The pressure used in various types of extinguishing apparatus varies considerably.

The volume of carbon tetrachloride in the container varies from one quart to one gallon for each compartment served, and depends on the volume of the compartment. The amount of carbon dioxide varies from 3 to 20 lb. per compartment.

Wide spraying of the extinguishing medium is an essential, and in order to insure this the gas or liquid used must be forced under pressure through manifolds or spray pipes to all parts of the plane in which fire may be expected to break out. Rose nozzles and rotating sprinklers have been tried with a measure of success.



Installation of Manifold on Water-Cooled Engine

Placing of CCl_4 bombs in easily broken glass containers which, shattering on impact, drench the engine compartment is of undoubted value in preventing crash fires, but their fragility makes them hard to handle. One type makes use of a carbon-dioxide cartridge with a cutter which releases the pressure into the fluid container. Another interesting development of foreign origin consists of two pressure systems, being automatic in action.

The starter system made use of a honeycomb container placed in the vee of an engine, which when heated evaporated a contained liquid, creating a vapor pressure which operated through a siphon and bell crank to release compressed air or carbon dioxide from a container communicating with a tank holding the extinguishing fluid, forcing it out to the fire through rotating nozzles. The same pressure served also to shut off the fuel supply, close the shutters, ground the magnetos, and close the throttle. The total dry weight of the system was below 25 lb. Leakage in the pressure system and possibility of premature operation appear to be the drawbacks to this well-thought-out scheme.

Manifolds

Discharge lines must be as short and as free from bends as possible. The container and actuating apparatus are not large

and may be located at any convenient point, provided the release operating gear is within reach of the operator and is distinctively marked.

As to material for the manifolds in the immediate vicinity of the fire, duralumin or similar alloys do not stand heat as copper or ferrous alloys will. The piping, in case the apparatus is not operated immediately upon starting the fire, will burn under the blow-torch effect of the fire in the strong air stream. Bronze or copper, which are easily formed, are recommended for this part of the equipment.

The remainder of the piping may be of light material, duralumin tubing having been found reliable in thin-wall form. The minimum allowable inside diameter of tubing is $^3/_8$ in. for carbon tetrachloride or CO_2. Annealed copper, aluminum, or duralumin tubing designed to bursting at 4000 lb. is required for carbon dioxide.

The location and size of outlet holes vary for each type of installation. For an open radial-engine installation, where carbon tetrachloride is used, it is usually sufficient to pipe the manifolding around the carburetor, and some form of enclosure is built to surround the installation and hold in the smothering gases. In general this is an easier matter to handle than where a vee or in-line engine is mounted in a closed cowling, as in the



Fire-Extinguisher Test Rig Showing Carbon-Dioxide Snow Extinguishing Fire in an Uncowled Air-Cooled Engine

latter case not only must the distributing manifold be led around the engine and interior of the compartment with a larger number of outlets, but the ventilation of the compartment must be arranged so as to give enough air flow to carry off oil and gasoline fumes without sweeping away the extinguishing gas too rapidly.

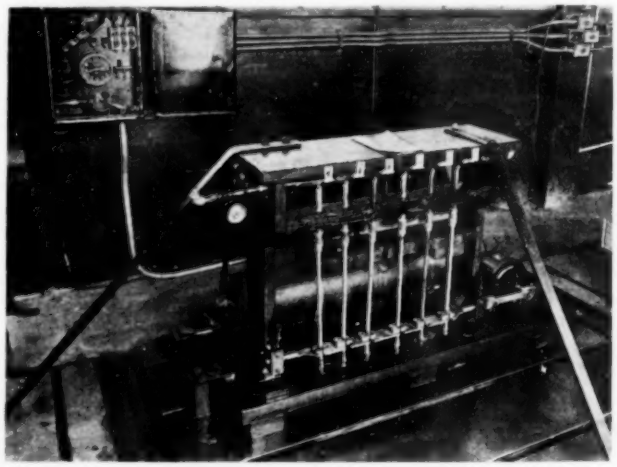
The direction of the outlet of the jets, when using a gas, is important, even to include jets directed against the fire wall behind the engine. The holes must be at least $^1/_{16}$ in. in diameter.

Prevention of the main cause of fires may be insured by selection of proper materials, design and shop practice regarding fuel and oil tanks and plumbing, with due regard to inspection at intervals to insure against loosening of joints by the ever-present vibrations, partial breaks in piping and tanks, and imperfect closing of carburetor-float joints or engine parting faces which would cause leakage of fuel or lubricating oil in flight, especially when the plane is to be placed in unusual attitudes.

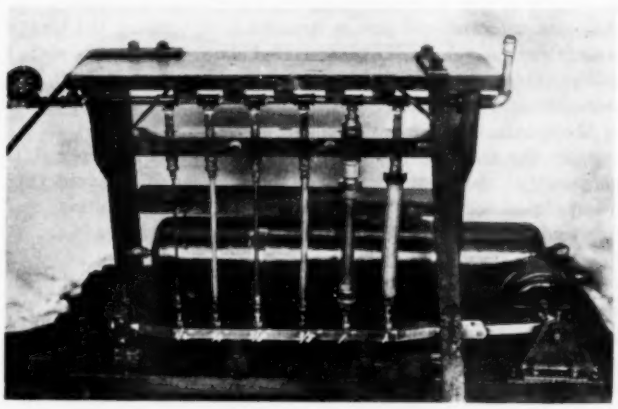
The importance of strength of fuel lines, tanks, and connections cannot be too strongly stressed, and material to be used must be capable of standing up under both vibration and shock tests. Here, as in some other regards to be mentioned later, the designer must allow safety to come before considerations of weight.

Strength, in this regard involves the factor of corrosion and protection against it, also the element of design for proper support of fuel lines to avoid a whip which may cause loosening of joints or embrittlement of the metal. Copper tubing requires torch annealing at the time of general overhaul of a plane to take care of this embrittlement.

The effect of broken fuel lines is twofold: The flow of gasoline to the carburetor is reduced, and the fuel is exposed and usually fed under pressure to the potential source of ignition. The reduced fuel flow to the carburetor may be sufficient to cause lean-mixture popping or backfiring, which under certain conditions is an ignition source.



Vibration Tester for Solid Fuel Lines Showing Automatic Shut-Off Gear



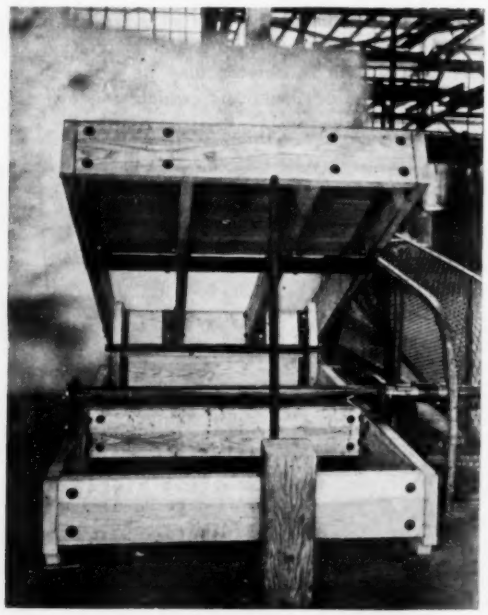
Vibration Tester as Used on Flexible Fuel Lines

Choice of material for fuel lines for Navy aircraft is determined by tests. The threading and belling qualities of solid tubing material are obtained by shop tests, and the tensile strength and elongation are determined in the usual way; then specimens of the tubing are placed in a vibrator rigidly secured at the upper end, with the lower ends attached to a solid bar which is shuttled back and forth by a motor-driven eccentric. The tubing is filled with gasoline under pressure, and the apparatus is designed to stop operation when a rupture occurs anywhere in the pressure system.

This method of tests provides for a severe reversal of stresses of a vibratory nature, and introduces compression and tension. The tubing and attached fittings are, by means of this vibrating jig, subjected to a punishment probably exceeding the maxi-

mum stresses encountered in aircraft, but which forms a basis for comparison between different types of tubing and fittings.

Recently nickel, copper after various treatments, aluminum, and duralumin samples have been subjected to comparative tests to determine their relative value for fuel lines, and in addition a thorough investigation of the properties of several brands of flexible tubing is being conducted for the same purpose. In this case an additional test is imposed to determine the tensile strength of the tubing when assembled, as the fittings are seated to the tubing ends. Loads in tension are applied in 50-lb. increments at 5-min. intervals to take up any cold flow of solder in the joints.



Vibration Tank Tester Showing Construction

After the material for the fuel lines has been selected it is necessary to design the layout of the lines so as to avoid long unsupported sections which tend to whip under vibration, and to avoid nips or possibility of wearing where bulkheads are to be passed.

On complete overhaul of a plane it is recognized as good practice to anneal the fuel lines, if of copper, usually by torch. It is questionable whether duralumin tubing should be replaced or similarly annealed. The condition of bends and belled joints is the index in this case.

Tanks

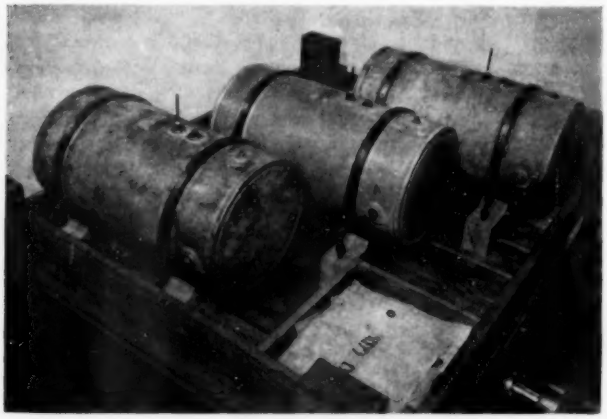
Fuel tanks are also subjected to vibration testing by securing them to a heavy table under mounting conditions similar to those encountered in a plane, the table being vibrated at a high rate of speed with the tanks partly filled with water. The means for vibrating the table are of considerable variety, the simplest way of securing an uncushioned shock being to rotate a large hex bar of steel upon which the table rests. Wear of the bar is the main trouble with this method. The punishment is very severe in this case, as in the tubing tests, and 25 hours appears to be enough to settle the question of suitability.

A shock test by allowing a partly filled tank, swinging on a long line from overhead suspension, to strike a cushioned wall seems desirable for fuel tanks designed for combat or stunting planes. A typical test of a fire-extinguishing system is conducted as follows:

A fire is lighted by torch in the gasoline around a small radial engine with flooded carburetor, and a fuel-supply line sawed partly through at two points near the carburetor, mounted in the slipstream of a pusher propeller driven by an engine placed in front of the test rig.

Gasoline is fed to the fuel system at the rate of 50 to 60 gal. per hr. After the fire has burned for a period of not over 20 sec. the fire extinguisher is operated, the variables in the test being slipstream velocity, amount of extinguisher charge, internal diameter, and length of tubing in the manifold. Installing or removing engine cowling is an additional variable.

Time for the fluid to reach the discharge jets and time to extinguish the flames are recorded if possible, as well as time to empty the system. Usually, 20 to 30 trials are attempted with each combination of variables in order to secure fair averages. For further testing a vee-type engine completely cowled, a pan of gasoline in the vee is used, and in some cases oily waste or charred wood is placed in the engine compartment. For a particularly hot fire the addition of naphthalene flakes in the open pan of gasoline will aid.



Vibration Tank Tester With Tanks in Place

The results of these tests are reported to the Navy Department, and where proprietary apparatus or materials are involved the results are not available for general publication except in the form of specifications for the supply of material and the construction of aircraft which are written around the test results.

Discussion

H. E. Newell.[2] Commander McCord's paper is an able statement of the fire-prevention and protection problem introduced by the airplane. It is unquestionably true that when applied to the airplane the shortcomings of the various types of fire-extinguishing devices and systems are emphasized; nevertheless, it is the writer's opinion that there are available today systems and devices that will afford a very substantial degree of protection if properly installed.

Available extinguishing agents are capable of successfully coping with the types of fire likely to be encountered on the airplane. Adequate means of application, however, stands out as a feature of pronounced importance and concerning which greater knowledge is essential.

The solution of the fire hazard attendant to crashes is indeed a difficult problem. By its very nature it tends to prevent mitigation through installation, no matter how rugged, and thus

[2] Engineer, National Board of Fire Underwriters, New York, N. Y.

largely offsets the extinguishing value of the fire-extinguishing agent provided. In the writer's opinion this hazard will be reduced only by the development of safer fuels, coupled with improved features of design and construction.

With reference to reignition of airplane fires originally due to broken fuel lines, this can possibly be mainly overcome by providing extinguishers of larger capacity on the plane. This condition frequently develops at or near hangars, and experience otherwise has also demonstrated the need there for extinguishers of capacity greater than the usual first-aid appliances, such as a 50-lb. or 100-lb. CO_2 extinguisher on wheels or others of type suited to gasoline fires in the form of generators or so-called engines.

In connection with carburetor backfires, screens are very important. It is suggested that there be provided a separate compartment or baffled space for a screened carburetor intake.

The statement concerning fires that occur outside the engine or other closed compartment is very true and should be borne in mind. Extinguishers to protect such spaces could be designed and would be successful, but the probability hazard is low, and, further, the weight of such systems would be prohibitive.

With reference to fires on the ground and in hangars no mention is made of carbon dioxide or carbon tetrachloride as fire-extinguishing agents in Commander McCord's paper. For first-aid purposes both of these extinguishing agents have proved effective, and extinguishing devices using these agents have been examined and listed by Underwriters' Laboratories.

In connection with fires of this type, however, it is of pronounced importance that there be at hand extinguishers of greater capacity than those commonly used for first-aid purposes.

With reference to exhaust stacks, it is the writer's impression that crash tests indicated that long exhaust stacks were the main cause of fire. Commander McCord has shown rather conclusively that, so far as temperature is concerned, short stacks are preferable. In comparing the manifold type of stack and the short-finned stack he states that the temperatures in the latter run below 475 deg. fahr., which is merely another way of saying that the temperatures were below the point of self-ignition. The suggestion has been made that stacks be air-cooled in order to reduce temperatures.

In connection with pressure-type gasoline systems, the writer heartily concurs that substituting CO_2 for air pressure would accomplish nothing regarding fire prevention in connection with the crash hazard and might effect combustion of gasoline due to the CO_2 dissolved, as stated in the paper.

Contrary to common belief, fires are much harder to extinguish and are considerably more voluminous with low or zero slipstream than with high-velocity slipstream. Actual tests have shown that under certain conditions fires could be blown out by wind velocities, where gasoline leak or drip was not excessive and slipstream high. Wood in the engine section should be prohibited, if possible.

Much wider use should be made of carbon dioxide as an extinguishing agent for airplane fires. It is realized that the question of weight is involved, but it is felt that the importance of this factor has been somewhat overstressed. Certainly, so far as commercial aviation is concerned, if the question of weight is the principal factor standing in the way of the development of a more successful type of extinguishing system, it should not be lightly discarded. Mention has also been made of methyl bromide; objection to this agent has been made on the ground that it is toxic. The writer questions whether the toxic properties of this material are such as to rule it out of consideration. Methyl bromide as a fire-extinguishing agent has been used in Europe for many years. So far as its use on airplanes is concerned, the writer is inclined to think that the possibility of the formation of dangerous concentrations would be rather remote, for the reason that the infiltration of air would in all probability be such as to prevent the building up of such dangerous concentrations.

In this general connection the writer is inclined to believe that the possible dangers of carbon tetrachloride, so far as toxicity is concerned, have been considerably overstressed. A recently issued bulletin of the United States Public Health Service contains a table which compares the toxic properties of various refrigerants. This rather definitely indicates that methyl bromide is more dangerous to life than carbon tetrachloride. However, as indicated before, the writer is of the opinion that neither of these extinguishing agents involves dangers worthy of serious consideration in connection with their use on airplanes.

So far as the life hazard of CO_2 is concerned, it should be borne in mind that systems employing this material discharge in a few seconds and that the possibility of producing harmful results in so short a time is remote. The CO_2 systems of the total room-flooding type are in fairly general use throughout the country, and as such systems are installed where persons are continuously at work it is apparent that the possible danger to life involved is not considered material.

In connection with the train of operations mentioned in the paragraph immediately preceding the subject "Manifolds," the complication of the system described in this paragraph suggests the desirability of advising a word of caution with regard to giving encouragement to automatic systems involving a chain of operations, the failure of one of which might interfere with the primary object of the equipment.

In the last paragraph immediately preceding the subject "Tanks," the author states that the annealing of copper fuel lines is a recognized good practice. The writer wonders if any means of control of the extent and uniformity of annealing may be obtained. It appears to him that over-annealing may be dangerous and that failure to properly anneal may not be easily detected. In this connection it may be similarly important to look into the condition of the distributing tubing of carbon-tetrachloride and carbon-dioxide systems. An extinguisher system is of course not subject to the constant test of performing essential service as is a fuel line, and for this reason it might be well to give a word of advice as to the need for inspecting and testing the distributor lines at frequent intervals. If such inspection and test is not made, the writer anticipates that such lines may become cracked on account of embrittlement or worn through at points of contact with surfaces on which the lines are mounted. With systems of such limited capacity as those which must be provided for airplanes a relatively small crack might seriously rob the supply of extinguishing medium to critical points in the system of orifices.

DR. S. A. REED.[3] The writer, in his earlier engineering career, was for some years on work for the Fire Underwriters Boards, and finally for several years was one of three consulting engineers acting for a committee of the National Board of Fire Underwriters to investigate the conflagration hazard in all the large cities of the United States.

It seems to me that Commander McCord has made real progress toward the establishment of correct standards for fire prevention and extinction in aircraft. The writer's acquaintance with the subject has given him a great appreciation of the large share taken by automatic extinguishing means in restricting successfully the aggregate annual fire loss of the country, which it is believed Mr. Newell will substantiate. The writer therefore hopes that Commander McCord will give a favorable presumption to automatic appliances which may come up for his investigation.

[3] Aeronautical Engineer, New York, N. Y.

Commander McCord referred to the caution with which one should accept tradition and cited the effect of a blast of air, which may be favorable if strong and unfavorable if weak. The writer is encouraged to give a similar word of caution with reference to the use of water on an oil fire. It is generally said that water is wrong, and that only some extinguishing fluid or gas, or something like sand, should be used. The writer thinks that this is also a question of degree. If there are pools or ponds or reservoirs of oil or combustible fluid, then water may aggravate the situation. But the writer can say from experience that oil-smeared surfaces or conditions, even with minor leaks of oil or gasoline, are quite amenable to prompt extinction by a good souse of water, and it is believed that the universal taboo should be qualified, so that many small fires may not fail of extinction because no other means but water is present.

Author's Closure

With reference to Dr. Reed's discussion of the use of water, it should be pointed out that gases or vapors are the most convenient extinguishing agents and that in most cases, where the operator cannot reasonably expect to have ready access to the fire or be able to direct a stream of water, except for steam formed by contact with hot surfaces, water in comparison with other materials mentioned must rank low. In order to insure reaching all vital points with a liquid extinguishing medium a large quantity must be carried and the distribution manifold is necessarily complicated and heavy.

The remoteness of danger due to the use of the gases of varying degrees of toxicity, as pointed out by Mr. Newell, is probably a matter of fact, and it is impossible to establish the degree to which this danger may exist, from any figures that the author has seen regarding fire statistics.

While this is largely a point for academic discussion, it is believed that there are plenty of nonpoisonous gases, as yet untried, which would prove useful and that research along new lines is warranted.

Regarding the annealing of copper fuel lines, it is of course advisable to use furnace annealing where possible, for uniformity. This requires special apparatus and is not ordinarily possible for the private owner or the small-scale operator of aircraft who must resort to torch methods.

Proper supporting of lines and protection against the effects of vibration may save many an embrittled or overannealed pipe line.

AER-52-33

The Problem of the Propeller With Minimum Loss of Energy

By ENRICO PISTOLESI,[1] PISA, ITALY

In previous papers the author has indicated the necessity of a thorough study of propeller problems, and has indicated the basis upon which such a study should be established. In his present contribution he undertakes to determine the distribution of velocity in an infinitely distant section of the slipstream which, for a given thrust, will result in a minimum loss of energy.

IN A PAPER by the author in 1928,[2] he indicated the necessity of a thorough study of propeller problems, keeping in mind the partial vacuum which is formed within the slipstream because of the effect of the tangential velocity of the fluid particles. Before that, in another paper[3] he indicated the basis upon which such a study would have to be established. Recently, M. Roy has published a discussion of the subject.[4] In this he considers what happens in the section which is at an infinite distance in the slipstream of the propeller. Among the problems to which he refers, but does not solve, there is that of the minimum loss of energy, a problem which has already been studied by Betz,[5] although he does not take into consideration the partial vacuum of the slipstream.

In the present paper it is assumed that the slipstream is entirely influenced by the propeller. This avoids the case, although an interesting one, considered by Roy, in which the active part of the disk area of the propeller is limited to an annulus at the rim. The present research will consist of determining the distribution of velocity in an infinitely distant section of the slipstream which, for a given thrust, results in the minimum loss of energy.

1 Let r indicate generally the radius of a point in the slipstream in the aforementioned section, V_0 the original velocity of the current, v the axial increment, ωr the tangential or circumferential velocity, Ω the rotational velocity of the propeller, p the pressure at a point of the section, p_0 the pressure of the undisturbed fluid, and R the radius of the section It is assumed that the propeller has infinite blades so that v, ω, and p will be only functions of r.

Since in the section under consideration the radial velocity is zero, the following relation[6] obtains:

$$p - p_0 = \rho r^2\omega\left(\Omega - \frac{\omega}{2}\right) - \rho v\left(V_0 + \frac{v}{2}\right) \quad [1]$$

Also,[7] for the pressure, we have the differential relation:

$$\frac{dp}{dr} = \rho\omega^2 r \quad [2]$$

Differentiating Equation [1], we have:

$$\frac{dp}{dr} = \rho\,\frac{d(r^2\omega)}{dr}\left(\Omega - \frac{\omega}{2}\right) - \frac{1}{2}\rho r^2\omega\,\frac{d\omega}{dr} - \rho(V_0 + v)dv$$

Eliminating dp/dr by means of Equation [2], we obtain:

$$(\Omega - \omega)d(r^2\omega) = (V_0 + v)dv \quad [3]$$

This is the fundamental differential relation which binds the increments to one another and with the radius r. It is easy to prove that this equation is identical with the one given by Roy in the aforementioned work, in which if we consider $\omega r = u$, we have:

$$(\Omega r - u)\left(\frac{u}{r} + \frac{du}{dr}\right) = (V_0 + v)\,\frac{dv}{dr}$$

The equation of thrust T is derived from the application of the theorem of momentum, taking into account the underpressure in the slipstream:

$$T = \int_0^R 2\pi r\rho(V_0 + v)v\,dr + \int_0^R 2\pi r(p - p_0)dr \quad [4]$$

Analogously, the total energy communicated to the fluid flowing into the slipstream in a unit of time is given by

$$E = \int_0^R 2\pi r(V_0 + v)(p - p_0 + \frac{1}{2}\rho(V_0 + v)^2 + \frac{1}{2}\rho r^2\omega^2 - \frac{1}{2}\rho V_0^2)dr \quad [5]$$

Eliminating $p - p_0$ from Equations [4] and [5] by means of Equation [1], we obtain:

$$T = \rho\int_0^R 2\pi r\left[r^2\omega\left(\Omega - \frac{\omega}{2}\right) + \frac{v^2}{2}\right]dr \quad [6]$$

$$E = \rho\int_0^R 2\pi r(V_0 + v)r^2\Omega\omega\,dr \quad [7]$$

To this last formula, moreover, we can arrive by the calculation of the couple Q to be applied to the shaft of the propeller, taking as a basis the theorem of the moment of momentum:

$$Q = \int_0^R 2\pi r\rho(V_0 + v)r^2\omega\,dr \quad [7a]$$

since manifestly $E = Q\Omega$.

Equations [6] and [7] are also found in Roy's contribution.

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[1] Professor of Engineering, Royal School of Engineering.

[2] "Problemi di Eliche" (Propeller Problems), *L'Aerotecnica*, vol. 8, no. 10, 1928.

[3] "Un Problema Fondamentale per la Teoria delle Eliche" (A Fundamental Problem for the Theory of the Propeller), *L'Aerotecnica*, 1927, no. 6.

[4] "Contribution a la Théorie de l'Helice Propulsive" (Contribution to the Theory of the Propeller Screw), Travaux du Cercle d'Études Aérotechniques, fasc. 3, 1929.

[5] "Schraubenpropeller mit geringstem Energieverlust" (Screw Propeller with Small Loss of Energy), *Nachrichten* von der K. W. Gesellschaft der Wissenschaften zu Göttingen, 1929.

[6] See Pistolesi: "Le Equazioni Differenziali del Moto dei Fluidi Applicate al Campo di Velocita Prodotto dall'Elica" (The Differential Equations of the Motion of the Fluid Applicable to the Field of Velocity Produced by the Propeller), Report of R. Accad. dei Lincei, July, 1922.

Presented at the Fourth National Aeronautic Meeting, Dayton, Ohio, May 19 to 22, 1930, of The American Society of Mechanical Engineers.

[7] See Pistolesi: "Lezioni sulle Eliche" (Lessons on Propellers), R. Politecnico di Torino, Perotti, 1924.

2 It is now necessary to determine the minimum of E, or of Q, given T, which is a problem in the calculus of variations. We can consider the function under the integral signs to be functions of v or of ω, v and ω being connected by the differential relation [3]. It is necessary, however, to make an exchange of variables, placing

$$r^2\omega = y \quad [8]$$

Equation [3] then becomes

$$\left(\Omega - \frac{y}{r^2}\right) dy = (V_0 + v)dv \quad [9]$$

Equation [6],

$$T = \rho \int_0^R 2\pi r \left[y\left(\Omega - \frac{y}{2r^2}\right) + \frac{v^2}{2} \right] dr \quad [10]$$

and Equation [7a],

$$Q = \rho \int_0^R 2\pi r(V_0 + v)y dr \quad [11]$$

Indicating by $-K$ Lagrange's factor, we have:

$$\frac{d}{dy}\left\{ V_0 + v)y - K\left[y\left(\Omega - \frac{y}{2r^2}\right) + \frac{v^2}{2}\right]\right\} = 0$$

or

$$\frac{d\,[(V_0 + v)y]}{dy} - K\left(\Omega - \frac{y}{r^2} + v\frac{dv}{dy}\right) = 0$$

and from [9],

$$\frac{d}{dy}\,[(V_0 + v)y] = K(V_0 + 2v)\frac{dv}{dy}$$

Integrating:

$$(V_0 + v)y = C + {}^1/_4K(V_0 + 2v)^2$$

an expression that, including the term ${}^1/_4KV^2$ in the constant C, can be easily transformed into:

$$(V_0 + v)y = C + K(V_0 + v)v \quad [12]$$

a very simple relation that connects the two increments.

Equation [12] may also be written:

$$y = Kv + \frac{C}{V_0 + v} \quad [13]$$

Differentiating [13] and comparing the value of dy/dv deduced from it with that given by [9], we obtain

$$\frac{V_0 + v}{\Omega - \omega} = K - \frac{C}{(V_0 + v)^2} \quad [14]$$

Equations [12] and [14] permit us to determine v and ω in functions of r.

Because of this we put $r^2\omega$ in place of y in [12] and keep the auxiliary unknown $V = V_0 + v$. With this, [12] and [14] become respectively:

$$Vr^2\omega = C + KV^2 - KV_0V \quad [12a]$$

$$\frac{V}{\Omega - \omega} = K - \frac{C}{V^2} \quad [14a]$$

We deduce now for V an equation of the fourth power,

$$V^4(K^2 + r^2) - KV^3(KV_0 + r^2\Omega) + CV(KV_0 + r^2\Omega) - C^2 = 0 \quad [15]$$

The constants K and C are determined on the basis of conditions at the boundary of the slipstream, through Equation [1], in which we get $p - p_0 = 0$, and assign either the value v_1 or ω_1 of one of the two increments for $r = R$, or the value of T, from which v_1 and ω_1 may be later derived.

3 The solution and discussion of Equation [15] is a rather difficult and complicated matter.

For brevity, we shall limit ourselves to two cases. The first is one in which $V_0 = 0$ *for a fixed propeller*. Then $V = v$, and Equation [15] becomes:

$$v^4(K^2 + r^2) - Kv^3r^2\Omega + Cvr^2\Omega - C^2 = 0 \quad [16]$$

which cannot be materially simplified. If we put

$$\xi = \frac{2}{K}, \qquad \nu = \frac{v}{K\Omega} = \xi\frac{v}{r\Omega}, \qquad \alpha = \frac{C}{K^3\Omega^2} \quad [17]$$

ξ, ν, and α are dimensionless ratios and we shall have:

$$\nu^4(1 + \xi^2) - \nu^3\xi^2 + \alpha\nu \mp \alpha^2 = 0 \quad [18]$$

This equation contains only one arbitrary constant α in the variation of which there are quite possible infinite types of distribution of ν in the function of ξ. These possible types have an extreme case corresponding to $\alpha = 0$, for which following equation holds:

$$\nu^4(1 + \xi^2) = \nu^3\xi^2$$

from which, eliminating the triple solution $\nu = 0$, as lacking significance:

$$\nu = \frac{\xi^2}{1 + \xi^2} \quad [19]$$

Further, from the second one of Equation [15] we have:

$$\frac{\omega}{\Omega} = 1 - \frac{\nu^3}{\nu^2 - \alpha} \quad [20]$$

which in the case where $\alpha = 0$ gives

$$\frac{\omega}{\Omega} = 1 - \nu = \frac{1}{1 + \xi^2} \quad [21]$$

The formulas [19] and [20] can be interpreted geometrically, placing $\xi = r/K = \tan\varphi$. From this it follows that

$$\nu = \sin^2\varphi; \frac{\omega}{\Omega} = \cos^2\varphi$$

In the other cases the curve of ν may be traced in functions of α for each value of ξ, solving Equation [18] with respect to ξ[3]

$$\xi^2 = \frac{\nu^3 + \alpha\nu - \alpha^2}{\nu^3(1 - \nu)} \quad [18a]$$

From Equation [20] we then obtain the value of ω/Ω.

4 The other limiting case is when v is very small in relation to V_0. Equation [12] may then be simplified as follows:

$$V_0r^2\omega = C + KV_0 \quad [22]$$

from which

$$r^2\omega = \frac{C}{V_0} + Kv$$

Differentiating and substituting in [3],

$$K(\Omega - \omega) = V_0 + v$$

$$r^2K\Omega - K\frac{C}{v_0} - K^2v = r^2V_0 + r^2v$$

[3] This may also be done with [15].

and finally:

$$v = \frac{r^2(K\Omega - V_0) - KC/V_0}{r^2 + K^2} \quad \ldots\ldots [22]$$

$$\omega = \frac{K(K\Omega - V_0) + C/V_0}{r^2 + K^2} \quad \ldots\ldots [23]$$

If we put:

$$\xi = \frac{r}{R},\ c^2 = \frac{K^2}{R^2},\ a = \frac{K\Omega}{V_0} - 1,\ b = -\frac{C}{KV_0^2},$$

$$\nu = \frac{v}{V_0},\ \text{and}\ \mu = \frac{K\omega}{V_0}$$

we have:

$$\nu = \frac{a\xi^2 + bc^2}{\xi^2 + c^2} \quad \ldots\ldots [22a]$$

$$\mu = \frac{c^2(a - b)}{\xi^2 + c} \quad \ldots\ldots [23a]$$

Bearing in mind Equation [1], where r is put equal to R (whence $\xi = 1$), $p - p_0 = 0$, and where v and ω may be disregarded in the presence of V_0 and Ω, we then obtain easily between a, b, c, and $v = V/R\Omega$ the relation:

$$\gamma(a + bc^2) = c(a - b) \quad \ldots\ldots [24]$$

Moreover it is easy to verify between a, b, and c the relation

$$c = \gamma(1 + a) \quad \ldots\ldots [25]$$

which permits us to substitute to [24] the following formula:

$$b = \frac{a^2}{c^2 + a} \quad \ldots\ldots [26]$$

The following relation, moreover, is true:

$$\nu + \mu = a$$

and Equation [22a] may accordingly be written:

$$\nu(\xi^2 + c^2) = a\xi^2 + bc^2 \quad \ldots\ldots [22b]$$

It is not without interest to compare the preceding relation with one which is derived upon the basis of the same hypothesis that v is negligible compared with V_0, whenever the underpressure in the slipstream is not considered.[9]

$$\nu(\xi^2 + r^2) = K\xi^2$$

The two expressions differ because of the substitution of c to γ in the first member and because of the addition of the constant bc^2 in the second. Putting, as in the aforementioned reference,

$$\frac{\xi^2}{\xi^2 + c^2} = \cos^2\varphi, \text{ whence } \frac{c^2}{\xi^2 + c^2} = \sin^2\varphi$$

Equation [22a] becomes:

$$\left.\begin{aligned} \nu &= a\cos^2\varphi + b\sin^2\varphi \\ \mu &= (a - b)\sin^2\varphi \end{aligned}\right\} \quad \ldots\ldots [26]$$

For $\xi = 1$, we have:

$$\tan\varphi_1 = c$$

If we place $\tan\varphi_2 = cb/a$ and $\gamma = \tan\mu$, it follows from [24] that

$$\mu = \varphi_1 - \varphi_2$$

[9] See E. Pistolesi, "I Propulsori Elicoidali e i Recenti Progressi dell'Aerodinamica" (The Screw Propeller and Recent Progress in Aerodynamics). Report of Ist. Sperim. Aeronaut., vol. 8, p. 235.

which expresses in a more suggestive manner the relation between the values which rule in the problem.[10]

5 Let us now calculate T and E by means of Equations [6] and [7] upon the basis of the formulas in the preceding paragraph, neglecting, in accordance with the hypothesis which has been assumed, the square of the incremental velocity; or

$$T = 2\pi\rho\int_0^R r^3\Omega\omega dr \quad \ldots\ldots [6a]$$

$$E = 2\pi\rho\int_0^R r^3\Omega\omega V_0 dr \quad \ldots\ldots [7a]$$

We find that $E = TV_0$ (whence $\eta = 1$, as a natural consequence of the procedure) and moreover:

$$\tau = \frac{T}{\rho R^4\Omega^2} = \pi\gamma c(a - b)\left[1 + c^2\log\frac{c^2}{1 + c^2}\right] \quad \ldots\ldots [27]$$

or, from [24]:

$$\tau = \pi\gamma^2(a + bc^2)\left[1 + c^2\log\frac{c^2}{1 + c^2}\right] \quad \ldots\ldots [27a]$$

or also, taking note of the fact that a, b and c are connected by Equation [26]

$$\tau = \pi\gamma^2 a\left(1 + \frac{ac^2}{a + c^2}\right)\left[1 + c^2\log\frac{c^2}{1 + c^2}\right] \quad \ldots\ldots [27b]$$

The calculation of τ, in case that the underpressure is not taken into account, assuming that the law of the distribution of the increment v is unchanged, is effected upon the basis of the formula:

$$T = \int_0^R 2\pi\rho r V_0 v dr$$

and leads to the following results:

$$\tau' = \pi\gamma^2 a\left[1 + c^2\log\frac{c^2}{1 + c^2}\right] - \pi\gamma^2 bc^2\log\frac{c^2}{1 + c^2}$$

The difference between the two results should be noted.

Equation [27b] may be transformed by dividing both members by $\pi\gamma^2$, and as a matter of fact $\frac{\tau}{\pi\gamma^2} = \frac{T}{\rho\pi R^2 V_0^2}$ is the coefficient of thrust, referred to the area of the section (which, in the given hypothesis, is equal to the disk area) and to the square of the velocity. We then have:

$$\frac{\tau}{\pi\gamma^2} = a\left(1 + \frac{ac^2}{a + c^2}\right)\left[1 + c^2\log\frac{c^2}{1 + c^2}\right] \quad \ldots\ldots [27c]$$

Note that in the hypothesis ν = a constant, and always neglecting the underpressure, we simply find that

$$\nu = \frac{\tau}{\pi\gamma^2}$$

This is to be noted for the comparison.

It is scarcely necessary to state that the expression [27], combined with [25] and [26], permits the determination of the values a, b, and c, τ and γ being known.

For brevity's sake the many conclusions which might be drawn from the results obtained are omitted.

[10] Instead of the ratio μ, one can make use of u/V_0, where $u = r\omega$, giving

$$\frac{u}{V_0} = \frac{c(a - b)\xi}{\xi^2 + c^2} = (a - b)\sin\varphi\cos\varphi$$

AER-52-34

Flying Boat Design

By ADOLF K. ROHRBACH, BERLIN, GERMANY

This paper tells of some of the problems encountered in designing flying boats, and compares their characteristics with those of landplanes. Five outstanding considerations include: the size of boat, aerodynamic qualities, floating system, structural principles, and general arrangement. The best shape for a flying-boat hull must be determined by experiment.

PROBLEMS encountered in the design of flying boats are in some respects rather different from those pertaining to the design of land machines. Whereas land machines operate from solid fields, flying boats have to contend with a water surface, the condition of which depends very much on the weather and on the place. The reaction between boat and water in turn depends very much on the shape of the boat.

In analyzing the design of a flying boat, there are five main problems: the size of the boat, the aerodynamic qualities, the floating system, the structural principles, and the general arrangement, which combines the best solutions for the first four problems.

Size of Boat. As far as the ground operation of landplanes is concerned there is practically no advantage in a bigger machine over a smaller one, whereas bigger flying boats, like bigger surface vessels, are much more stable in rough seas. In flight a big commercial flying boat of 20,000 lb. and over is considerably more economical than a landplane of equal weight. It is an established fact that the weight of the flying structure of large machines is a greater portion of the gross weight than of small ones. This is particularly so if heavy planes are built with relatively large wings and small wing loadings, because much weight or aerodynamical resistance is required for these widely stretched wings and corresponding fuselage and tail-plane structures.

On account of the restricted size of landing fields and the high ceiling needed to fly over mountains, big landplanes must have relatively small wing loadings and landing speeds. The result is that for commercial landplanes the greatest efficiency seems to be reached with a size of about 10 to 12 tons. Beyond this size the increased weight of the structure becomes more and more pronounced, and counteracts the payload carrying capacity. In the case of the flying boat, the effect of increased size is less pronounced. It is admitted today by many authorities that the landing speed of large flying boats within reason can be increased over that which is good practice in small seaplanes. The reason is that a certain wave representing a certain weight of water means less to a large heavy boat than to a small light seaplane.

This increase of the landing speed is about 20 to 25 per cent in a big flying boat, and means a relatively smaller span, smaller length of tail, and smaller area of tail planes. This more concentrated flying structure is so much lighter than that of a landplane of the same gross weight would have to be, that the most economical sizes for commercial flying boats today have a gross weight of about 20 to 25 tons, which is more than twice that of the most economical landplane. The *Inflexible*, a giant bomber which was designed for the British Air Ministry, has about the same wing area as the *Romar* flying boat, although the weight of the former is 15 tons against a weight of 19.5 tons for the latter. The span of the *Inflexible* is 150 ft. and of the *Romar* but 120 ft. In accordance with these mechanical laws and weight proportions, the heaviest landplanes in existence have a gross weight of about 15 tons, whereas the heaviest flying boat, the Dornier Do-X, has flown with a weight of 48 tons, which was taken off after a run of 160 sec. Though the Do-X has not yet passed its development period, its weight empty of about 35 tons with equipment is 73 per cent of the 48 tons, whereas the *Romar*, with a gross weight of 19.5 tons including equipment, has a weight empty of 11.2 tons, or 57.5 per cent of the gross weight. These figures prove that the Do-X has already passed the most economical size from a weight standpoint.

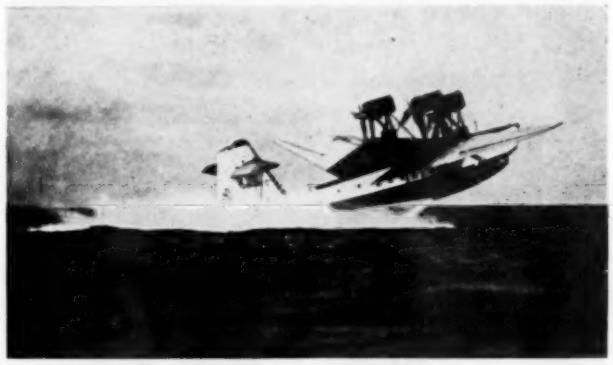


Fig. 1 The Flying Boat Romar

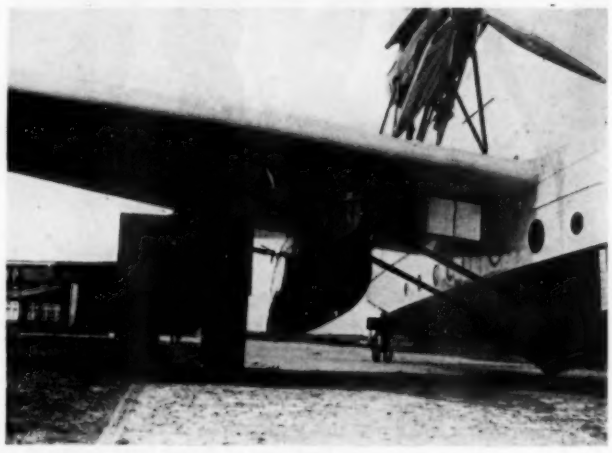


Fig. 2 Method of Attaching Floats

As far as the cost of construction is concerned it is hard to say how it will influence the most economical size. The greater number of machines that can be sold in smaller sizes reduces their cost, whereas the building cost of a big machine compared to its carrying capacity is relatively smaller for the same reasons which in other fields reduce the cost of one big unit against that of a combination of small units of equal total power. The heavier

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

wing loading of a big flying boat reduces somewhat its speed of climb, but a high ceiling is not needed and even a heavily loaded four-motored flying boat has enough margin of power to fly for any length of time with one motor dead.

In regard to the emergencies of forced landings, a big landplane is usually much worse off than a smaller one, which can make forced landings relatively easy, and if it should not be able to take off again without much trouble can be transported to some flying field. The opposite is true of flying boats which cannot count on outside help as quickly as landplanes. For this reason it is very important that the flying boat be self-contained, and it is obvious that the larger machine, with its greater seaworthiness, with its spacious rooms, and more numerous crew, certainly offers a great



FIG. 3 METHOD OF MOUNTING ENGINES

advantage over a small machine with a crew of two. It is understood that if the technically best size of flying boat today is 22 to 25 tons, it will steadily be increased by every improvement that is contributed to the art.

The *Romar*, with the experience of today, powered by four Hornet motors, will with a weight of 19.5 tons carry up to 45 passengers over a distance of a few hundred miles and several thousand pounds of payload over a distance of 2000 miles, allowing a 25 per cent distance reserve for the longest leg of the Europe-South America route. Within two or three years it will be possible to carry a payload of several thousand pounds over a distance of 3000 miles allowing the necessary 50 per cent distance reserve for the flight between the Azores to Bermuda, where strong headwinds are encountered over the greater part of the way.

Aerodynamics. The monoplane type of flying boat, especially in large sizes, is preferable because of greater speed and greater simplicity. The aspect ratio of monoplane flying-boat wings varies in different types from 1:5 to 1:10. The wings of some types of flying boats are rectangular. Tapered wings, though more costly to build, have the advantage of lighter weight and smaller rolling moments in a side wind while the machine is floating on the water. The double tapered wing has a greater ratio of thickness to chord at the root than at the tip, and has least weight and sometimes smaller air resistance through the thinning out of the wing profiles near the tip. It is important that the wing profiles be designed so that there is a minimum variation of aerodynamic moment or, as some put it, a minimum movement of center of pressure for various angles of incidence. Obviously the smaller this variation of aerodynamic moment the smaller are the counter moments that are to be provided by the tail planes. This in turn means smaller tail surfaces and a shorter tail for a certain controllability at a certain speed.

Flying boats, in order to be safe at stalled landings in heavy seas, should have a very good controllability at lowest speeds. The moments that can be exercised with full elevator or full rudder must be considerably greater than those needed to counteract the aerodynamic moments of the wing and than those needed for land machines. Therefore the smaller the aerodynamic moment of the wing, the greater the controllability that can be obtained with a certain area of tail planes and length of tail, and correspondingly with a certain weight expended to build these parts.

It has been found necessary to balance all control surfaces of big boats. The most satisfactory method up to now has been to place the axle about 15 to 22 per cent of the chord length behind the leading edge. Rather large vertical fins are necessary in order to provide longitudinal stability in spite of the great portion of the boat that is in front of the wing. A dihedral angle of from 4 to 8 deg. is important in order to have good lateral stability at stalling speed. If all the before-mentioned aerodynamic conditions are fulfilled, the flying qualities of a big flying boat are as perfect as those of the best training planes. The *Romar*, for instance, is stable in all three directions. Curves up to a bank of 70 deg. can be flown with the rudder alone. The machine recovers from a sideslip if controls are not touched, and it has been reported that during the last flights of 12 and 17 hours' duration the controls remained practically untouched during the entire time.

Floating System. The floating system has to fulfil the following conditions: Flotation at rest must be longitudinally and laterally stable under all possible conditions of wind and load. At moderate speeds on the water the flying boat must be as perfect a surface vessel as possible; there must be good controllability, and not too much spray must be thrown up in a rough sea with a strong side wind. At high speeds close to take-off speed, the running must be free from pounding, and at all speeds the resistance should be the minimum possibly attainable.

A flying boat with a hull sufficiently long to support the tail planes has automatically enough longitudinal floating stability. In order to obtain lateral stability it has been found best to have two side floats of substantial size, not too far away from the hull and well immerged at rest. The lateral stability of this arrangement is so that a side wind of 30 m.p.h. will not force the leeward wing into the water. Flying boats of this type have been on the water for many hours in gales up to 55 m.p.h. without any special precaution. It is believed that the boats always will be able to ride out gales of similar force floating behind a drift anchor. The high bow, which cannot cut under waves, is of extreme importance in these conditions.

At taxiing speeds of about 15 m.p.h. the side floats give the greatest hydrodynamical lift, aiding the boat to plane at about 25 m.p.h., and during planing the side floats are entirely clear of the water. In order to avoid shocks during planing, the bottom

has a pronounced V of about 27 deg. medium angle. The outer part of the bottom V is curved so that the water is thrown downward with a corresponding increase in lift. In front of the main step the keel is widened to a narrow gliding surface with hollow sections which firmly support the planing boat shortly before take-off. The V shape is so steep that, in spite of the curves of the outer part of the bottom, the boat does not pound in rough seas. The bottom aft of the main step has straight V sections until near the second step where it is shaped somewhat similar to the main step, though less pronounced, in accordance with the smaller hydrodynamic forces acting on the second step. Aft of the second step, the bottom has a straight V rising sharply in order to be free from the water at all speeds.

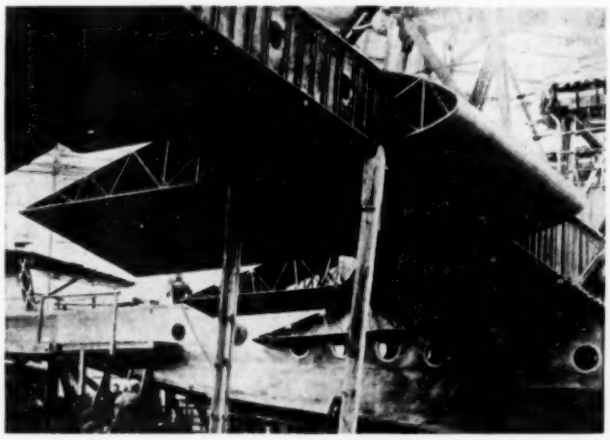


FIG. 4 BOX-SPAR WING GIRDER AND NOSE

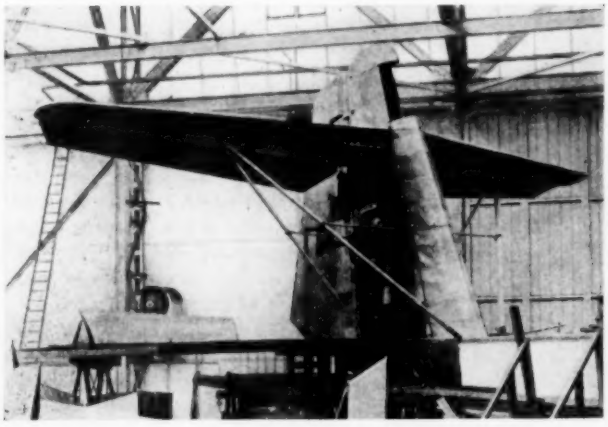


FIG. 5 CONSTRUCTION OF TAIL UNIT

The hydrodynamical problems to be solved in order to produce a flying-boat bottom that will plane with the smallest resistance and greatest lift are very similar to those of water turbines where a maximum torque obtained from the flowing water is the object. Therefore no genius can design the best shape of a flying boat hull, but this will be found by experiments alone. The company with which the author is connected has carried out more than 2000 towing tests in a model basin, running several hundred models over a total distance of more than 1000 miles. The results obtained by these model tests have been checked by hundreds of actual take-off tests with several types of flying boats. The take-off conditions of a new type can, on this experience, be predicted with an accuracy of 5 to 8 per cent, regarding weight. Though some progress has been made there is still much to perfect.

The very deep sharp V and the narrow gliding surface in front of the main step have the advantage that the boat is planing in a very high position; that means about 4 ft. higher than the floating position at rest. It is obvious that this high planing position offers excellent protection for wings and all other parts of the structure when taking off in a rough sea. So far take-offs have been made in waves of 12 to 15 ft. On the other hand, the small water resistance of this shape of hull has resulted in a good take-off from a calm sea with maximum load, which is not astonishing considering that all the towing tests have been made on smooth water.

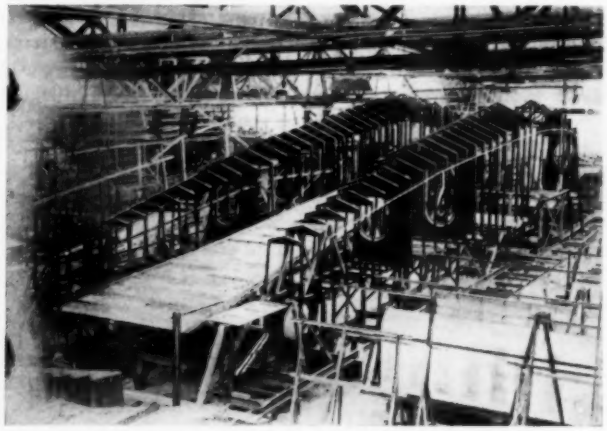


FIG. 6 SKELETON HULL SHOWING SHIPBUILDING METHOD OF CONSTRUCTION

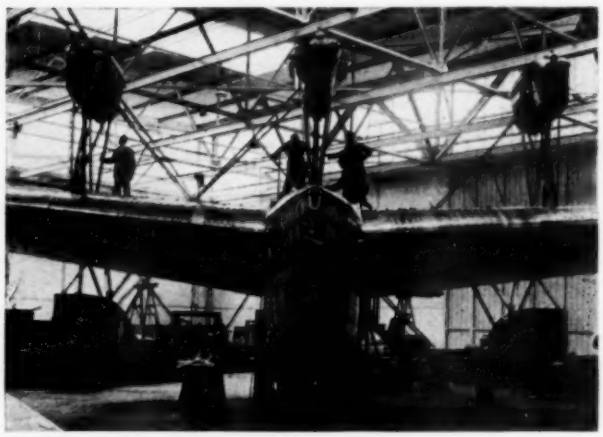


FIG. 7 THICKNESS OF CHORD OF WING AT ROOT AND V-BOTTOM

Structural Principles. Materials for flying boats, aside from workability, must have a good ratio of strength to weight, and unlike the materials of land machines their qualities must not be changed by the influence of seawater. Therefore in the course of further development wooden structures and canvas covering of wings should be replaced by duralumin sheet construction. Great care must be taken to avoid corrosion by use of suitable materials, working methods, and protective means, such as coats of other metals, paint, grease, etc. All structural details should be of so simple design and built of so many separate parts that everything can be easily inspected and maintained from all sides.

Watertightness forms the best anticorrosive protection for the

inside of many parts, since it is the only means to exclude all corroding moisture. It is mostly for this reason that the duralumin wing-box girders of later flying boats are watertight over the greater part of the span. As a secondary result it may be of interest that the buoyancy of this wing-box girder alone could keep the machine afloat indefinitely.

In big duralumin flying boats, the flat skin plates of wing and hull, by means of stiffening profiles, should take a substantial part of the stress. Very much weight and labor would be needed, however, to provide stiffening profiles so close together that considerable compression or bending forces could be transmitted by the skin. Therefore the structural members of all parts are arranged so that profile girders take the compression stress, whereas the skin plates transmit the shearing forces by means of rather widely spaced stiffeners. The shearing stress of a flat skin is much greater than that of a curved one of equal weight. Therefore, and because of cheaper construction, flying boats have flat sides and decks, and it is believed that this type of construction will be universally adopted. The mechanical qualities of skin plates under shearing stress have been systematically studied by numerous experiments.

The weight prediction for a new type is correct today within 5 per cent. The whole development of the flying-boat design in about one year has reached the stage where for a given design one can predict with great accuracy not only its flying performance but also what it will do on the water. This means that flying boats can now be designed with certain predetermined and commercially desirable qualities and performances. This has brought us only to the beginning of the real development in flying-boat design which will be based on their transportation utility.

AER-52-35

The Artificial Horizon

By ELMER A. SPERRY, JR.,[1] BROOKLYN, N. Y.

This paper describes the "artificial horizon," an instrument developed recently to aid blind flying. Through the use of a gyroscope a gravity device is constructed with a very slow rate of motion that will maintain a horizontal position in an airplane with sufficient accuracy to make a most useful instrument. Test data for the instrument are given.

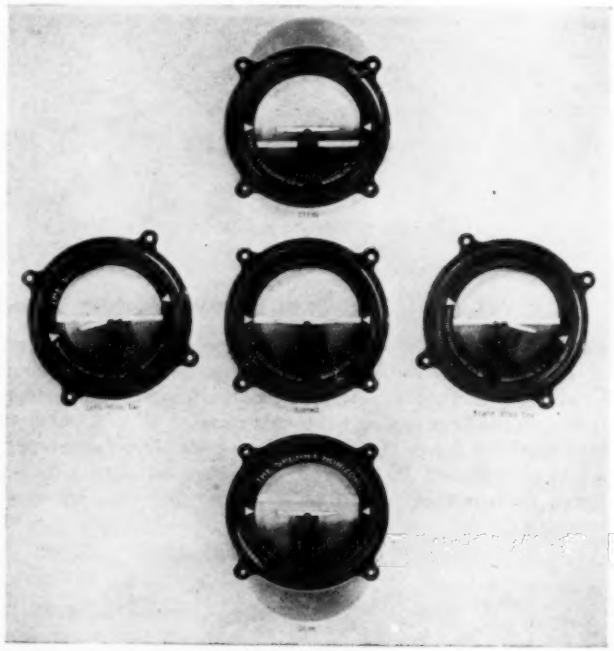
THERE has been a long-felt need for an improved instrument to aid blind flying, an instrument to show at a glance the true horizontality of the airplane at all times. Flying blind on a straight course for any length of time by instruments is a difficult and very tedious job. Making quick turns while flying blind and immediately resuming normal straight flight is practically impossible with the instruments of today.

Up to the present no instrument shows the true horizontality of the airplane at all times, but merely gives an indication of something else from which the horizontality must be determined by the pilot. The altimeter shows when the plane gains or loses altitude. The rate of climb shows when the plane climbs or descends. The air-speed and tachometer indications decrease when the plane climbs and increase when it dives. But these instruments do not immediately indicate a nosing up or down of the plane. Their inherent lag makes it difficult for a pilot to fly blind by them without oscillating or pitching.

The turn indicator and bank indicator are used for determining lateral horizontality while flying blind as well as indicating yaw and proper bank. However, the pilot must translate the indications of these instruments in order to determine his lateral position, and it takes a great proficiency to interpret the "language" of these instruments before a pilot feels at ease while flying blind by them.

Gravity devices such as a pendulum or bubble level behave quite well in an airplane when flying a steady, straight course through smooth, clear air. In bumpy air an undamped gravity device will oscillate badly, and during maneuvers of the airplane will respond to acceleration or centrifugal forces. Gravity devices damped by being connected directly to the airplane as by a dashpot or a viscous, restricted-flow liquid will not oscillate as much. To some extent, ship's movements will be imparted to the device through the damping means. Gravity devices that are damped in a different manner, such as two pendulums of different periods or a pendulum and a balanced mass, connected together in some manner which dissipates energy, or a continuous-circuit viscous-liquid bubble, will not be energized as much by ship's motion but will respond to acceleration and centrifugal forces quite readily.

These pendulous devices for aircraft can be divided roughly into three classes: Those having quite a rate or period of motion, those having a relatively slower rate of motion by reason of being well damped, and those designed to have a very slow rate of motion of only a few degrees per minute which are aperiodic. Instruments in the first class respond to every movement of the plane and oscillate badly. These might be called bump indicators and are not satisfactory for aircraft use. Those in the second class do not oscillate as much, though they still respond to centrifugal forces readily. Accelerometers, banking indicators, and bubble inclinometers are in this class and serve useful purposes. Those in the third class do not oscillate, and respond to centrifugal forces only according to their rate and the duration of the forces. There are only a few instruments of this class in use today.



THE HORIZON INDICATOR FOR VARIOUS POSITIONS OF THE FLIGHT OF THE PLANE

Instruments of the third class can be made to maintain the vertical on the airplane with a surprisingly high degree of accuracy due to the fact that centrifugal or acceleration forces are of comparatively short duration when undirectional as, for instance, when the airplane increases or decreases air speed. When centrifugal forces exist for any considerable length of time, such as when an airplane continues to turn in circles, they are in the same direction with respect to the airplane, that is, athwartship, but are not in the same direction with respect to the earth. Therefore, the mean effective centrifugal force with respect to any particular direction in azimuth equals zero for a complete turn. The greatest mean-effective centrifugal force with respect to any direction, therefore, is for a turn of 180 deg. If the turn is less than 180 deg. the mean-effective centrifugal force will

[1] Sperry Gyroscope Company.

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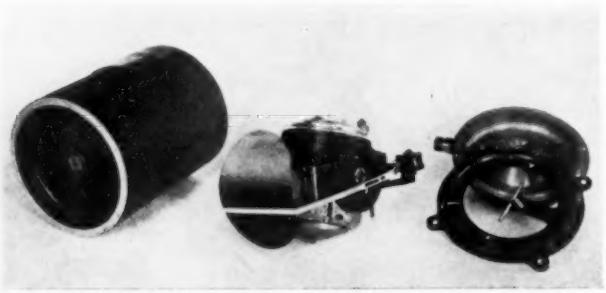
NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

be less, and if the turn is greater, that part which is over 180 deg. will cancel an equal amount diametrically opposite, making the total effect again smaller.

The mean centrifugal force acts for the entire time of the turn. For instance, an airplane flying at 100 m.p.h. makes a 180 deg. turn in 30 sec., banking 25½ deg. The centrifugal force at every moment during the turn is 0.43 lb. on every pound in the airplane, exerted athwartship or radially to the turn. The mean centrifugal force for the entire turn acting in one direction is 0.265 lb. acting for 30 sec.

If a pendulous device of class 3 had a rate of response of 1 deg. in 30 sec., it would only depart 1 deg. from the true vertical during the above turn, whereas a pendulous device of class 2 would depart 25½ deg. during the turn, as the banking indicator does. If the airplane made a complete circle instead of only a 180 deg. turn, the slow-responding pendulous device (of class 3) would go off only a small amount during the turn, but would be left correct again at the end of the turn.

It has been found possible by using a gyroscope to construct a class 3 gravity device of slow rate that will maintain a horizontal position in an airplane with sufficient accuracy to make a



THE INTERNAL PARTS OF THE ARTIFICIAL HORIZON

most useful and perfect blind-flying instrument. An instrument of this kind was constructed in such a manner that its rate could be adjusted to meet normal operating conditions. This was installed on the Guggenheim Fund Consolidated N. Y. 2 and flown by Lieutenant James H. Doolittle during his blind flying experiments at Mitchell Field. Test data from an instrument adjusted for a rate of approximately 4 deg. per min. are given herewith.

TEST RESULTS OF INSTRUMENT

90 m.p.h. air speed average Turn	Bank	Time for 180 deg. turn	Center of pointer	Right end of pointer	Recovery
180 deg. left	40 deg.	15 sec.	Held	Up 1/32	Immediate
180 deg. left	25 deg.	28 sec.	Down 1/32	On O. K.	Immediate
180 deg. left	17 deg.	43 sec.	Down 1/32	Up 1/32	Immediate
180 deg. left	7 deg.	1 min. 45 sec.	Down 1/16	Up 1/16	Immediate
180 deg. right	36 deg.	18 sec.	Held	Up 1/32	Immediate
180 deg. right	29 deg.	26 sec.	Held	Up 1/32	Immediate
180 deg. right	14 deg.	50 sec.	Down 1/32	Up 1/32	Immediate
180 deg. right	6½ deg.	1 min. 54 sec.	Down 1/16	Down 1/32	Immediate

2 min. straight course held between turns to allow instrument to settle.
Climb at 10 deg. for 5 min. after leveling off, pointer of indicator O. K.
Dive at 15 deg. for 4 min. after leveling off, pointer of indicator O. K.
Vibration, pointer, none.
Vibration, instrument board, slight.

This new horizon has a small circular face of about 4 in. diameter constructed for flush mounting in the instrument board. The face of the instrument consists of a smooth background, blue on the upper half to denote sky, shading to a dark grey on the lower half to represent ground. Horizontally across the middle of this field is a straight bar, both ends of which extend beyond the mask which surrounds the instrument face. This bar simulates the horizon. In front of the bar and the field is a small tail view of an airplane held at the center of the instrument face as part of the mask which tilts with the plane. If the plane climbs or noses down, the horizon-bar falls or rises respectively in just the same manner as the actual horizon appears to fall or rise as the pilot looks over the nose of his ship. As the plane banks, the bar remains horizontal, whereas the dial and the airplane silhouette tilt with the plane. The horizon bar is thus free to rise, fall, and tilt, and can assume any possible combination of these positions. The bar is held in the horizontal position by a small air-spun gyroscope mounted in a gimbal ring, and having its spinning axis vertical.

For a gyroscope spinning in this manner to maintain a true vertical axis, it is necessary to have gravity act on it in some manner, and this very fact has been the stumbling block of many early artificial horizon designs, because when gravity acts directly on a gyroscope (for example, by making it decidedly pendulous in its gimbal mounting) the gyroscope precesses at right angles to the tilt, and therefore starts an oscillation of slow period which has to be damped out before the axis can recover the vertical. In such a case, any maneuvering of the airplane or change of its course causes an acceleration pressure on the pendulous gyro which throws it off of the true vertical and starts an oscillation. In this horizon instrument the effect and the resulting oscillation has been entirely avoided by making gravity act indirectly on the gyroscope so that the gyroscope precesses in the correct direction from any tilted position directly to the vertical without passing through any period of oscillation. This is accomplished by utilizing the same air which drives the gyroscope to create an erecting or damping torque on the gyroscope, and cause it to seek and hold the axis vertical.

The air flow through the instrument is obtained by means of a venturi tube mounted in the air stream. The venturi is designed to give sufficient pressure at the normal flying speed of the ship to spin the rotor at about 10,000 r.p.m.

The gyro casing consists of two chambers. The gyroscope mounted in the upper element is spun by the air from three nozzles in the casing striking the peripheral blades of the rotor tangentially. This air then passes into the lower chamber which is of a hemispherical shape. Hanging from the center point of this chamber is a small universally mounted pendulum of such length and shape that it swings freely inside the hemisphere with a small clearance between itself and the shell.

At three equally spaced points on the hemisphere there are orifices through which the air exhausts. Each orifice carries a tube with an elbow in it so that the air exhausts tangentially to that portion of the hemisphere. The exhausting air causes a reaction on the gyro casing which is at right angles to the reaction from an open port at the same location. The pendulum bob inside the shell is designed so that it covers an equal portion of each of the three ports when the gyro and its casing are in the vertical position. The air reactions, therefore, from the three ports are equal and neutralize each other. If the position of the gyro is tilted, the pendulum closes the low port and opens the high ports. The air reaction creates a torque about the gimbal axis of the gyro mounting so that the gyro precesses directly back to the vertical position. The gyro, its casing, and the pendulum are all balanced as a unit so that the unit is only slightly pendulous, and therefore, it is impossible to throw it into a periodic oscillation due to acceleration forces.

The rate of precession which is caused by the air reaction is adjusted to such an amount that it is negligible during the time that an airplane can make the maximum change of course, that is, 180 deg. Since all other maneuvers give less acceleration force than the 180 deg. turn, the artificial horizon therefore holds its position through any amount of maneuvering or tossing about of the ship.

The gyro casing is mounted in pivot bearings in the gimbal ring. The gimbal ring is pivoted to the instrument casing about

the longitudinal axis of the instrument, freedom 90 deg. bank, 80 deg. climb, and dive. The sky-earth background of the instrument face is attached to this gimbal ring and therefore tilts with the gyro and horizon. Hence, the blue sky always remains above the horizon bar no matter how much the instrument or ship is tilted. The horizon bar is carried by a link from the rear part of the gyro casing through a pivot on the gimbal ring and thence out in front of the field as a light horizontal bar. The reason for this reversing linkage is that this corrects the horizon motion to the normal apparent motion of the real horizon as observed by the pilot when he climbs or dives. It is the correction of the behavior of this bar to the normal appearance of the real horizon for any position or maneuver that makes this instrument easy to fly by a new pilot, and entirely non-fatiguing for a pilot who has to fly long periods of time blind.

Acknowledgment is made for the valuable aid in the development of the artificial horizon to the following: Curtiss Aeroplane and Motor Company, Guggenheim Fund, National Air Transport, Transcontinental Air Transport, and Eastern Air Transport.

AER-52-36

Air-Transport Management

By D. W. TOMLINSON,[1] GLENDALE, CAL.

Problems which must be solved by a manager of air-transport lines are discussed in this paper. The operating department of the T.A.T.-Maddux Air Lines is divided into two groups: (1) flight and (2) engineering. The first corresponds to the operating department of a railroad, and the second takes care of inspection, maintenance, and general overhauling of equipment. There are many essentials to a well-organized air-transport company. Weather conditions must be studied and reported, radio is a necessity, and the flying force must be chosen for its keen sense of observation as well as for skill in flying. The author describes the duties of each class into which the force is divided and tells how the work is carried on with greatest success and reliability.

THERE are three general divisions of air-transport management: executive, operations, and traffic. This paper deals exclusively with certain problems confronting a manager of air-transport operations. One of the first duties of a manager of operations of an air line is to form his organization. From the head of the operations department down to the very ship cleaners, it is of vital importance that there be a clear-cut and thoroughly understood chain of authority and succession of responsibility. The duties of every man must be clearly defined. The limits of his authority and responsibility must be established and clearly set forth in written instructions promulgated to all personnel. It is not enough that the individual himself shall know the limits of his authority and responsibility; it is also important that every man in the organization be instructed regarding the duties and authority of those with whom he works.

The choice of leading personnel is the most important problem of the manager of operations. Having planned his organization and prescribed the duties of the key men, he must fill these positions, and the selection of such men must be based on merit and proven ability alone. A mistake here can upset the entire machinery. Friendship and outside influence must be completely ignored. Too much depends upon the choice of the men in these positions, not only from a business standpoint, but from the standpoint of human lives.

To be a successful leader, a man must be of unimpeachable character himself and must possess the outstanding qualities of judgment, initiative, thoroughness, reliability, knowledge of his work in all its phases, complete loyalty to his superiors, and an ability to maintain discipline once it is established. His personality must be such that the men will labor for him out of sheer respect for the man himself, not merely because of the authority vested in him.

The operations department of an air line should be divided for administration purposes into two groups: flight and engineering. The flight superintendent is the senior of the two subdepartments, and in addition he acts as assistant to the manager of operations. The chief engineer is responsible (through the flight superintendent) to the manager of operations for the efficient maintenance of all airplanes, engines, base machinery and automotive equipment.

Under the flight superintendent are the flying superintendents of divisions and all pilots. Pilots are permanently assigned to divisions. One pilot on each division is designated as division or section superintendent. He is held responsible for the efficient performance of duty and the observance of company orders and policies by the personnel under his jurisdiction. The flight superintendent is also the chief pilot, and is in charge of all personnel. In the flight office, over which he has supervision, are kept all operating, engineering, and personnel records. To assist in handling this work he has a chief clerk, a timekeeper, and a secretary. All pilots' schedules are made up monthly by the flight superintendent in collaboration with the division and section superintendents.

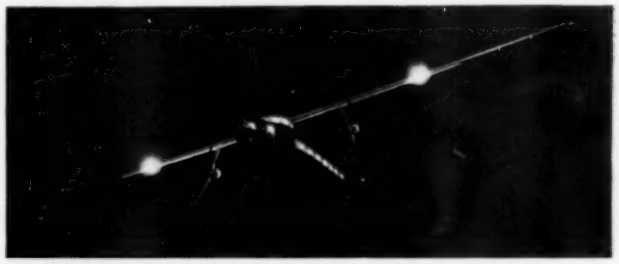


FIG. 1 FORD PLANE APPROACHING FIELD FOR NIGHT LANDING



FIG. 2 EMBARKING PASSENGERS AT GRAND CENTRAL AIR TERMINAL, GLENDALE, CALIFORNIA

As assistant to the flight superintendent, a senior pilot is designated. This pilot is picked on the basis of sound judgment and executive ability. In the event of the absence of both the manager of operations and the flight superintendent, this pilot succeeds to the full authority of the head of the operations department.

At irregular intervals the flight superintendent flies over the line in place of the senior pilot, for the purpose of keeping himself familiar with conditions on the route, and during his absence the senior pilot takes training for the position of flight superintendent, familiarizing himself with the administration work of that office.

Field managers are in charge of intermediate landing fields along the air-line route. Field crews vary, according to whether the field is merely a service stop or an overnight base. At fields where planes stop only to be serviced, the personnel consists of a weather man and a mechanic in addition to the field manager.

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[1] Vice-President in Charge of Operations, T.A.T.-Maddux Air Lines.

Presented at the Seattle Aeronautic Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Seattle, Wash., May 28 and 29, 1930.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

At overnight bases the complement is augmented by an office clerk, an additional mechanic, and a helper. The latter also acts as janitor and caretaker of the field.

A triple responsibility rests upon field managers. They are accountable to the traffic department for the handling of passengers, to the head of the operations department for general administration, and to the chief engineer for the mechanical condition of equipment under their charge.

Weather Service

The senior meteorologist and an assistant prepare the weather data at the base field. For the purpose of coordinating and standardizing the routine of weather reports, the senior meteorologist is in charge of the weather service. All weather data and meteorological reports are submitted by the weather men at intermediate fields to the senior meteorologist at the base field. While the individual weather men at intermediate fields are a part of the field complement, the weather service as a unit comes under the senior meteorologist, who in turn is responsible to the flight superintendent.

Radio

A radio engineer at the base field is in charge of all radio development work, installation of equipment in planes, and routine maintenance. Radio operators live at each radio station on the transcontinental division. The radio operators are a part of each field complement, and they are responsible to the radio engineer for the performance of their duties.

Control of Operations

An experienced pilot with a keen sense of responsibility is the only logical person to control air operations. No man without years of flying experience should be entrusted with authority over flying personnel. In addition there must be a sense of complete understanding and personal harmony between whoever is in charge and the flying crews themselves. In other words, the pilots must feel free to bring to the head of operations any problems with which they are confronted, knowing that he would not ask them to do anything he could not do himself.

At base fields there must always be one of the following on duty while planes are in the air: Manager of operations, flight superintendent, and senior pilot. Whenever the slightest doubt arises regarding any plane movement the chief despatcher is required to obtain the written approval of the head of the operations department on duty. No pilot is ever ordered to take off the ground against his personal wishes.



Fig. 3 Ford Planes Circling Aguascalientes, Mexico

After a pilot has received his instructions and has been released he alone is responsible for the safety of his plane. He may be advised regarding weather conditions, and suggestions may be made to him, but in the last analysis the pilot's judgment must be unquestioned. He alone, once he is in the air, is fully cognizant of the conditions immediately confronting him. Only under the following conditions may a pilot in the air be given direct orders:

Having landed at any field he may be ordered to remain on the ground. If in the air in sight of an established field, he may be ordered by radio to land at that field; if clear weather is known to prevail between a plane and an established field and if the plane has ample fuel, the pilot may be ordered by radio to proceed to that field.

In our own case the control of operations is delegated at distant terminals to duty pilots. For instance, the weather conditions at and adjacent to San Francisco present problems peculiar to that locality. They cannot be judged accurately from the weather reports which are received at present. Certainly, then, operations out of this area cannot be intelligently controlled by any one in Los Angeles. To overcome this, three pilots, picked because of their thorough knowledge of San Francisco weather conditions, are permanently based at Alameda. These men have proved themselves leaders and are respected as such by the other pilots. Each day one of the three is on duty at the Alameda base, under the title of "acting division superintendent." He has full control of operations out of Alameda, he makes recommendations to the head of the operations department on duty at Los Angeles, and he keeps the latter advised at all times of unfavorable or unusual flying conditions. These three pilots have the following schedule: Each of them in turn makes a round trip to Los Angeles one day, is off duty the next day, and on the third day is in charge of operations at Alameda.

The transcontinental division is another example of this system. The stretch of country between Clovis, New Mexico (the end of the run), and Los Angeles is too long for any one man to watch. Weather conditions over this route vary widely. It is therefore divided into two sections: Section A, from Los Angeles to Winslow, Arizona; section B, from Winslow to Clovis, New Mexico.

Air operations on section A are controlled by the head of the operations department on duty at Los Angeles. Operations over section B are controlled by a duty pilot at Clovis. Four pilots are assigned to this section. A pilot flies from Clovis to Winslow one day, he returns the next, is off duty the third day, and on the fourth day he is on duty in charge of operations over the section. Thus there is a pilot thoroughly acquainted with the terrain and the weather in charge of plane movements. This pilot, while on duty, is required to collaborate closely with the weather man at Clovis, and his decisions regarding plane movements over his section are final. He releases the westbound plane from Clovis. It proceeds to Albuquerque, and at that point is again released by him by wire. It proceeds to Winslow, and at this point responsibility for its movements passes from the hands of the duty pilot at Clovis and into the hands of the head of the operations department at Los Angeles. The same scheme of things prevails with the eastbound plane, which, after reaching Winslow, passes from the jurisdiction of the Los Angeles office into that of the duty pilot at Clovis, who is thereafter responsible for its movements.

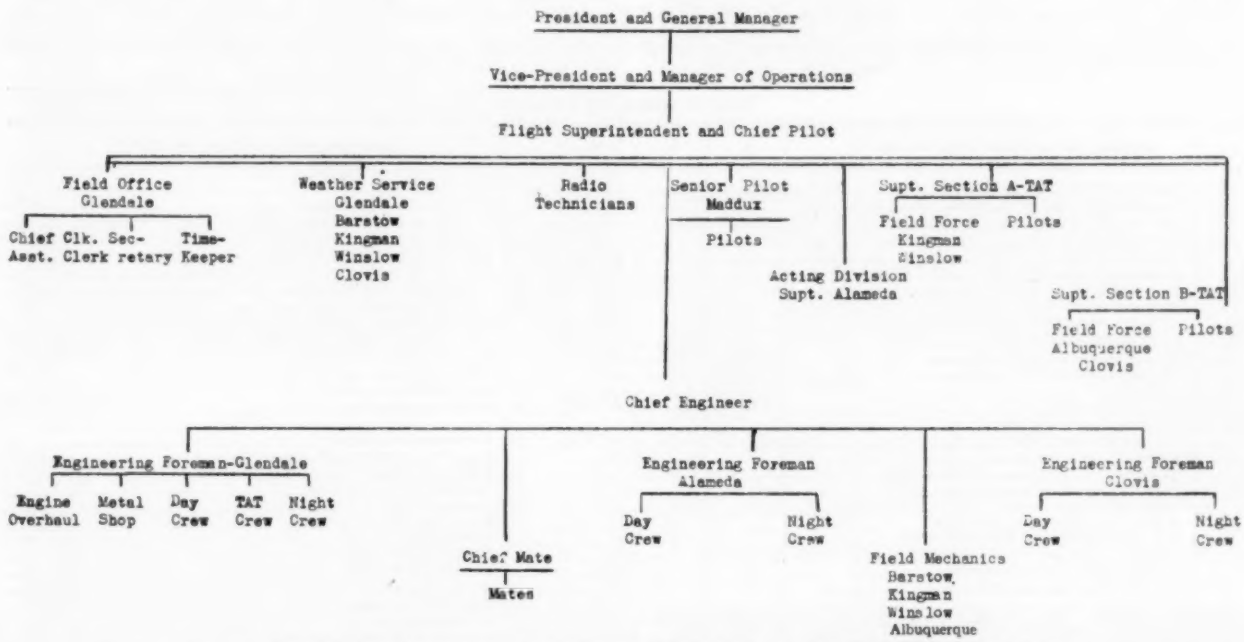


FIG. 4 ORGANIZATION CHART FOR OPERATIONS DEPARTMENT OF T.A.T.-MADDUX AIR LINES

This is based on the principle that two heads are better than one. It is believed that the pilot who is actually flying the plane has all he can do to take care of the immediate situation. He should not be asked while in flight to spend mental energy in analyzing weather conditions beyond his next stop. This job can be done by the duty pilot who, working in close cooperation with the weather service, can study carefully the reports and, together with his own knowledge of conditions gained by routine flying over the same route, arrive at a sound decision. Some may object to this arrangement on the ground that one irresponsible pilot may do untold damage. It is presumed, however, that every pilot has been selected as much for his sense of responsibility and judgment as for his flying ability.

It should be a cardinal principle in directing and managing air operations that a pilot need violate orders but once, or display poor judgment, or take unnecessary risks but once, to lose his job.

FORM F10 3M 1-30 DEP.

A.T.—MADDUX AIR LINES
WESTERN DIVISION
MAINTENANCE REPORT

PLANE NO. NC. 9641 DATE 1-31-30
FROM Glendale to A.C.M. STATION Basel
TO Glendale

				FLYING TIME	ENG IDLING TIME
Plane No. 14 Time to date since overhaul 1001-40	1st Flight	2:45	1004	hrs. 25 min.	min.
L Motor No. 1053 Time to date since overhaul 7-15	2nd Flight		10	hrs. — min.	min.
C Motor No. 1051 Time to date since overhaul 117-45	3rd Flight		120	hrs. 30 min.	min.
R Motor No. 1135 Time to date since overhaul 197-10	4th Flight		199	hrs. 55 min.	min.
L Prop. No. 5461 Time to date 425-10	5th Flight		427	hrs. 55 min.	min.
C Prop. No. 5462 Time to date /	6th Flight		/	hrs. / min.	min.
R Prop. No. 5464 Time to date /				hrs. min.	min.
Hours since oil change 29-45		29	2	hrs. 45 min.	min.
ENGINEER'S NOTES	Total	/		hrs. min.	min.

(15) → Change oil. Hook tail light up. Check L. motor, — 29
throughout, also R. Motor. Replace right elevator bottom wire — (4)
29

Signed Pastorius
Engineer

CHECK WORK TO BE DONE

	INITIAL WHEN WORK IS DONE	L	C	R		INITIAL WHEN WORK IS DONE		
	Clean motors					Inspect fittings		65
	Inspect carburetor and controls					Aileron and rudder hinges		65
	Inspect gas lines and flow					Elevator and tail skid bolts, tail post		65
	Inspect priming lines					Horizontal stabilizer adjustment		65
	Inspect Luckenheimer					Vertical stabilizer and fittings		
/	Check valve clearances	29		29		Tail skid shock		
/	Inspect valve springs				7	Landing gear and fittings	65	29
	Oil valve stems				7	Landing gear shock	5	29
	Oil valve stems					Wheel retainers and wheels		
	Pack Rocker Boxes					Aileron control wires, oil pulleys		
7	Grease rocker arms	29	29	29		Elevator wires, oil pulleys		
7	Remove push rods and grease	29		29		Control column and fittings		65
						Instruments		
7	Check magnetos	29-5		5-29	7	Switches and lights	65	5-29
	Check magneto couplings							
7	Check magneto points	29-5		5-29		Do necessary patching		
	Check magneto heads					Inspect wings ~~throughout~~ side		66
	Check wiring for broken insulation, etc.					Inspect fuselage throughout		65
7	Tighten and safety spark plugs	29		5		Test air pressure in tires		
	Inspect propeller blades				7	Wheel spokes	5	29
	Tighten all motor bolts and safety					Grease wheels		
	Tighten exhaust pipe bolts and safety					Brakes and controls		
	Tighten intake pipe bolts and safety					Clean ship thoroughly		
	Inspect battery and connections					Gas lines and valves		
	Inspect motor cowling and safety					Gas strainers		
	Check oil pump					Gas tanks and saddle		
7	Oil change	15	15	15		Air speed tubes		
	Oil magnetos					Chairs—nails—broken willows, etc.		
	Vaseline magnetos to keep rain out					Safety belts		
	Check booster magnetos					Rudder peddles		
	Tighten all motor mount bolts and safety					Battery OK		65
7	Clean oil screens	15	15	15		Cabin fire extinguishers		
	Tighten propeller hub and safety					Flares Copper Safety		65
	Tighten propeller hub bolts and safety					Clean cabin and cockpit		
	Check propeller alignment					Clean windows		
	Check spark plug gaps					Clean lavatory		
						Supply soap, toilet paper, cups		
	Installed new plugs left motor			29		Generator		
	Installed new tail light assm			29		Control Box		

Motor change—Out L C R Number _____ In L C R Number _____

Gas and Oil OK. 65

SERVICE

Station																		Totals
Gasoline—Gals.			189															
Oil—Qts.	L	C	R	L	C	R	L	C	R	L	C	R	L	C	R	L	C	R
	4 4	4 4	4 4															

I hereby certify the above ship inspected and ready for passenger transportation.

Signed L. Berkenkamp
Inspector

Accepted Tex Marley
Pilot

Accepted _____
Pilot

FIG. 5 MAINTENANCE REPORT FOR T.A.T.-MADDUX AIR LINES

The locomotive engineer who passes a stop signal is through. The pilot who in any way shows the least indication of lack of judgment, lack of respect for his company's policies, bad airmanship, or disloyalty to his superiors must be discharged immediately. Such action may occasionally entail some injustice to the individual, but the majority must be considered and also the fact that today human lives are at stake in this pilot's care, to say nothing of the thousands of dollars which are bound up in equipment under his control. The success of an aviation enterprise must not be jeopardized out of sentiment for some one individual.

Engineering Force

The mechanical force and the flying mates are under the chief engineer. At the base field and at outside fields where planes

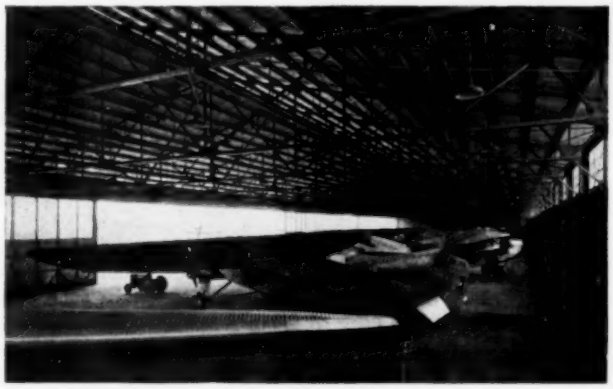


Fig. 6 Interior of Hangars at Grand Central Air Terminal, Glendale, California



Fig. 7 Despatcher's Office at Western Division Headquarters, T.A.T.-Maddux Air Lines

are regularly based overnight and maintenance work is done, there is an engineering foreman. He is responsible to the chief engineer for maintenance of equipment and for efficient workmanship by his men. In accordance with established routine, he assigns planes to routes. He is in charge of the operating line where engines are warmed up. The base-field force is further divided under the engineering foreman, as follows: (*a*) An engine-overhaul foreman is in complete charge of the engine-overhaul shops; (*b*) a metal-shop foreman is in complete charge of the metal shop and all metal work on planes or equipment; (*c*) a night foreman is in complete charge of the night crew.

The latter is a most important assignment, as the major portion of routine maintenance of planes and motors is done at night. All engines are changed at night. In order properly to fill the position of night foreman a man must be exceptionally competent. Although the chief engineer is available on call,



Fig. 8 Metal-Work Shops at Western Division Headquarters, Glendale, California



Fig. 9 Engine Overhauling Shop at Western Division Headquarters



Fig. 10 Motor Dismantling and Cleaning Department at Western Division Headquarters

the night foreman has frequently to cope with situations requiring knowledge of ship movements the following day. It is no small task to detail a night crew in such a manner as to complete the varying work necessary on as many as twelve trimotored Fords. This demands not only executive ability, but a thorough understanding of the capabilities of each man in the crew.

Inspection Forms

An inspection form is made out for each flight. This printed form shows the total plane hours, hours since overhaul of each motor, hours since etching, and check of each propeller, plane, and motor numbers, and (in column form) a list of all routine maintenance work which may be required. Before a plane leaves the operating line the chief mate inspects the plane, and if it is all right he signs the form in the proper place. The pilot then signs as accepting the plane for flight. Upon completion of the trip the mate enters the flying time for the trip. This brings the plane, engine, and propeller hours up to date.

Under the heading "Remarks" are noted any special comments regarding behavior of motors or work to be done thereon. Routine servicing required is checked in the proper place on the form. The mate then signs the form as engineer of the plane.

On arrival at the base field this form is taken up by the engineering foreman. He is responsible that the work indicated on the inspection form be performed. In event that major repairs or something unusual is required, he may consult with the chief engineer.

All foremen and mechanics are assigned numbers. Mechanics are detailed by the engineering or night foreman (as the case may be) to perform the required work. As each operation is performed and checked off, the man doing that work places his number against the check mark on the inspection form. If work is done in addition to that required by the mate of the plane, an entry is made on the last inspection form, and the number of the mechanic performing this work also appears thereon.

Thus it is possible to check on the quality of every mechanic's work. Faulty or careless workmanship can be traced directly to the man responsible. Errors by mechanics are severely dealt with.

Engine Operation

A considerable departure from the present standards of engine operations is employed. Pratt & Whitney Wasp engines are run between 400 and 500 hr. without overhaul. It is the author's belief that a smooth-running engine should not be removed from a plane and given a major overhaul merely because its log shows that it has run, say, 250 hr. An aircraft engine, such as the Wasp, will give definite indications that an overhaul is due well in advance of actual failure or even serious wear. Such engine operation presupposes painstaking and careful routine service. The periods between routine greasing, checking, and changing of oil are set very conservatively and are rigidly observed. Most satisfactory results have been obtained.

As an experiment twelve Wasp engines were allowed to go between 850 and 900 hr. before being removed from the planes for major overhaul. All these motors were top-overhauled at approximately 450 hr. At this time valves were ground and new rings installed. When torn down for major overhaul the maximum cylinder wear was 0.003 in. No cylinders were over 0.001 in. out of round. The master rod bearings showed a wear of 0.0015 in. beyond normal. A few knuckle pins were too loose and required replacing. Aside from the foregoing and new rings and exhaust valve guides, these engines were in splendid condition.

Further operating experience proved top overhauls to be uneconomical. For this reason, and because top overhauls entailed keeping a plane out of commision during a flying day, all engines are now removed for major overhaul after between 400 and 500 hr. of service. A special night crew changes all engines. Wasp engines are changed by a crew of three mechanics, this operation requiring 8 hr. to perform.

Flying Mates

The chief mate is directly responsible to the chief engineer for the assignment of mates to planes and for the manner in which they discharge their duties. The chief mate is also chief inspector of planes and is required to inspect every plane before it leaves the operating lines. The mates are flying mechanics. About half the mates are transport pilots. The other half hold limited commercial licenses. They must be expert mechanics.

A mate is assigned to each plane. He remains with this plane on all trips, except on his day off, when a relief mate is assigned. A mate is held rigidly accountable for the condition of his plane and its motors. It has been found that men of the right caliber take great personal pride in their particular planes.

With the present excess supply of pilots it is possible to find many well-trained and experienced transport pilots of first-pilot timber who are glad to avail themselves of the chance to train as mates with a view to being selected as pilots on the line as vacancies occur. A board composed of the flight superintendent, senior pilot, and division superintendents selects such pilot material from among the mates employed on the line.

As a mate is selected for the pilot availability list, he is given one hour's solo in tri-motored Fords, at the end of which time he is required to make turns and approaches to landings on one- and two-motor combinations. He is also checked for night flying and given ten solo night landings. In this manner there are always available mates who can be moved up to pilot's rating whenever they are needed.

Discipline

The day of the haphazard barnstormer or fixed-base operator is past. Discipline in the air is as essential today as in any other line of commercial endeavor. Perhaps it is more vital in aviation than in any other line, both on the ground and in the air, because of the human lives directly involved. Many fatal accidents and much avoidable inefficiency may be laid directly to lack of ordinary discipline; and hand in hand with discipline go the morale of the organization—that unspoken, intangible *esprit de corps* which exists in the rank and file—the pride of every man in his job, the knowledge that an order issued is an order to be obeyed, and the determination to work for "happy landings" which extends from the pilot in the cockpit to the humblest ship-cleaner on the field.

APM-52-1

Natural Frequency of Gears

By R. E. PETERSON,[1] EAST PITTSBURGH, PA.

The natural frequency of machine parts as related to the noise problem in machine operation is discussed in a general manner. Vibration phenomena in disks and rings are described in detail, following which the author discusses the manner in which a gear vibrates at its natural frequency. An empirical formula is given for the natural frequency of a gear in terms of its dimensions and material.

ONE of the tendencies in engineering design at the present time is to increase the speed of rotating machinery. In many cases noise and vibration become the limiting condition as the speed is increased, such being frequently the case in gear operation. This has stimulated considerable research in sound and vibration phenomena, which will eventually make it possible to deal with problems of this type in a more fundamental manner. This paper deals with a type of vibration which under certain conditions becomes important in rotating machinery.

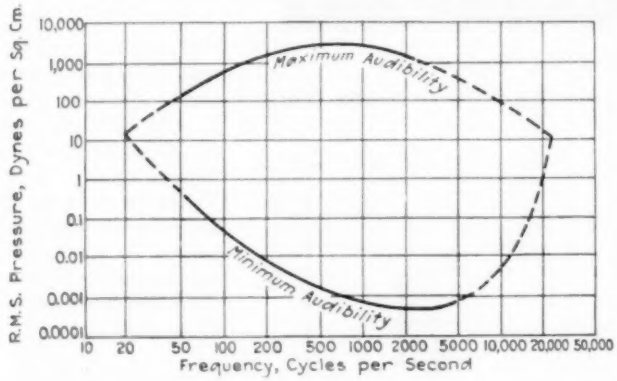


FIG. 1 RANGE OF AUDITION

If a machine part is suspended by a wire and struck with a hammer, a number of sound frequencies can be elicited by striking in various places. The lowest of these sound frequencies corresponds to the "fundamental" mode of vibration of the part, and the higher sounds are "overtones." Although all of the component sounds are "natural frequencies," it will be convenient to refer to the "fundamental" as the "natural frequency" of the machine part in question.

A study of "natural frequency" of machine parts is important for two reasons. First, if the part in question is subject to impacts it will vibrate at its natural frequency and thereby generate sound, provided the frequency is within the range of audition shown in Fig. 1.[2] Each impact will instigate a train of damped waves, as shown in Fig. 2, while the frequency is always the same, being the natural frequency of the machine part.

In gear operation it is common experience that an idling condition is apt to result in quite pronounced ringing. Particular examples of this occur in street cars during coasting and in punch presses when idling. This ringing sound has been shown experimentally to be of a frequency corresponding to the natural frequency of the drive gear. Light loads seem to be particularly conducive to the impact type of excitation referred to above. The second reason for the importance of "natural frequency" is the possibility of resonance. If the machine is operated at such a speed that a vibration occurs which is of the same frequency as the "natural frequency" of a machine part, a resonant condition occurs which may result in a tremendous increase in the amplitude of vibration. Fortunately resonance does not occur very often in gear operation according to present practice. In railway gearing, for instance, the number of tooth contacts per second is of the order of 100–300, while the natural frequency of the gear is 800–1200 cycles per second. However, with increased speeds and teeth of finer pitch, resonant conditions will occur more frequently than at present.

It will be noticed from Fig. 1 that the natural-frequency values of railway gears (800–1200 cycles per sec.) are such as to be quite close to maximum aural sensitivity. It is also apparent from Fig. 1 that to produce audibility of a sound of 1000 cycles per second requires only about one-hundredth of the pressure necessary to produce audibility at 100 cycles per second. The high sensitivity of the ear to frequencies of the order mentioned tends further to increase the importance of natural-frequency phenomena of gears.

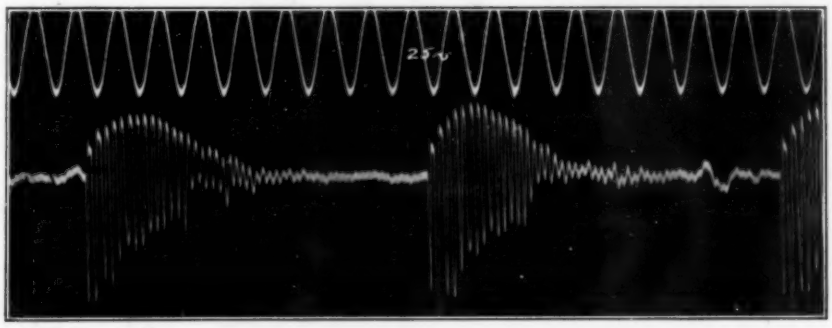


FIG. 2 TRAIN OF DAMPED WAVES DUE TO IMPACTS

Various methods of determining the frequencies of sounds are available, the following of which were tried in connection with this investigation:

1 Variable resonator
2 Microphone, amplifier, and oscillograph
3 Microphone, amplifier, and tuning-circuit analyzer.

The latter method, making use of an analyzer developed by Messrs. Spooner and Foltz,[3] proved to be the most expedient

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] H. Fletcher, "Useful Numerical Constants of Speech and Hearing." *Bell System Tech. Jl*, vol. 4, no. 3, p. 2.

[3] T. Spooner and J. P. Foltz, "Study of Noises in Electrical Apparatus." *Jl. A.I.E.E.*, Mar., 1929, p. 199.

and was used in obtaining the data given in this paper. Oscillograms taken according to the second method mentioned are shown in Figs. 2 and 3.

This investigation was made with particular reference to railway gearing of a rolled forged-steel type. The results, however, if judiciously used, may be applied to other types of

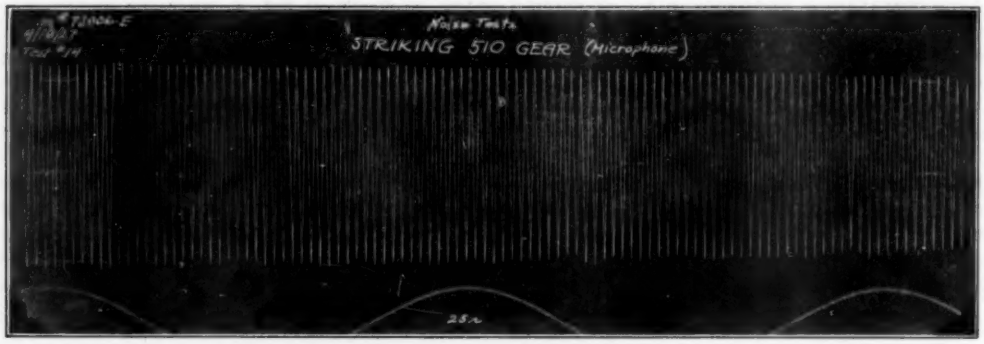


FIG. 3 OSCILLOGRAM TAKEN WITH MICROPHONE, AMPLIFIER, AND OSCILLOGRAPH

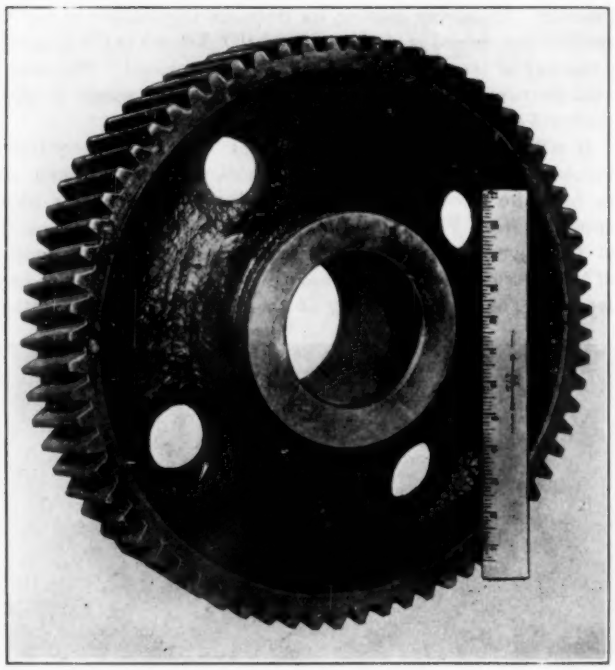


FIG. 4 TYPICAL RAILWAY GEAR

gearing, as well as to wheels, disks, rotors, and other circular machine elements. A typical railway gear is shown in Fig. 4, and a cross-section of such a gear with the notation used is shown in Fig. 5.

The first step in our analysis is to determine the upper and lower limits of frequencies which are possible in such a design, keeping the same face width and diameter. Considering first the unbored blank (before cutting teeth), when $w = a$, the result is a disk having a width a and an outside diameter d_o; and when $w = 0$, the result is a ring of width a and thickness t_b. The disk will have the highest frequency possible for a given width and diameter of gear space, while the ring will have the lowest frequency for the same space. An actual gear will have a frequency intermediate between the disk and the ring, depending on the web dimensions. It is clear, then, that a fundamental study of the vibration of disks and rings forms a logical basis for this work.

DISK VIBRATION

The natural frequency of a thin disk was solved analytically by Kirchoff[4] in 1850.

$$p^2 = \mu \frac{Eh^2}{\rho a^4} \quad \ldots [1]$$

where p = frequency $\times 2\pi$
E = Young's modulus
h = half-thickness
a = radius
μ = numerical coefficient depending on mode of vibration and on Poisson's ratio.

The mode of vibration having the lowest frequency has two nodal diameters (Fig. 6), for which $\mu = 10.61$ when a value of 0.3 is used for Poisson's ratio. Using a value of 30×10^6 lb. per sq. in. for E and a density of 490 lb. per cu. ft., the following formula will give the lowest frequency of a thin steel disk:

$$f = 208{,}400 \frac{w}{d^2} \quad \ldots [2]$$

where f = frequency of vibration in cycles per second
w = thickness of disk in inches
d = diameter of disk in inches.

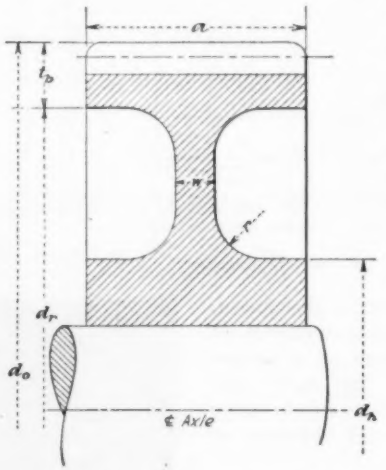


FIG. 5 CROSS-SECTION OF A GEAR, SHOWING NOTATION USED

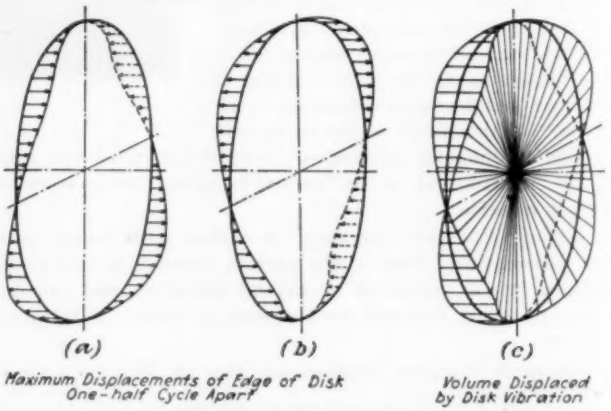


FIG. 6 VIBRATION OF A DISK (FUNDAMENTAL MODE)

[4] "Über die Schwingungen einer elastischen Scheibe," *Crelles Jl.*, vol. 40, 1850; *Pogg. Ann.*, vol. 81, 1850. See also "The Vibrations of a Spinning Disk," H. Lamb and R. V. Southwell, *Proc. Royal Soc., A.* vol. 99, p. 272.

Inasmuch as the foregoing formula applies only to thin disks it was necessary to determine frequency values experimentally for relatively thick disks. Three different diameters were used, and the results are given in Table 3 and shown graphically in Fig. 7. It will be noticed that for thin disks the agreement between theory and experiment is close, but that for thicker disks the slopes of the experimental curves decrease with increasing thickness.

Ring Vibration

The natural frequency of a ring was solved analytically by Hoppe in 1871 for radial vibrations such as are shown in Fig. 8.[5]

$$p^2 = \frac{EI}{\gamma r^4} \times \frac{s^2(s^2-1)^2}{1+s^2} \quad \ldots\ldots\ldots\ldots\ldots [3]$$

where p = frequency $\times 2\pi$
I = moment of inertia of cross-section

TABLE 1 DATA ON FREQUENCIES OF RINGS

(Thickness in radial direction = 1 1/16 in.)

Type of vibration	Mode of vibration	Outside diameter, d_o (inches)	Width in inches (in axial direction) 3 1/2	2 1/2	1 1/2	5/8
Radial	4 loops	10 3/8	1230	1230	1220	1225
		12 7/8	760	..	720	740
		18 3/4	320[a]	300[a]	310[a]	300[a]
	6 loops	10 3/8	3300	3400	3350	3350
		12 7/8	2080	..	2050	2000
		18 3/4	920	920	930	905
Axial	4 loops	10 3/8	1980	1900	1470	720
		12 7/8	1350	..	920	435
		18 3/4	640	575	420	200[a]
	6 loops	10 3/8	..	..	4050	2000
		12 7/8	..	..	2850	1250
		18 3/4	1780	1750	1230	570

[a] Approximate (tuning not sharp).

TABLE 2 RATIO OF FREQUENCY OF SIX-LOOP VIBRATION TO FOUR-LOOP VIBRATION

Type of vibration	Outside diameter, d_o (inches)	Width in inches (in axial direction) 3 1/2	2 1/2	1 1/2	5/8
Radial	10 3/8	2.68	2.77	2.75	2.74
	12 7/8	2.74	..	2.85	2.71
	18 3/4	..	..	..	..
Axial	10 3/8	..	..	2.76	2.78
	12 7/8	..	..	3.10	2.88
	18 3/4	2.78	3.05	2.93	..

Avg. ratio = 2.83.

r = mean radius of ring
s = number of complete wave lengths in circumference
E = modulus of elasticity
γ = mass per unit length of ring (on arc).

The fundamental type of vibration of a ring is similar to that of a disk (Fig. 6) in that there are two nodal diameters, mutually perpendicular. Therefore, in the Hoppe formula [3] $s = 2$ for the fundamental mode. The first "overtone" will be a type of vibration having three nodal diameters ($s = 3$) as shown in Fig. 8 (*b*).

Using values of $E = 30 \times 10^6$ lb. per sq. in., $\sigma = 0.3$, and a density of 490 lb. per cu. ft., the following formula results, giving the lowest frequency for radial vibrations of steel rings of rectangular cross-section:

$$f = 25{,}000\ \frac{t}{r^2} \quad \ldots\ldots\ldots\ldots\ldots [4]$$

where f = frequency in cycles per second
t = thickness in radial direction (inches)
r = mean radius (inches).

It will be noticed that according to formula [4] the frequency

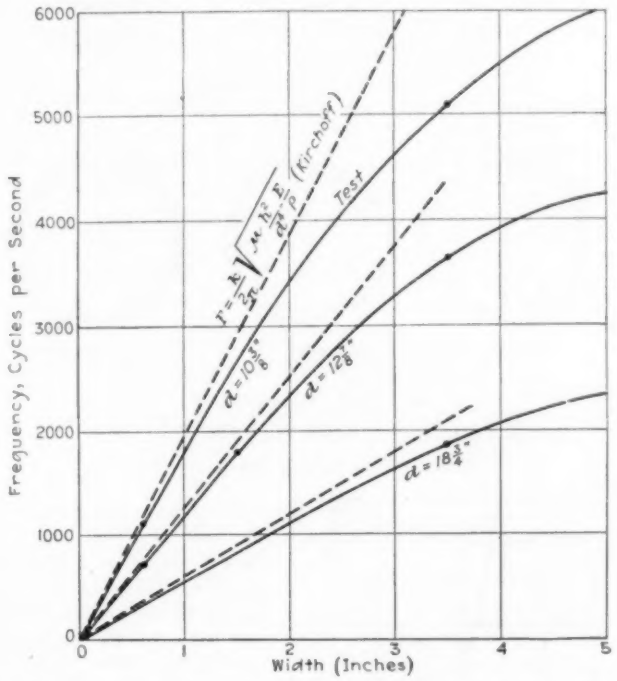


Fig. 7 Natural Frequency of Disks (Four-Loop Axial Vibrations)

TABLE 3 DATA ON FREQUENCIES OF TEST BLANKS

(Four-loop axial vibrations. All dimensions in inches)

Blank	Outside diam., d_o	Rim diam., d_r	Rim thickness, t_b	Hub diam., d_h	Face, a	Web thickness, w	Fillet radius, r	Hub length, h	Frequency, f	Notes
A-1	12 7/8	..	..	..	3 1/2	..	..	..	3450	Disk
A-2	12 7/8	8	2 7/16	6	3 1/2	5/8	1/4	3 1/2	2400	..
A-3	12 7/8	9 1/2	1 11/16	6	3 1/2	5/8	1/4	3 1/2	2000	..
A-4	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	1/4	3 1/2	1680	Fin. blank
A-5	..	..	..	..	..	..	..	..	1360	Teeth cut
A-6	..	..	..	..	..	..	..	..	1360	Hardened oil-treat
B-1	12 7/8	..	..	..	3 1/2	..	..	3 1/2	3650	Disk
B-2	12 7/8	10 3/4	1 1/16	6	3 1/2	2 1/2	1/4	3 1/2	3030	..
B-3	12 7/8	10 3/4	1 1/16	6	3 1/2	1 1/2	1/4	3 1/2	2310	..
B-4	12 7/8	10 3/4	1 1/16	6	3 1/2	3/4	1/4	3 1/2	1600	..
B-5	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	1/4	3 1/2	1530	Fin. blank
B-6	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	1/4	2 1/2	1520	..
B-7	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	1/4	1 1/2	1490	..
B-8	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	1/4	5/8	1470	..
B-9	12 7/8	10 3/4	1 1/16	6	3 1/2	0	..	..	1350	Ring
B-10	12 7/8	10 3/4	1 1/16	6	1 1/2	..	..	..	920	Ring
C-1	12 7/8	..	..	..	3 1/2	..	..	..	3650	Disk
C-2	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	13/16	3 1/2	1820	..
C-3	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	3/4	3 1/2	1630	..
C-4	12 7/8	10 3/4	1 1/16	6	3 1/2	5/8	1/4	3 1/2	1500	..
C-5	12 7/8	10 3/4	1 1/16	6	2 1/2	5/8	1/4	3 1/2	1490	..
C-6	12 7/8	10 3/4	1 1/16	6	1 1/2	5/8	1/4	3 1/2	1456	..
D-1	18 3/4	..	..	..	3 1/2	..	..	..	1880	Disk
D-2	18 3/4	9 1/2	4 5/8	6	3 1/2	5/8	1/4	3 1/2	1450	..
D-3	18 3/4	13	2 7/8	6	3 1/2	5/8	1/4	3 1/2	1150	..
D-4	18 3/4	16 5/8	1 1/16	6	3 1/2	5/8	1/4	3 1/2	700	..
D-5	18 3/4	16 5/8	1 1/16	6	3 1/2	5/8	1/4	2 1/2	700	..
D-6	18 3/4	16 5/8	1 1/16	..	3 1/2	5/8	1/4	5/8	680	..
D-7	18 3/4	16 5/8	1 1/16	..	3 1/2	0	..	..	640	Ring
D-8	18 3/4	16 5/8	1 1/16	..	2 1/2	..	..	..	575	Ring
E-1	10 3/8	..	..	..	3 1/2	..	..	..	5100	Disk
E-2	10 3/8	8 1/4	1 1/16	6	3 1/2	2 1/2	1/4	3 1/2	4700	..
E-3	10 3/8	8 1/4	1 1/16	6	3 1/2	1 1/2	1/4	3 1/2	3700	..
E-4	10 3/8	8 1/4	1 1/16	6	3 1/2	5/8	1/4	3 1/2	2400	..

[5] Rayleigh, "Theory of Sound," p. 417. Lord Rayleigh gives a solution for the radial vibrations of a long, thin tube. This differs from Equation [3] by the "plate modulus" $1/(1-\sigma^2)$, where σ = Poisson's ratio, the difference in frequencies being of the order of about 4 per cent. Equation [3] is the one which is applicable in this case, since in gears the axial dimension is less than the radius.

of radial vibrations is independent of the width (in the axial direction) of the ring. This is verified experimentally by the data of Table 1. The close agreement between theoretical and experimental values is shown by Fig. 9.

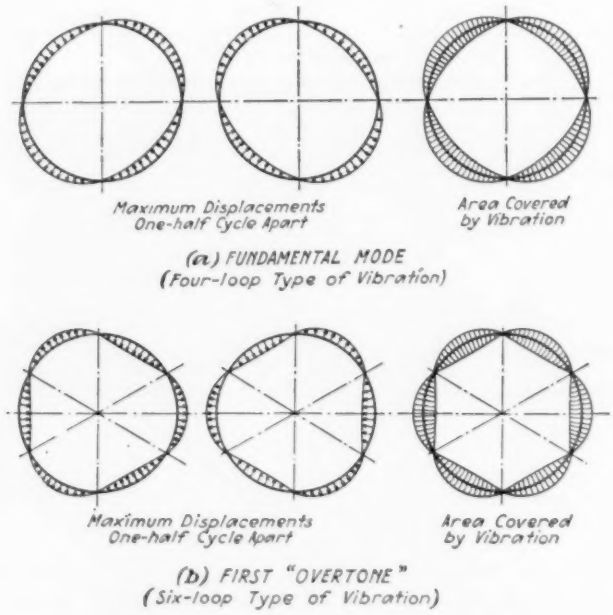


FIG. 8 RADIAL VIBRATIONS OF A RING

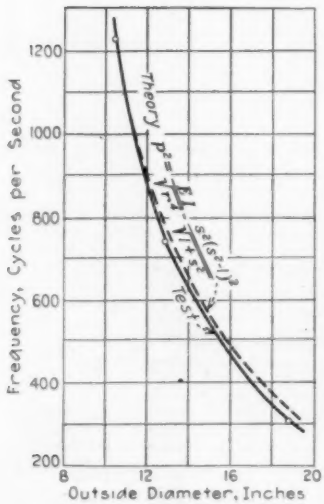


FIG. 9 NATURAL FREQUENCY OF RINGS (FOUR-LOOP RADIAL VIBRATIONS)

(Steel rings $1^1/_{16}$ in. thick; widths, $^5/_8$ in., $1^1/_2$ in., $2^1/_2$ in., and $3^1/_2$ in. See Table 1.)

Frequencies corresponding to both four-loop and six-loop types of vibration are given in Table 1. According to the Hoppe formula [3], the ratio of the frequency of a six-loop vibration to the frequency of a corresponding four-loop vibration should be 2.82. An average of the ratios of the experimental values as given in Table 2 (omitting approximate values) is 2.83. This furnishes an excellent check on the data, and also indicates a method of obtaining the value of a fundamental frequency of a ring which is too low for the apparatus (below 200–300 cycles per sec.) by obtaining the first overtone and dividing by the above ratio.

The frequencies of rings when vibrating in an axial direction (similar to Fig. 7) were also determined experimentally and are given in Table 1 and shown graphically in Fig. 10. The axial type of vibration of a ring is of particular interest because it furnishes the lower frequency limit referred to previously. This is due to the fact that the web of the gear prevents the rim from vibrating in a strictly radial manner such as in Fig. 8. In an axial direction, however, the web has much less stiffening effect, so that such vibrations are easily elicited. Therefore only the axial-frequency data are used in building up the frequency formula for gears.

VIBRATIONS OF A GEAR

Although the web of the gear prevents the rim from vibrating in a strictly radial manner, there is a radial vibration of the rim due to the axial vibration discussed above. This is clear from the cross-section (*a*) in Fig. 11, which shows that an axial displacement s is accompanied also by a rotation θ of the rim section. A similar cross-section (*c*) taken 90 deg. from (*a*) would show, at any particular instant of vibration, displacement and rotation in a direction opposite from (*a*). A nodal cross-section (*b*) would have no motion in an axial plane, remaining always as shown in (*b*). From these remarks it can be concluded that the rim of the gear vibrates in the form of a warped surface, and not as a cylindrical surface as shown in Fig. 8.

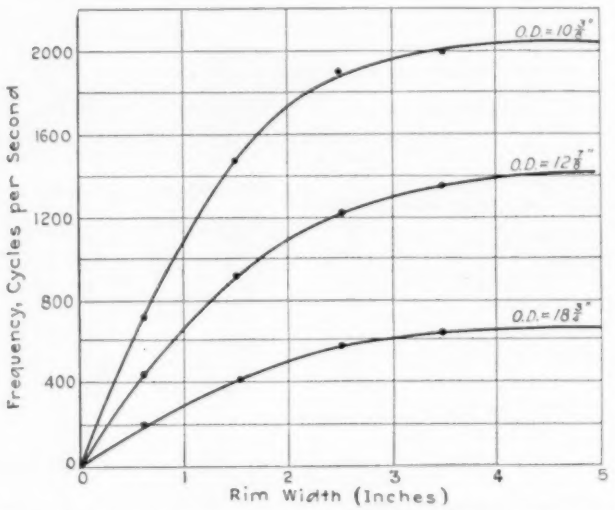


FIG. 10 NATURAL FREQUENCY OF RINGS (FOUR-LOOP AXIAL, FOR RIM THICKNESS = $1^1/_{16}$ IN.)

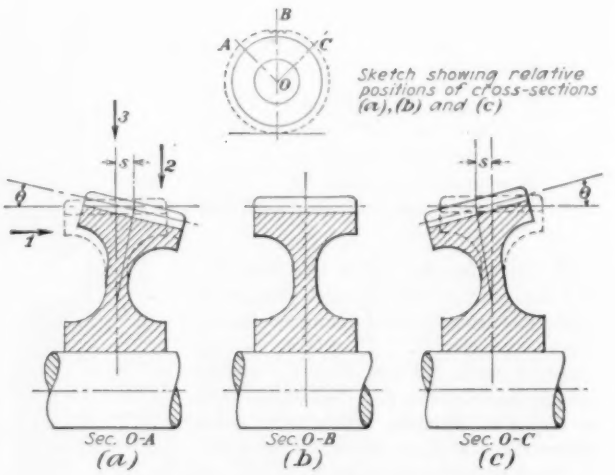


FIG. 11 VIBRATION OF A GEAR RIM

This can be verified experimentally, in that the "natural-frequency" sound can be elicited not only by striking the rim axially [direction 1 of Fig. 11 (*a*)], but also by striking an edge of the rim radially (direction 2). The "natural frequency" cannot be elicited by striking directly over the web [direction 3 as applying to the dotted outline of Fig. 11(*a*)], which should be obvious from the foregoing remarks. It follows that meshing gears in which the mating-tooth elements are not parallel ("riding on

an edge") will be more prone to ring than gearing properly aligned.

The nodal positions of a gear rim can be demonstrated by the set-up shown in Fig. 12. The gear standing on the table will have a node at *a* where the gear makes contact with the table. The remaining nodes are then at *b*, *c*, and *d*. A point *e* midway between *a* and *b* (45 deg. from the vertical diameter) will correspond to a position of maximum amplitude of a loop. In performing the demonstration, the gear is struck a series of successive blows with a babbitt hammer at *e*. The sound is received by the microphone *P* and its frequency determined by the analyzer *A*. The sensitivity of the apparatus is adjusted so that practically a full-scale reading results on the meter *M*. If the hammering is continued with the same vigor, but the point of contact is slowly shifted along the rim from *e* to *b*, the meter reading will drop correspondingly from a maximum to practically zero. Continuing the hammering from *b* to *c* along the rim will cause the meter reading to rise and fall again. Proceeding all around the rim shows practically zero readings at *a*, *b*, *c*, and *d* (nodes) and maximum readings at points midway between the nodes. Although this phenomenon is best demonstrated by means of the apparatus as described, it is distinctly noticeable by listening with the unaided ear; the effect, however, is somewhat masked due to contact noise of the hammer.



FIG. 12 SET-UP FOR DEMONSTRATING NODES

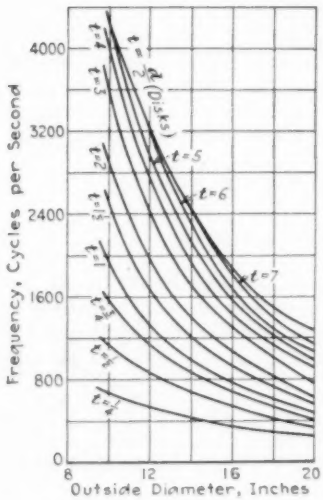


FIG. 13 NATURAL FREQUENCY OF RINGS (FOUR-LOOP AXIAL, FOR WIDTH = 2½ IN.)

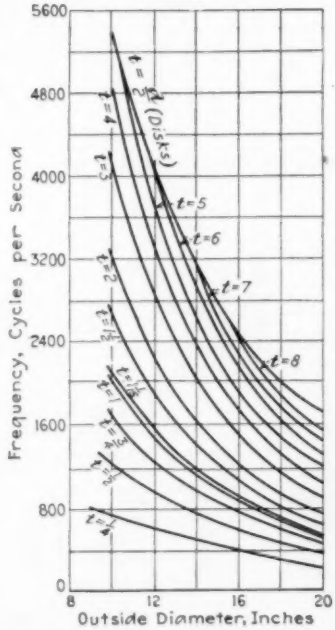


FIG. 14 NATURAL FREQUENCY OF RINGS (FOUR-LOOP AXIAL, FOR WIDTH = 3½ IN.)

NATURAL-FREQUENCY FORMULA

As previously explained, the frequency of a gear will be intermediate between that of a ring represented by the rim and a disk represented by the gear space. From Figs. 7 and 10, a series of ring-frequency charts were made for ring widths of 2½ in., 3½ in., 4 in., and 5 in. (Figs. 13, 14, 15, and 16). These widths were chosen to correspond to the face widths of railway gears. From the data given, other similar charts can readily be made for other widths. The upper curve in each chart represents disk frequencies (the ring thickness then being equal to half the outside diameter). The upper and lower frequency limits can be determined by use of one of the charts shown. Thus for a gear having an outside diameter d_o = 16 in., a face $f = 3\frac{1}{2}$ in., and a rim thickness t_b = 1 in., the frequency range becomes 860–2560 cycles per sec., corresponding to the ring and disk frequencies of Fig. 14. The actual frequency of the gear will be somewhere between the limits determined above, depending on the stiffening effect of the web.

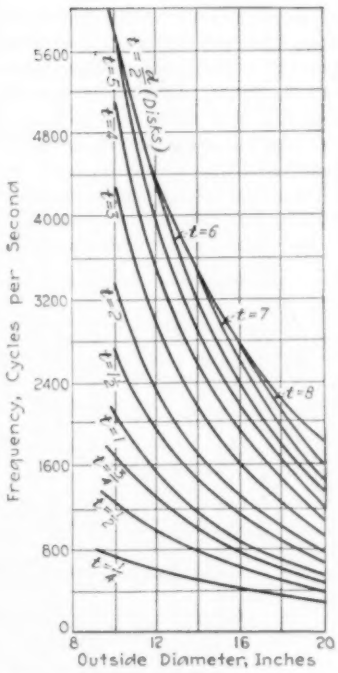


FIG. 15 NATURAL FREQUENCY OF RINGS (FOUR-LOOP AXIAL, FOR WIDTH = 4 IN.)

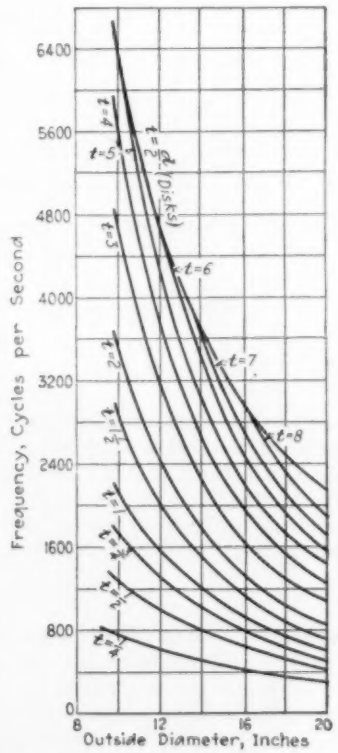


FIG. 16 NATURAL FREQUENCY OF RINGS (FOUR-LOOP AXIAL, FOR WIDTH = 5 IN.)

If the stiffening effect of the web be denoted by K_w, the

actual effect may be formulated as applying to the frequency range discussed above in the following manner: $K_w(f_d - f_r)$, where f_d = frequency of disk and f_r = frequency of ring. The web factor K_w depends on the ratio w/a, and must be such that $K_w = 0$ when $w/a = 0$, and $K_w = 1$ when $w/a = 1$. The web factor also depends on the ratio of d_h/d_r, and must be such that $K_w = 1$ when $d_h/d_r = 1$.

From the data of Table 3, a chart (Fig. 17) was made giving K_w as a function of d_h/d_r, with w/a as a parametric relation.

The data for web effect were taken on blanks having a fillet

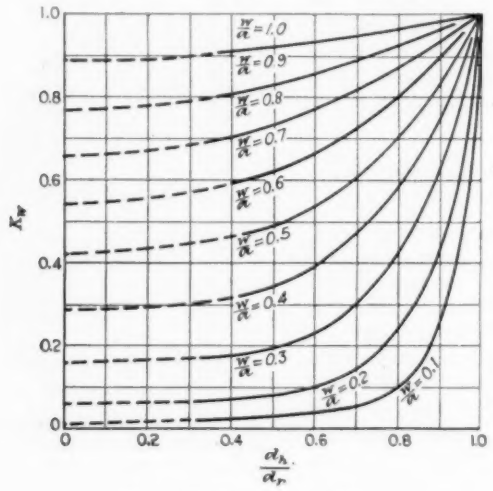


FIG. 17 WEB EFFECT K_w

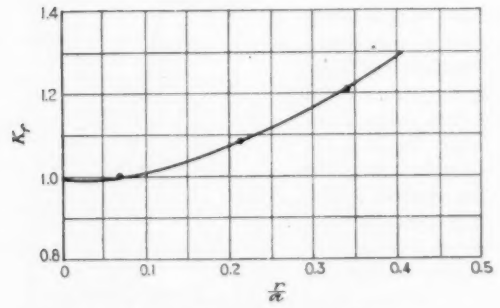


FIG. 18 FILLET EFFECT K_f

radius $r = 1/4$ in. To determine the effect of fillet radius, a series of tests were made wherein only the fillet radius was changed. The results of these tests are shown in Fig. 18, where the abscissas are given as a function of r/a, the fillet factor K_f applying to the entire blank. Thus the frequency of the steel blank (before cutting the teeth) becomes

$$f_b = K_f[f_r + K_w(f_d - f_r)] \quad [5]$$

where f_b = frequency of steel blank, cycles per sec.
K_f = fillet factor (Fig. 18)
K_w = web factor (Fig. 17)
f_r = frequency of ring (Figs. 13–16)
f_d = frequency of disk (Figs. 13–16).

The effect of cutting teeth on the frequency of the blank was determined for four gears (Table 4). On the basis of these rather incomplete data, Fig. 19 was drawn. For the types of gearing investigated, the effect of cutting teeth may be approximated by $K_t = 0.9$. Thus the formula for the complete steel gear becomes

$$f_g = 0.9\ K_f[f_r + K_w(f_d - f_r)] \quad [6]$$

where the notation is as given above for the frequency of the blank.

The effect of heat treatment on natural frequency was shown to be negligible (Table 5). The four holes usually bored through the web also have practically no effect on frequency. The same is true of bore size, the hub diameter being taken account of in the web factor K_w.

Formulas [5] and [6] are applicable only when the material is steel. A formula will now be given for use in connection with materials other than steel.

In Equation [1] the material is represented as a function of $\sqrt{E/\rho}$ which is the velocity of sound in the material.[6] This is also true in Equation [3], and is quite generally the case for solids other than disks and rings.

For steel having $E = 30 \times 10^6$ lb. per sq. in. and a density

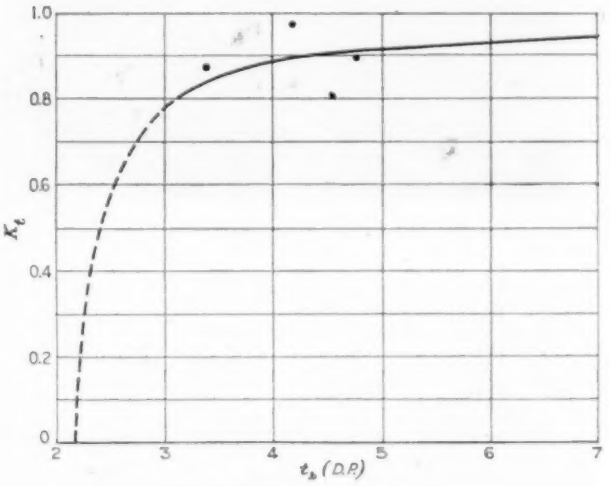


FIG. 19 EFFECT OF TEETH K_t

of 490 lb. per cu. ft., the velocity becomes 16,800 ft. per sec. The comparative effect of materials may therefore be entered as $V/16{,}800$ or $V \times 5.95 \times 10^{-5}$ where V = velocity of sound in the material in ft. per sec. (See Table 6.) Formula [6] then becomes

$$f_g = 5.35 \times 10^{-5}\ V\ K_f[f_r + K_w\ (f_d - f_r)] \quad [7]$$

TABLE 4 EFFECT OF CUTTING TEETH ON FREQUENCY

No. of teeth	D.P.	t_b	Frequency — Blank	Frequency — Gear	K_t
56	3½	1.19	1520	1480	0.97
69	4	1.19	1140	1020	0.90
69	3	1.13	730	645	0.88
53	4¼	1.06	1680	1360	0.81
					Avg. 0.89

TABLE 5 EFFECT OF HEAT TREATMENT ON FREQUENCY

No. of teeth	D.P.	Frequency — Before heat treatment	Frequency — After heat treatment	Diff.	Per cent diff.
75	3	444	455	+11	+2.5
69	4	900	895	− 5	−0.6
53	4¼	1360	1360	0	0

After completing the work just described, it was thought desirable to check the frequencies obtained by formulas [5] and [6] for a number of gears picked at random against the actual frequencies as determined by the analyzer. This was done for twelve gears, and the results are given in Table 7. The average numerical error (departure from test value) is

[6] Barton, "Text Book of Sound," vol. 1, p. 183.

5 per cent. The errors are plus and minus, so that the average algebraic error is less than 1 per cent. It may therefore be concluded that the above formulas give good results for gears having a rim thickness between 3/4 in. and 1 1/2 in. and an outside diameter between 12 in. and 36 in. It may be possible to extrapolate the data to sizes out of the range specified, but the accuracy of such a procedure has not as yet been checked.

In order to determine the natural frequency of a gear, the following measurements should be made with an ordinary scale:

(1) d_o = outside diameter
(2) d_r = rim diameter (inside)
(3) d_h = hub diameter
(4) a = face
(5) w = minimum web thickness
(6) r = fillet radius (estimate).

$$t_b = \frac{d_o - d_r}{2}$$

The frequency is then obtained by substituting in formula [6] values obtained from one of the ring charts (Figs. 13–16, depending on face width), and other values obtained from Figs. 17 and 18.

It is hoped that the data given in this paper may also prove useful in connection with applications other than gearing.

ACKNOWLEDGMENT

Acknowledgment is due Messrs. T. Spooner and J. P. Foltz for the use of the sound analyzer and for assistance in obtaining the data, to Mr. J. P. Den Hartog for suggestions in regard to presentation of the ring frequency equations, and to Messrs. C. Bethel and J. M. Lessells for instigation and encouragement of gear-noise research.

TABLE 7 NATURAL FREQUENCIES OF GEARS

(Check of computed and test frequencies. All dimensions in inches)

No. of teeth	D.P.	O.D. d_o	Face, a	Hub diam., d_h	Rim thickness, t_b	Web thickness, w	Fillet radius, r	Frequency Test	Frequency Comp.	Frequency Diff.	Per cent difference
74	4 1/4	17.66	3 1/2	6 1/2	0.96	7/8	15/16	980	981	+ 1	+ 0.1
69	4	17.59	4	7	0.86	7/8	19/16	1070	1110	+ 40	+ 3.7
58	3 1/2	16.91	4	7	1.06	1	1 1/2	1200	1170	− 30	− 2.5
54	3	18.35	5	8	1.13	1 7/8	1 15/16	1170	1050	−120	−10.2
59	2 1/2	24.08	5	10 1/2	1.36	7/8	2 1/16	665	690	+ 25	+ 3.8
Blank	...	16.44	4	7 1/2	1.19	1 3/16	1 1/2	1520	1540	+ 20	+ 1.3
56	3 1/2	16.44	4	...	1.19	1 3/16	1 1/2	1480	1390	− 90	− 4.4
Blank	...	17.69	4	7 3/4	1.19	3/4	1 1/2	1140	1090	− 50	+ 5.5
69	4	17.69	4	7 3/4	1.19	3/4	1 1/2	1020	980	− 40	− 6.1
Blank	...	23.44	5	8	1.13	1 1/8	2	730	770	− 40	− 3.9
69	3	23.44	5	8	1.13	1 1/8	2	645	690	+ 45	+ 7.0
75	3	26.50	5	7 3/4	1.25	7/8	2	444	497	+ 53	+12.0
69	4	17.63	3 1/2	7	0.94	3/4	1 1/4	900	936	+ 36	+ 4.0
							Avg. numerical difference				5.0
							Avg. algebraic difference				+ 0.8

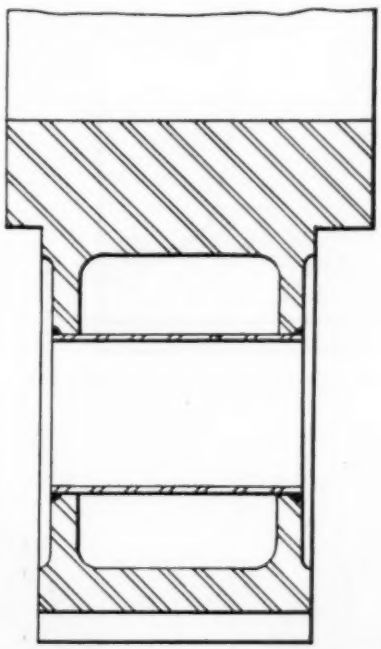


FIG. 20 CAST-STEEL NON-RESONANT CORK-FILLED GEAR

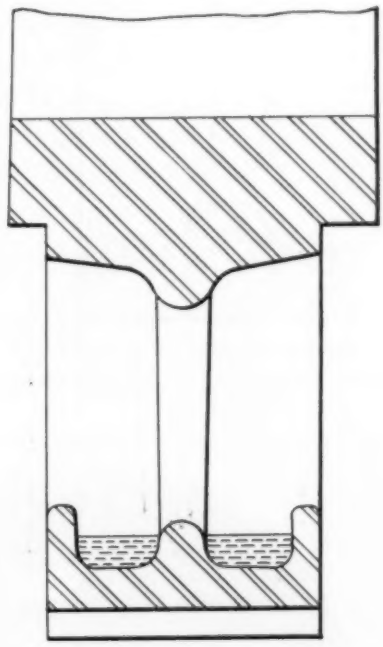


FIG. 21 CAST-STEEL NON-RESONANT GREASE-TYPE GEAR

TABLE 6 VELOCITY OF SOUND IN METALS

(From Smithsonian Tables)

Substance	Temp., deg. cent.	Velocity, ft. per sec.	Observer
Aluminum	...	16,740	Masson
Brass	...	11,480	Various
Copper	20	11,670	Wertheim
Copper	100	10,800	Wertheim
Copper	200	9,690	Wertheim
Iron and soft steel	...	16,410	Various
Iron	20	16,820	Wertheim
Iron	100	17,390	Wertheim
Iron	200	15,480	Wertheim
Cast steel	20	16,360	Wertheim
Cast steel	200	15,710	Wertheim
Lead	20	4,026	Wertheim
Nickel	...	16,320	Masson
Tin	...	8,200	Various
Zinc	...	12,140	Various

Discussion

A. A. Ross.[7] 'It may be of interest to know how we attempt to change this natural frequency in railway gears. Everybody will agree that the average street car is noisy. The gear is blamed for the larger part of the noise, although in reality it is responsible for a very small part of it.

[7] Engineer, Gear Department, General Electric Company, West Lynn, Mass.

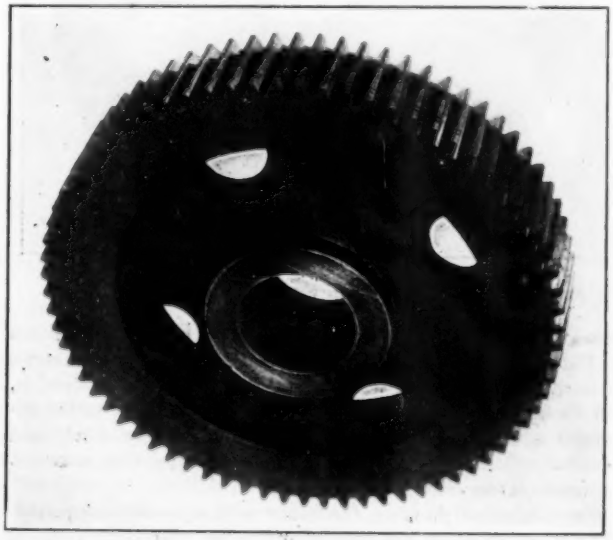


FIG. 22 CAST-STEEL GEAR SHOWN IN PLAN IN FIG. 21

In 1922 and 1923 we began a study of some commercial method of deadening railway gears or to change the natural frequency in order to overcome the very objectionable high-pitch ringing it stick. This led to the type of gear shown in Figs. 21 and 22, which are cast-steel gears with circumferential webs on each side of the rim which collected the grease in the gear pan. This acted as a satisfactory deadener, and in order to produce this same effect on the forged gear, grooves were cut on the under side of the rim and sheet-metal rings were snapped into the grooves, as shown by Figs. 23 and 24. These sheet-iron rings were snapped in place in much the same way as the retaining rings that hold an automobile tire, and after the rings were snapped in, the ends were spot-welded.

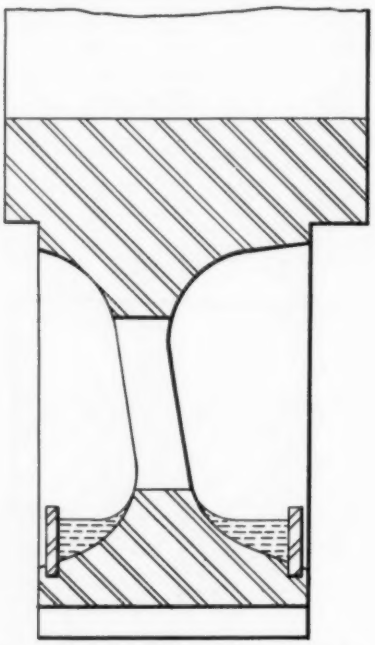


FIG. 23 FORGED OR CAST NON-RESONANT GREASE-TYPE GEAR

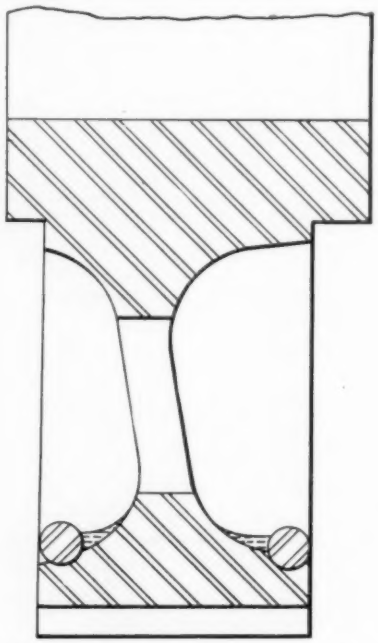


FIG. 25 FORGED OR CAST NON-RESONANT RING-TYPE GEAR

Both the cast and forged grease-type gears were effective from a deadening point of view, but the customers objected to the additional amount of grease which it was necessary to put into the pan, their objection being that much grease was thrown out on the street. Several thousands of this type of gear have been placed in service.

To overcome the objections to the added amount of grease it was discovered that sheet-steel rings, shown in the forging in Figs. 23, 24, 25, and 26, had a considerable deadening effect without the grease, and this led to what we call our standard non-resonant type of railway gears, as shown in Figs. 25 and 26. By this plan a groove was cut on the under side of the rim, both in the casting and the forging, and $^3/_{16}$-in. cold-rolled rods were wound into rings and snapped into the grooves, and the ends were spot-welded. It is surprising to notice the deadening effect between this type of gear and one without rings.

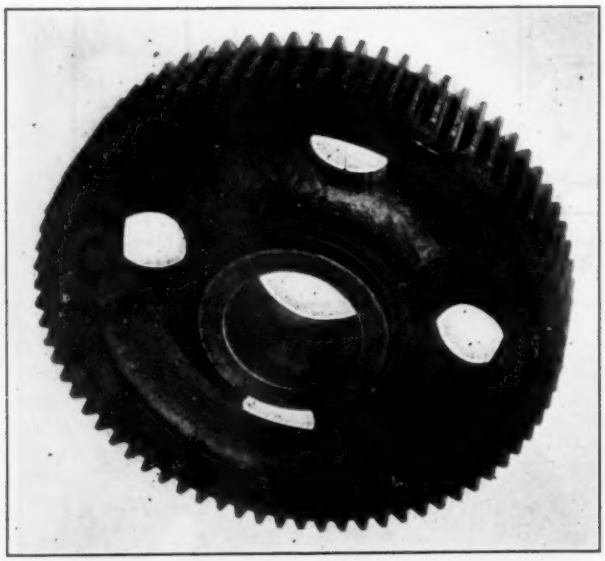


FIG. 24 CAST-STEEL GEAR SHOWN IN PLAN IN FIG. 23

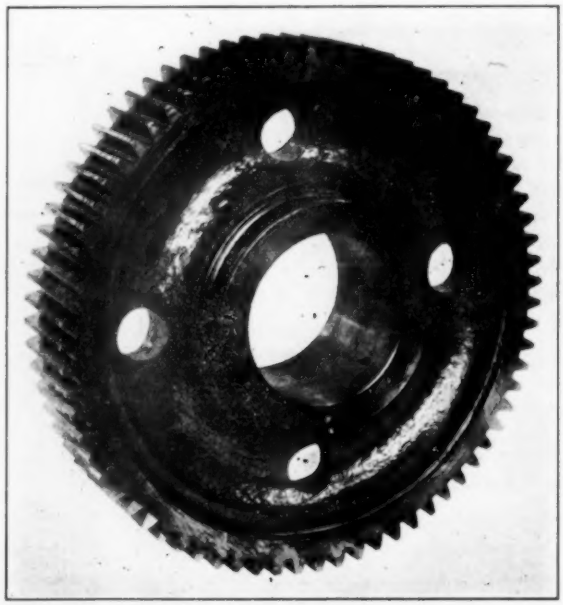


FIG. 26 FORGED OR CAST GEAR SHOWN IN PLAN IN FIG. 25

sound when the car is coasting. This brought out a gear as shown by Fig. 20. This is a double-web, cast-steel gear with a space between the webs filled with ground cork. This was effective, and about 500 were placed in service, but they were rather expensive to make and could be furnished only in cast steel; and since the rolled forged gear was the popular type, they were not a commercial success.

We then tried painting the gears with a heavy compound, which when dried on was very effective, but vibrations loosened the compound, and we could find no satisfactory way of making

GEORGE B. PEGRAM.[3] It is stated in the paper that the number of tooth contacts per second in railway gearing is 100 to 300,

[3] Dean, School of Mines, Engineering, and Chemistry, Columbia University, New York, N. Y. Mem. A.S.M.E.

while the natural frequency of the gear is 800 to 1200 cycles per second. Likening the system to a pendulum which receives timed impulses at definite intervals of a few swings, it might be expected that with the natural frequency of the gear 1200 cycles per second, for example, distinct resonance of this type might appear when the number of tooth contacts is, say, 300 per second, giving an impulse every fourth vibration. Does this actually occur?

The curves in Fig. 7, representing a comparison of frequencies of circular plates as obtained empirically with those obtained from Kirchoff's formula, are very interesting. They show that the theoretical formula is very satisfactory if the thickness is small in comparison with the radius of the plate. For thicker plate, the discrepancy obtained can be explained by taking into account the effect of shearing forces. Due to these forces, the flexibility of the plate increases, hence the lowering of the frequency with increase in the thickness of the plate.

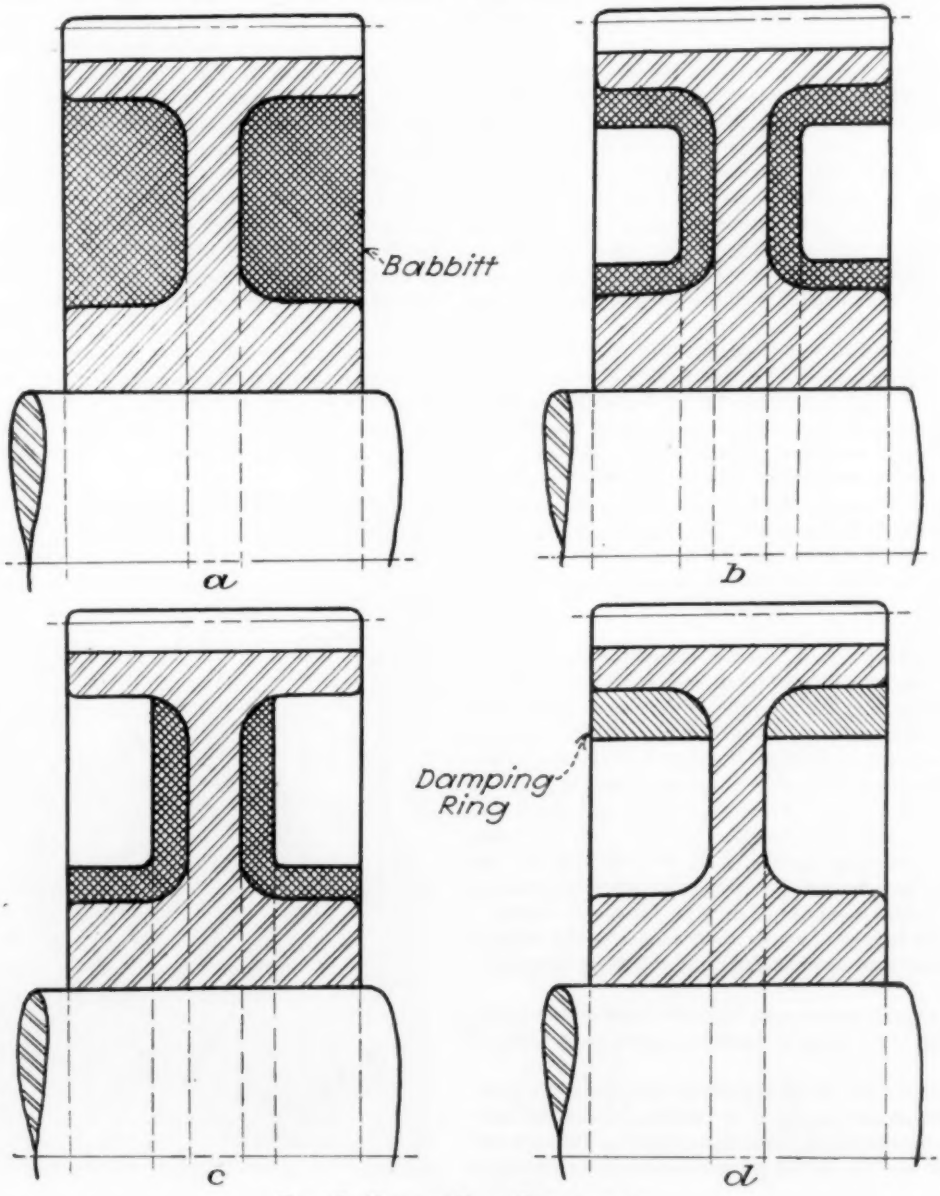


FIG. 27 DAMPING-RING DEVELOPMENT

S. TIMOSHENKO.[9] The author must be congratulated on his very interesting paper. By using empirical method, a table of frequencies of gears of various proportion was established which enables a designer to predict with sufficient accuracy the frequency of vibration of a gear, knowing the dimensions and physical properties of the material.

SANFORD A. MOSS.[10] The author has given a remarkable treatment of a subject which, so far as the writer has been able to find out, has never before been discussed. The method of vibration of a thin disk has long been known. Spreading sand on a horizontal disk, which is then set into vibration, results in the collection of

[9] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[10] Engineer, Thomson Research Laboratory, General Electric Company, Lynn, Mass. Mem. A.S.M.E.

the sand along two diameters, which are thereby shown to be at rest, with spaces of maximum oscillation between.

As the writer understands it, the author shows that the well-known ringing sound of a noisy gear is exactly the same sort of thing. His demonstration of this and his method of computing the period are novel and remarkable. Of course, it is well understood that poorly cut teeth and other items contribute to gear noise. Nevertheless, the author's treatment of actual frequency of the gears is a fundamental item.

However, the mere scientific analysis of the fundamental principles of noisy gears leaves the subject very much in the condition of the successful operation where the patient died. Not only must we have the scientific analysis which the paper presents but also means of eliminating the noise. There are a number of methods of doing this which wholly depend upon the stopping of the natural vibration which the author discusses, and some mention of these should be included in the paper. All of these methods accomplish the same thing that is accomplished by the soft pedal of a piano when it touches a vibrating string and so serves to damp the vibration.

A. A. Ross, of the Metal Gear Department of the General Electric Company, has accomplished much in this direction. The device which he now has in commercial use, consisting of metal rings fitting in grooves at each edge of the gear rim, makes a remarkable difference in the noise. A gear without the ring, when tapped with a hammer, makes the ringing sound which the author so ably treats. If, then, Mr. Ross's rings be snapped in place, the ringing sound entirely disappears, leaving only a low thud. The exact reason for all of this is completely given by the author. The writer is informed that the noise of a gear in service is similarly affected.

J. P. Den Hartog.[11] A. A. Ross mentions the fact that the ringing of a gear can be prevented by slipping in two rings, having initially a larger diameter than the slot in the gear in which they fit. The action of these rings, in the writer's opinion, is due to the small relative motions between the rings and the gear while vibrating. This causes dissipation of energy, which is in a way proportional to the product of the relative displacements and the friction. It is clear that when the pressure between the ring and the gear is very small or zero, there is no friction and the scheme does not operate. On the other hand, when the pressure between ring and gear is very great or when the ring is welded to the gear in a large number of points, there cannot be any relative motion, consequently no energy dissipation, and again the scheme will not work.

Between these two extreme cases, however, there must be some pressure at which the ringing is deadened in the most effective manner.

I would like to ask Mr. Ross whether he ever has noticed any differences in the deadening action of various rings in the same gear, and if so, whether he has found any correlation between this deadening action and the initial pressure between the rings and the gear.

Author's Closure

The discussion has been unusually profitable, and the author wishes to express his appreciation to all who have participated. A particularly interesting feature was the discussion of the methods of damping natural frequency ringing of gears. The manuscript for this paper was prepared over two years ago, and although damping rings were in use at that time, not enough was known concerning their behavior to enable the author to include an accurate scientific treatment of the damping phase of the subject. Our initial experience with damping means was rather interesting. We realized from the manner in which a gear vibrates (Fig. 11) that practically all of the vibration was in the rim. To check this an experiment was made, using babbitt as a damping medium. A gear was filled with babbitt as shown in Fig. 27*a* and was very dead when struck with a hammer. Then the babbitt was cut out until a $^3/_8$-in. shell remained, as shown in Fig. 27*b*, which arrangement was also dead. The next step was to cut the babbitt away from the rim portion as shown in Fig. 27*c*, resulting in a gear which had a lively ring when struck with a hammer. This confirmed the theory as to the source of vibration and made it clear that the place to apply damping means was to the rim. A large number of damping rings have been applied since that time, but it is only recently that the fundamental action involved has been realized. The damping takes place in the manner described by Dr. Den Hartog in his discussion—i.e., a sliding friction due to relative motion between the ring and the rim. That a relative motion will ensue is clear from a consideration of Fig. 11. Dr. Den Hartog points out that at very loose or very tight pressures the ring should be ineffective. This has been known for some time to be the case, but only recently has the author been able to obtain experimental verification, making use of the sound analyzer and a source of constant energy excitation in the form of the hammer-blow apparatus shown in Fig. 28.

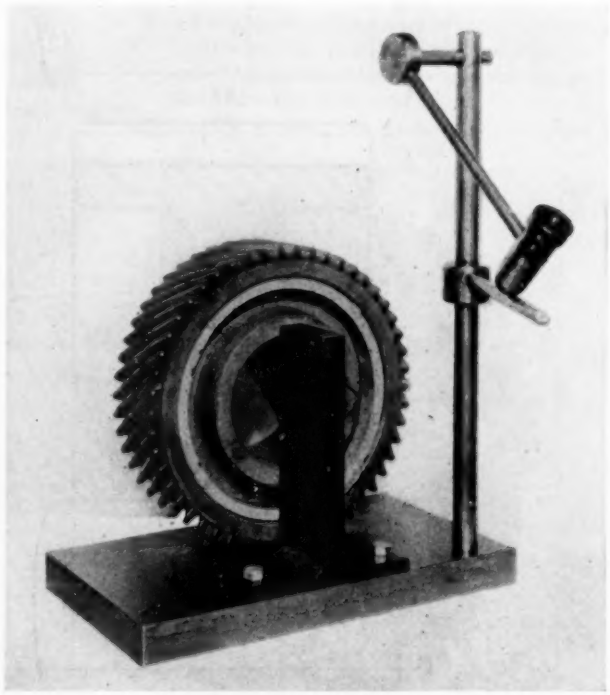


Fig. 28 Hammer-Blow Apparatus

The author feels that Mr. Ross is to be congratulated on producing an inexpensive damping device. The fundamental action in all devices of this kind seems to be the same, and it is interesting to note that results of quite similar nature were evolved from developments which proceeded independently along radically different lines.

Dean Pegram raised the question of resonance due to excitation which is a sub-multiple of the natural frequency of the gear. Such resonance does not occur,[12] as can be shown from

[11] Engineer, Research Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

[12] Frith and Buckingham, "Vibration in Engineering," p. 36.

general vibration properties.[13] The ratio of periods mentioned by Dean Pegram is 0.25 and by reference to Fig. 255 of "Applied Elasticity,"[13] it is seen that the resonant effect is about 1.05. This means an increase in amplitude of about 5 per cent, which is small compared to resonant effects of several hundred per cent.

[13] Timoshenko and Lessells, "Applied Elasticity," p. 331.

The remarks of Dr. Timoshenko concerning plate theory are interesting. The effect of the shearing forces is capable of analytical solution, and it may prove interesting for some one to work this out and compare the results with the actual frequency measurements on thick plates as given in this paper.

In conclusion, the author wishes particularly to express his gratitude for the discussion by Dr. Moss and for his comments before and after the presentation of the paper.

APM-52-2

Factor of Safety and Working Stress[1]

By C. RICHARD SODERBERG,[2] VESTERAS, SWEDEN

This paper contains a general discussion of the fundamentals on which working stresses in machine parts should be based. The terms "failure" and "factor of safety" are also discussed. Rules for the determination of working stress as established by the East Pittsburgh Works of the Westinghouse Co. are given. It is stated as the author's opinion that the engineering profession is in need of a general code on the subject of working stress. The beginning of such a code has been made in the "Code for Design of Transmission Shafting" approved by the American Standards Association in 1927. In an appendix, this code is also discussed. It is hoped that this paper may lead to some action on the part of the A.S.M.E. toward the establishment of a general standard or code on the subject.

Introduction

CALCULATION of the strength of a mechanical structure is properly subdivided into two problems:

(1) Determination of the stresses actually existing in the various parts of the structure.

(2) Determination of the danger of failure of the various parts of the structure under these stresses, or, as it is usually expressed, determination of the working stresses corresponding to a certain factor of safety.

Both are problems in the theory of elasticity, and both have been the subject of much theoretical and experimental study. In practical engineering work, however, the former problem has been given far more attention than the latter.

One of the reasons for this is the difference in the nature of the two problems. The results in the former appear as solutions of certain mathematical problems, and have a satisfactory degree of preciseness and accuracy. The final conclusions in the second problem, on the other hand, are based upon experiments both difficult in execution and interpretation and extending into several branches of engineering science. In addition, our knowledge of the mechanism of strength and failure of engineering materials is discouragingly incomplete, and it is only in recent years that definite and reliable principles have started to emerge out of the controversy between different theories.[3]

It seems safe to state, at the present time, that there are very few engineering problems where it is not possible by theoretical reasoning or experiments to arrive at a practical solution of the existing stresses, provided that the nature of the problem warrants the expenditure of the necessary effort. When the consequences of these stresses are to be predicted, however, the solution, if at all possible, is very uncertain, and it is only in certain simple problems that the results of independent engineers will agree. This condition will remain even if we eliminate the element of variability in our materials; it exists partly because of the lack of knowledge on the subject and partly because of the lack of clear definitions.

From a practical point of view it is obvious that a stress calculation, however complete and correct, is utterly useless unless it is associated with an evaluation of the stresses in terms of failure.

The "technical worth" of engineering materials was discussed by A. B. Kinzel[4] in an interesting paper before the A.S.M.E. in 1928, and the subsequent discussion indicated that this general subject is one of great interest to mechanical engineers. It appears desirable to the author to give this subject further attention, and the present paper has been written in the hope that it might lead to some action, on the part of the A.S.M.E., toward the establishment of a certain standard or code on the subject.

The beginning of such a code has already been made in the "Code for Design of Transmission Shafting" approved by the American Standards Association in 1927. This represents a nucleus of valuable material which deserves to be enlarged.

It is the opinion of the author that the engineering profession is in need of a general code on the subject. It is freely admitted that our knowledge on the subject of failure of materials is insufficient for a definite standard in the ordinary sense of the word, but as long as it is sufficient to design machinery successfully it deserves to be classified in a system where the terms have a significance as definite as that of any other engineering science. This is, unfortunately, not the case at the present time.

It is not the object to give definite recommendations for the subject matter of this code. The paper contains a general discussion of the terms "failure" and "factor of safety." Appendix No. 1 gives an example of rules on working stresses established by the East Pittsburgh Works of the Westinghouse Electric & Mfg. Co., and Appendix No. 2 gives a discussion of the above-mentioned "Code on Design of Transmission Shafting." The latter gives an example of deductive reasoning which shows that in many instances the knowledge available at the present time may be systematized into rules no more complicated than purely empirical short cuts.

Failure

Popularly, the word "failure," when applied to the qualities of strength in an engineering structure, signifies a disastrous event: a boiler exploding or a bridge falling down. Technically the word "failure" signifies a cessation of the proper functioning of the structure; whether the consequences are disastrous or not depends on the manner in which the potential energy stored in the structure is dissipated.

This cessation of the proper functioning of the structure may correspond to stresses near the ultimate strength, the yield point, or the elastic limit, or to stresses far below any of these values. In brittle materials it may be accompanied by an actual crack in the structure, in ductile materials by a certain yielding. The term "failure," therefore, involves the nature of operating conditions, the nature of the stress conditions, and the behavior of the materials under these conditions.

A very important part of the work toward a code on working

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[1] The outline of this paper was planned in connection with the activities in a committee on "Mechanical Calculations" at the Westinghouse Electric & Mfg. Co. Acknowledgment is hereby extended to Dr. S. Timoshenko, Dr. A. Nadai, Messrs. J. M. Lessells, M. Stone, W. J. Merten, and others for a number of the ideas expressed in the paper.

[2] Allmänna Svenska Elektriska Aktiebolaget. Assoc-Mem. A.S.M.E.

[3] Dr. A. Nadai, "Der bildsame Zustand der Werkstoffe," Julius Springer, 1927.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., Dec. 2 to 6, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] A. B. Kinzel, "Evolution of the Technical Worth of a Steel From Physical Test Data," Trans. A.S.M.E., vol. 49–50 (1927–1928), paper no. APM-50-12.

stresses must therefore be the classification of operating conditions, stress conditions, and materials. This is probably one of the most difficult points, and one that is most likely to create controversy. It is obvious, however, that this classification must conform to the coordinated knowledge of the behavior of the materials. For example, the common method of classifying stresses into static stresses and dynamic stresses is dangerously vague. The dynamic stresses in a rotating hoop and the static stresses in a cylindrical vessel under pressure obviously belong to the same group of steady stresses. It is entirely possible that the designer may find it prudent to employ different working stresses for the two applications, but there is no reason why the working stress—or the factor of safety—should not be defined in the same manner in the two cases. The same state of stress may also require different treatment for two materials. For example, the case of combined torsion and bending in a rod requires one treatment for mild steel and an entirely different treatment for cast iron.

Fig. 1 gives an example of a scheme of classification, which is given here merely as a possible plan.

It goes without saying that the object of a classification such as the one given in Fig. 1 is to systematize the available knowledge of the subject. Several points and combinations, such as the general subject of impact stresses, or the behavior of materials under elevated temperatures and variable stresses, are practically unknown at the present time, because the need for the solution of these problems has not become sufficiently acute. These must be left open until the incentive for the knowledge appears, when a more or less complete solution will also be forthcoming. Many combinations may be justifiable as approximations only. For example, the distinction between brittle and ductile materials is one which is associated with the state of stress as well as with the material itself. Under certain conditions of stress even very brittle materials, such as marble and cast iron under compression, behave as ductile materials. Whether or not this would invalidate a classification of engineering materials into ductile and brittle materials is a point which must be settled by considering the importance of exceptional cases of this kind. It does not appear to the author that this would be the case.

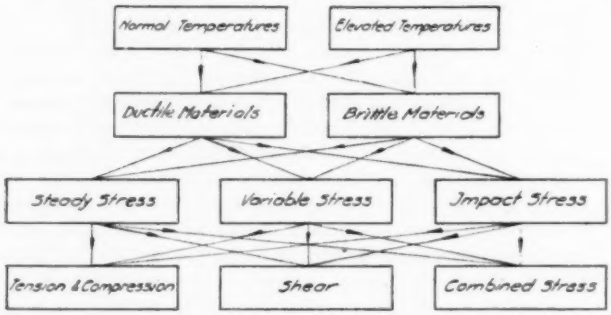


FIG. 1 CLASSIFICATION OF CONDITIONS OF FAILURE

FACTOR OF SAFETY

Having classified the conditions contributing to failure, we arrive at the central part of the problem, the specification of working stresses. Here we may also expect considerable controversy. A considerable number of engineers would probably insist upon having the stresses specified in pounds per square inch for the various subtitles of the classification and for different standard materials. While a certain amount of conventional information of this kind may be of value, it is feared that such a code would miss the point altogether. There is only one agency that can specify working stresses with finality, and that is the engineer, or the firm, responsible for the operation of the device. This is the only agency which has access to the experience from previous undertakings of the same kind, and apart from the question of public safety, it is the agency which carries the chief consequences of failures.

Past experience may be interpreted more or less intelligently, however, and it is here that a code, representing a synopsis of the knowledge on the subject, will be of value. The chief function of the code, therefore, should be to aid the designer in interpreting his experience. We may express this by stating that the code should *define* the factor of safety and the designer should *decide* on its actual value.

Unfortunately, the term "factor of safety" is one of the most abused terms in our engineering vocabulary. Fundamentally, it should represent the margin of the possibilities of the material, which is not taken into account in the functioning of the structure in question. Actually, it is used as a symbol for the designer's relative ignorance of the factors contributing to failure. This ignorance is not always unavoidable, and uncertainty of the existing stresses or conditions of operation must necessarily influence the degree to which the possibilities of the materials are brought into use, but it is not necessary that it should influence the definition of the factor of safety.

As usually expressed, the factor of safety is the ratio of the stress which will cause failure to the actual stress. If we denote the former by S_i and the latter by S, the factor of safety,[5] n, is

$$n = S_i/S \quad \ldots\ldots [1]$$

According to this definition a factor of safety of unity corresponds to failure. It is obvious that this definition is somewhat unsatisfactory in that the factor of safety itself does not express directly the fundamental idea of the margin of safety. This objection could be removed by expressing the factor of safety as the ratio of the difference between the limiting stress and the actual stress, but in practice this method appears unnecessarily complicated.

Another and a far more attractive possibility is to drop the expression factor of safety altogether, and introduce in its place its inverted value. This might be named the *factor of utilization* or the *stress factor*. Denoting it by u, we shall have

$$u = S/S_i \quad \ldots\ldots [2]$$

It could be expressed as a fraction, or as a percentage. It expresses something tangible: *the degree to which the possibilities of the material have been brought into use.*

Naturally, no fundamental advantage other than clarity of expression can be obtained by the introduction of a new term of this kind. This appears of sufficient importance to warrant its introduction, however.

The selection of constants of this nature should, of course, be made with a view toward simplicity of application. In electrical engineering one uses impedance or admittance according to whether one or the other gives simpler expressions, the simplicity usually being synonymous with the possibility of combining elements by direct addition. In the present problem this argument is in favor of the factor of utilization, as shown by the following example.

Consider a mechanical structure subjected to a system of generalized forces P_1, P_2, etc. The stress in a certain member

[5] It should be noted that the factor of safety defined in this manner has a definite meaning only when there is a single stress which contributes to failure. The meaning is still definite for combinations of tension and shear, when a combined stress can be defined. For combinations of constant and variable stresses further qualifications are necessary.

determines the safety of the structure. If all forces produce stresses of the same kind, and if there is proportionality between loads and stresses, we see directly that the factor of utilization of the material in the member in question is

$$u = u_1 + u_2 + \ldots \quad [3]$$

where u_1, u_2, etc. are the factors of utilization corresponding to the forces P_1, P_2, etc. Obviously, the factor of safety can also be evaluated from the expression

$$1/n = 1/n_1 + 1/n_2 + \ldots \quad [4]$$

but it is certainly more convenient to use the former expression.

In many cases the law of superposition given by Equation [3] or [4] will hold even if the stresses produced by the generalized forces are not all of the same kind. Such a case is given in Appendix No. 2, where it is shown that, with a certain modification of the experimental results, the effects of variable and constant stresses may be superimposed.

The principal object of a code on working stresses would thus be a definition of the factor of utilization (or the factor of safety) for each combination of practical interest in the causes of failure. It is a matter of coordinating the available knowledge in the theory of elasticity, strength of materials, and metallurgy in such a form that the factor of utilization is a definite quantity for each case. It cannot be achieved without considerable labor, and many points must be established empirically until more knowledge is available, but it is believed that the results would be of great value to industry.

Appendix No. 1

THE FOLLOWING rules for determining working stresses were proposed at the East Pittsburgh Works of the Westinghouse Electric & Mfg. Co. in 1928. They are given here to show an example of the type of rules which the author has in mind, but should not necessarily be regarded as a definite recommendation for general application.

The term "factor of safety" has been retained in these rules because it was considered unwise to establish a new term for practical purposes until it had received the official sanction of the profession. It is obvious, however, that the general form of the rules need not be changed if the factor of safety were replaced by the factor of utilization.

RULES FOR WORKING STRESSES AT NORMAL TEMPERATURES

I—Introduction

The following general rules establish the fundamental principles upon which the determination of all working stresses is to be based.

All materials are divided into two groups: ductile materials, having an elongation of more than 5 per cent, and brittle materials, having an elongation of less than 5 per cent. The principles for determination of working stresses are fundamentally different for these groups.[6]

[6] Under certain conditions all materials can be deformed permanently to a large amount without visible signs of failure. This is true even of our most brittle materials such as cast iron or marble, when they are subjected to sufficiently large *pressures* and when at least one of the principal stresses differs from the others. Ductility and brittleness, therefore, are not strictly properties of the materials, but depend on the type of stress application as well.

It has been found, however, that a classification of ductile and brittle materials, based on their properties in straight tension, is satisfactory from a practical point of view. The limit of 5 per cent permanent elongation is arbitrary, but has been found to give a safe boundary between the two groups.

All conditions of stress application are divided into two groups: steady stresses and variable stresses. The principles for determining the working stresses are different for these two types of stress application.

Only three of the material constants usually available are used for the quantitative determination of the working stress. These are yield point and endurance limit for ductile materials, and ultimate strength for brittle materials. The remaining constants are often of importance in selecting materials for specific applications, but these are taken into account in a qualitative manner only.

One of the most important objects of these rules is to give the term factor of safety a uniform significance, so that the same factor of safety in all cases designates a certain definite ratio of the stress which will cause failure to the actual stress.

While the following rules define the working stresses from the point of view of failure, there are numerous other factors involved in a design which determine material dimensions, such as rigidity, corrosion, wear, the use of standard parts, etc. For these reasons it is impossible and impractical to specify inflexible rules for the determination of the factor of safety. Furthermore, this element always involves experience from past practice, commercial considerations, and the individual problems of each type of apparatus. In cases of steady stresses in ductile materials, the factor of safety may be as low as 1.25 when the stresses and the properties of the materials are fully known. Cases involving uncertainty of any one of these elements demand higher factors of safety, but very few applications of ductile materials under static stress should require higher factors of safety than 2.0. Ductile materials in applications of variable stress and all applications of brittle materials require factors of safety above 2.0 because of the uncertain element of stress concentration.

II—Nomenclature

Items	Symbols	Units
General symbol for stress	S	lb. per sq. in.
Tensile or bending stress	S_t	lb. per sq. in.
Shearing stress	S_s	lb. per sq. in.
Principal stresses, $S_1 > S_2 > S_3$	S_1, S_2, S_3	lb. per sq. in.
General symbol for constant component of stress in cases of variable stresses	S_c	lb. per sq. in.
General symbol for variable component of stress in cases of variable stresses	S_v	lb. per sq. in.
Ultimate strength—general symbol	S_u	lb. per sq. in.
Yield point in tension or compression	S_y	lb. per sq. in.
Endurance limit for complete reversal	S_e	lb. per. sq in.
Torsion moment	M_T	inch-pounds
Bending moment	M_B	inch-pounds
Factor of safety—general symbol	n	
Factor of stress concentration—general symbol	k	
Diameter of round section	d	inches
Ratio of variable component of stress to constant component—general symbol	α	
Constants for combining variable stresses	K_1, K_2	

III—Ductile Materials
(Elongation more than 5 per cent)

1 Static Stresses. **The yield point in tension or compression**

is to be taken as the basis for the determination of the working stress.

For materials which do not have a well-defined yield point, the stress at which the permanent deformation is 0.2 per cent is to be taken as the defined yield point.[7]

(*a*) *Simple Tension or Compression.* The working stress is determined by

$$S_t = S_y/n \quad \ldots\ldots [5]$$

(*b*) *Pure Shear.* The yield point in shear for ductile materials is one-half the yield point in tension. This has been verified by experiments and furnishes proof of the validity of the maximum-shear theory. The working stress in shear is determined by

$$S_s = S_y/2n \quad \ldots\ldots [6]$$

Torsion of a bar with round section is the most common example of pure shear. The working stress is determined by

$$S_s = \frac{\frac{M_T}{\pi d^3}}{16} = \frac{S_y}{2n} \quad \ldots\ldots [7]$$

[7] The stress-strain curves, which are obtained from annealed ductile materials, can be classified into three groups.

(*a*) *The first iron type* (Fig. 2a). The deformation is elastic up to a certain point, continuing into a peak, after which it drops to the yield point S_y.

(*b*) *The second iron type* (Fig. 2b). The transition between the elastic and the plastic state is rounded, but the yield point is well defined.

(*c*) *The copper type* (Fig. 2c). The permanent part of the deformation increases gradually with increasing stress. A definite and sudden yielding cannot be observed.

This type of material is particularly susceptible to cold working (permanent plastic deformation at normal temperature), and the yield point is raised by this process (Fig. 2d). When the stress is reversed, however, after the cold working, the yield point for the reversed condition of stress is not raised materially.

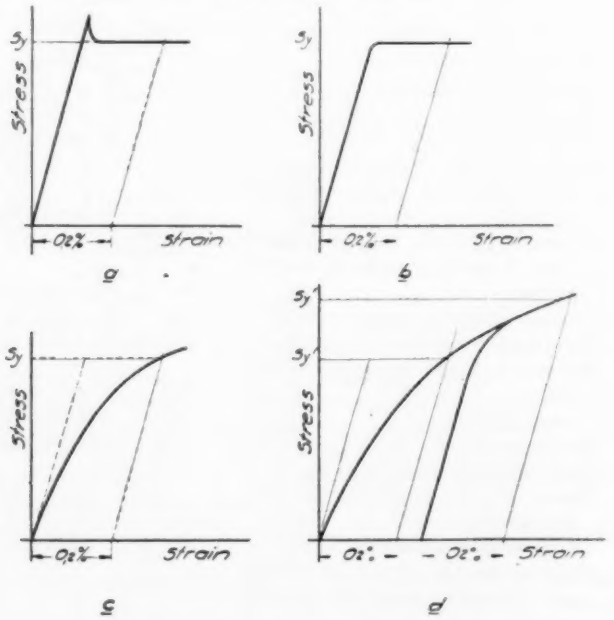


FIG. 2 TYPES OF STRESS-STRAIN CURVES

The selection of 0.2 per cent permanent deformation as the basis for a definite yield point in materials of this type is arbitrary, but in general it will give a stress at which the deformation begins to be objectionable.

See A. Nadai, "Der bildsame Zustand der Werkstoffe," Julius Springer, 1927. J. M. Lessells, "Concerning the Yield Point in Tension," Am. Soc. for Test. Matls, vol. 28, part 2, p. 387.

(*c*) *Combined Stresses.* The determination of the stress which causes failure is based upon the maximum-shear theory, which has been verified by experiments on ductile materials.

In the general case of three principal stresses (Fig. 3), failure occurs when the maximum difference between two of the principal stresses equals the yield point. If $S_1 > S_2 > S_3$ the working stresses are determined by

$$S_1 - S_3 = S_y/n \quad \ldots\ldots [8]$$

In this formula, the stresses must be taken with their proper signs (+ for tension, — for compression).

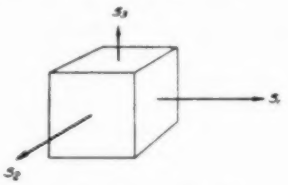


FIG. 3 THREE PRINCIPAL STRESSES

In the case of tension and shear (Fig. 4), the principal stresses are, in the order of magnitude,

$$\left.\begin{aligned} S_1 &= \frac{1}{2}\left[S_t + \sqrt{S_t^2 + 4S_s^2}\right] \\ S_2 &= 0 \\ S_3 &= \frac{1}{2}\left[S_t - \sqrt{S_t^2 + 4S_s^2}\right] \end{aligned}\right\} \quad \ldots\ldots [9]$$

The direction of S_1 is given by the angle θ, determined by

$$\tan 2\theta = \frac{2S_s}{S_t} \quad \ldots\ldots [10]$$

The working stresses are determined by the equation

$$S_1 - S_3 = \sqrt{S_t^2 + 4S_s^2} = S_y/n \quad \ldots\ldots [11]$$

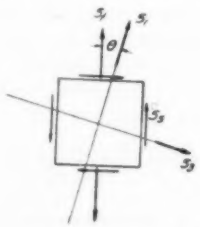


FIG. 4 TENSION AND SHEAR

Note that the combined stress $\sqrt{S_t^2 + 4S_s^2}$ is twice the maximum shearing stress, occurring in a plane at 45 deg. to the direction of S_1. This combined stress produces failure in the same manner as if it acted in pure tension or compression.

Torsion and bending of a bar with circular section is the most common example of combined tension and shear. The maximum tension and shear stresses are

$$S_t = \frac{\frac{M_B}{\pi d^3}}{32} \quad \text{and} \quad S_s = \frac{\frac{M_T}{\pi d^3}}{16} \quad \ldots\ldots [12]$$

The working stresses are governed by the relation

$$\sqrt{S_t^2 + 4S_s^2} = \frac{\sqrt{M_B^2 + M_T^2}}{\frac{\pi d^3}{32}} = \frac{S_y}{n} \quad \ldots\ldots [13]$$

2 Variable Stresses.[8] The general case of variable stress involves a cycle of stress which consists of a static component S_0, and a variable component S_v. (See Fig. 5.) The frequency of variation does not have a perceptible influence upon the behavior of the material, and need not be taken into account in the determination of the working stress.

The working stresses for ductile materials under variable stress are based upon yield point in tension or compression, and the endurance limit for complete reversals of stress. With a few exceptions for non-ferrous metals, the endurance limit is attained experimentally for 10 to 20 million reversals of stress, and may be considered to hold for an infinite number of reversals.

The conditions of stress concentration must be taken into account for all cases of variable stresses. For ductile materials

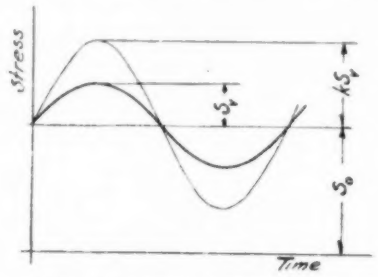


FIG. 5 GENERAL STRESS CYCLE

this applies to the variable component of stress only. Thus, if the stress S_v is obtained by ordinary elementary formulas, the actual variable stress is kS_v, where k is the factor of stress concentration, representing the ratio of maximum variable stress to average variable stress. (See Fig. 5.)

(*a*) *Simple Tension or Compression.* The actual factor of safety of a material under the combined influence of a static stress S_0 and a variable stress S_v (Fig. 5), both expressed as tensions or compressions, is defined in the following manner.

The factor of safety under the static stress S_0 when acting alone is

$$n_0 = S_y/S_0 \quad [14]$$

The factor of safety under the variable stress S_v, when it is acting alone is

$$n_0 = S_e/kS_v \quad [15]$$

where S_y and S_e are the yield point and the endurance limit, respectively. The actual factor of safety n is defined by the relation

$$\frac{1}{n} = \frac{1}{n_0} + \frac{1}{n_y} = \frac{S_0}{S_y} + \frac{kS_v}{S_e} \quad [16]$$

For the derivation of this formula see Appendix No. 2.

(*b*) *Pure Shear.* Let the static and variable components of the shearing stress be S_0 and kS_{sv}. The factor of safety is determined by:

$$\frac{1}{n} = \frac{1}{n_0} + \frac{1}{n_v} = \frac{S_0}{2S_y} + \frac{kS_{sv}}{2S_e} \quad [17]$$

(*c*) *Combined Stresses.* The stress which produces failure is defined by the maximum-shear theory.

For any specific application, the constant and the variable components of the shearing stresses are determined for an arbitrary plane. The factor of safety for this plane is then evaluated by Equation [16] or [17] as a function of the position of the plane. The particular plane which gives a minimum factor of safety is considered as the dangerous plane, and the working stresses are determined from the condition of failure in this plane.

General Case of Variable Principal Stresses (Fig. 6). The actual conditions of each case determine whether the dangerous plane is parallel to S_1, S_2, or S_3. In either case it is inclined 45 deg. to two of the stresses. Assuming it to be the plane indicated on Fig. 6, the factor of safety is determined by

$$\frac{1}{n} = \frac{S_{01} - S_{02}}{S_y} + \frac{k_1 S_{v1} + k_2 S_{v2}}{S_e} \quad [18]$$

from which the working stresses may be determined. This expression may also be written

$$\left(S_{01} + k_1 \frac{S_y}{S_e} S_{v1}\right) - \left(S_{02} - k_2 \frac{S_y}{S_e} S_{v2}\right) = \frac{S_y}{n} \quad [19]$$

In this form the equation compares with Equation [8].

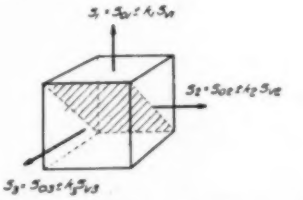


FIG. 6 THREE VARIABLE PRINCIPAL STRESSES

General Case of Variable Tension and Shear (Fig. 7). Using notations suggested in Fig. 7, the tension stress has a constant component S_1, and a variable component $\alpha_1 S_1$, magnified by a factor of stress concentration k_1. The shearing stress has a constant component S_s and a variable component $\alpha_s S_s$, magnified by the factor of stress concentration k_s.

The dangerous plane d–d is found to be determined by

$$\tan 2\theta = \frac{K_1}{K_2} \frac{S_1}{2S_s} \quad [20]$$

where the constants K_1 and K_2 are determined by

$$K_1 = 1 + \alpha_1 k_1 \frac{S_y}{S_e} \qquad K_2 = 1 + \alpha_s k_s \frac{S_y}{S_e} \quad [21]$$

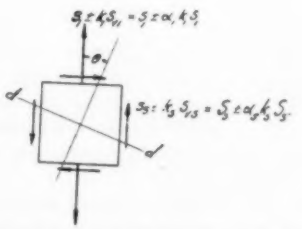


FIG. 7 VARIABLE TENSION AND SHEAR

The working stresses are determined by

$$\sqrt{(K_1 S_1)^2 + (2K_2 S_s)^2} = \frac{S_y}{n} \quad [22]$$

Note that Equation [22] is very similar to the one given for steady stress, Equation [11]. The existence of a variable component of stress merely gives rise to multiplication factors for the stresses, determined from [21].

In ordinary shaft applications the bending stress is completely

[8] For deductions see Appendix No. 2.

alternating with no constant component, and the working stresses are determined by [22], where

$$S_1 = \frac{M_B}{\frac{\pi d^3}{32}}; \quad S_s = \frac{M_T}{\frac{\pi d^3}{16}}; \quad K_1 = k_1 \frac{S_y}{S_e}; \quad K_2 = 1 \ldots [23]$$

that is

$$\sqrt{\left(k_1 \frac{S_y}{S_e} S_1\right)^2 + 4S_s^2} = \frac{S_y}{n} \ldots\ldots [24]$$

In Diesel and single-phase application a certain fraction (α_s) of the torque moment is also variable. In that case

$$K_1 = k_1 \frac{S_y}{S_e} \quad K_2 = 1 + \alpha_s k_s \frac{S_y}{S_e} \ldots\ldots [25]$$

that is, the working stresses are determined by

$$\sqrt{\left(k_1 \frac{S_y}{S_e} S_1\right)^2 + \left[2\left(1 + \alpha_s k_s \frac{S_y}{S_e}\right) S_s\right]^2} = \frac{S_y}{n} \ldots [26]$$

IV—Brittle Materials
(Elongation less than 5 per cent)

1 *Static Stresses.* For all brittle materials the ultimate strength is taken as a basis for determination of the working stress. Most brittle materials behave differently for tension and compression so that it is necessary to specify separately the ultimate strength in tension S_{ut} and the ultimate strength in compression S_{uc}.

The conditions of stress concentration must be taken into account for brittle materials by introducing a factor of stress concentration k, even for static stresses.

(*a*) *Simple Tension or Compression.* The working stress for tension or compression is expressed, respectively, by the formulas

$$S_t = \frac{S_{ut}}{kn} \quad \text{and} \quad S_c = \frac{S_{uc}}{kn} \ldots\ldots [27]$$

(*b*) *Pure Shear.* The ultimate strength in shear for brittle materials has been found by experiments to agree with the ultimate strength in tension S_{ut}. The working stress is thus determined by

$$S_s = \frac{S_{ut}}{kn} \ldots\ldots [28]$$

(*c*) *Combined Stresses.*[9] The failure of most brittle materials under combined stresses depends on more complicated relations between the principal stresses than for the ductile materials. In general, the condition of failure can be expressed as a function of the sum and the difference between the greatest and smallest of the principal stresses.

In the general case of three principal stresses (Fig. 3), it is necessary to consider separately the following cases.

When at least one of the principal stresses is a tension stress, and the maximum compression stress (if there is one) does not exceed the ultimate strength in tension S_{ut}, only the largest tension stress contributes to failure, and the material may be said to fail according to the maximum-strength theory. If the greatest tension stress is S_1, it is limited by

$$S_1 = \frac{S_{ut}}{k_1 n} \ldots\ldots [29]$$

When at least one of the principal stresses is a tension stress, and the maximum compression stress exceeds the ultimate strength in tension S_{ut}, both the tension stress and the largest compression stress contribute to failure, and the material may be said to fail according to a modified form of the maximum-shear theory. If the largest tension stress is $k_1 S_1$ and the largest compression stress is $k_3 S_3 > S_{uc}$, the working stresses are determined by

$$\left(2 - \frac{S_{ut}}{S_{uc}}\right) k_1 S_1 - \frac{S_{ut}}{S_{uc}} k_3 S_3 = \frac{S_{ut}}{n} \ldots\ldots [30]$$

Note that in this equation the stress S_3 is to be taken with a negative sign.

In cases of combination of tensile and shearing stresses (Fig. 4), the principal stresses are first calculated by aid of Equation [9] and then the working stresses are determined from [29] or [30].

2 *Variable Stresses.* In the case of alternating or pulsating stresses in brittle materials, the endurance limit for tension or compression is taken as a basis for the working stress.

Brittle materials have very unfavorable properties under variable stress, however, and their behavior under variable stress cannot be predicted with any degree of certainty. For this reason, brittle materials should be avoided where variable stress is involved. In those exceptional cases where this is not possible, the working stresses should be determined on the basis of the maximum-strength theory, and very large factors of safety must be used.

Appendix No. 2

VARIABLE STRESSES

Introduction

IN THE "Code for Design of Transmission Shafting," approved by the American Standards Association in 1927, certain general formulas and rules are presented, through which the more recent conceptions of the mechanism of failure of ductile materials are incorporated in shafting design.

This pamphlet describes in a clear and concise manner the

[9] The rules for determining working stresses in brittle materials have been based on Mohr's theory, somewhat modified.

In the ordinary Mohr diagram, Fig. 8, draw a circle corresponding to a tension stress S_{ut} and another corresponding to a compression stress S_{uc}. Draw a third circle with radius S_{ut} through the center. Draw a tangent L to this circle and the compression circle. Any stress combination which gives a largest Mohr circle that falls within the line L and the circle S_{ut} will not produce failure. The rules given for brittle materials have been based on this definition.

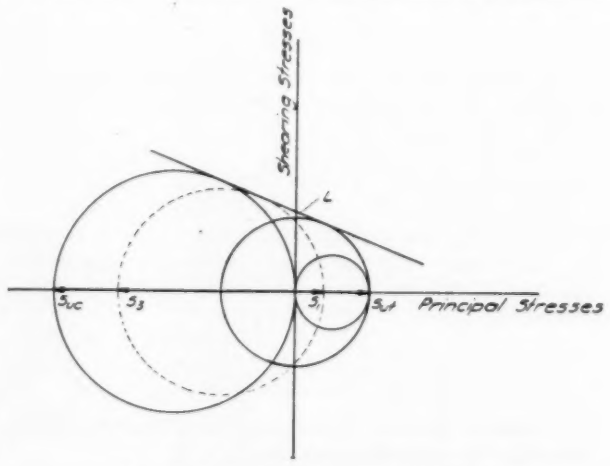


Fig. 8 Failure of Brittle Materials

more important of the existing strength theories. It seems to the author, however, that the subject of variable stresses might be advantageously treated somewhat more fully, particularly since there are many groups of shaft constructions, notably shafts for rotating electrical machines, where the alternating stresses very often are the major stresses. In particular, it seems desirable to establish a more definite distinction between shock and fatigue stresses.

If the word "shock" is taken in its ordinary meaning, a stress applied by shock would grow to its full value in a very minute element of time. For this type of stress application it is questionable whether the ordinary treatment based on the maximum-shear theory holds at all, or whether the qualities expressed by yield point, ultimate strength, endurance limit, and ductility have their ordinary significance.[10] This particular problem, the impact strength of shafts, might form the subject of a separate extension of the Code.

If the word "shock" denotes a suddenly applied load, where the time element is large in comparison with the duration of an impact, as is usually the case in most shafts, there are two possibilities. When the number of shocks during the lifetime of a shaft is small, the application may be considered one of steady stress. When the number of shocks is great, the application must be considered a fatigue problem. In a formal rule, the actual boundary must be established; this might be put at about 10^4 cycles. At any rate, we are forced to coordinate the problem under one or the other of steady or variable stress, unless we are willing to accept the complication of an endurance limit which is a function of the number of cycles to which the shaft will be subjected. This is, indeed, the only logical solution in important cases of mass production of shafts, but for ordinary cases it appears somewhat too complicated.

If we exclude the case of actual impact, therefore, the distinction between shock and fatigue reduces itself to a difference in the actual maximum value of the stresses. The case of shock is characterized by particular uncertainty regarding the maximum stress. This is of practical importance and must be taken care of, but it does not appear logical to modify the working stress in order to take care of this uncertainty; it should be taken care of in the computation of the stress.

It seems logical, therefore, to recognize the following types of stress application: steady stress, impact stress, and variable or fatigue stress.

The first appears to be adequately covered by the Code, as far as shafts are concerned. The second represents a type of stress application which is very imperfectly known and which probably must be covered by empirical short cuts until more knowledge is available. The third type, the fatigue problem proper, which is the subject of the present discussion, might be covered by less empirical rules than those given in the Code.

The method recommended by the Code for evaluating the action of shock and fatigue stresses consists in "weighting" the stress by a factor $K > 1$, which is given different values for different conditions of stress. Thus, if the shaft is subjected to a variable bending stress S_b and a variable torsion stress S_s, computed by ordinary formulas, the working stress is expressed as

$$\sqrt{(K_1 S_b)^2 + (2K_2 S_s)^2}$$

instead of the usual expression derived from the maximum-shear theory

$$\sqrt{S_b^2 + 4S_s^2}$$

It is a very interesting fact that the results obtained by analytical reasoning are exactly the same, and definite formulas are obtained for K_1 and K_2. It is not impossible that the method had its origin in a reasoning of the same kind as the one presented below; if not, it is another example of common sense anticipating the results of mathematical deduction.

Fundamental Laws of Failure

The object of the following discussion is to deduct the consequences of those principles which may be considered established for the failure of ductile materials. Briefly, they can be enumerated as follows.

1 *Steady Stresses.*

(*a*) Failure is predicted by a material constant, the yield point, which for ordinary steels, at least, can be said to be fully defined.

(*b*) Failure is due to the maximum shear stress and consists, in the initial stage, of permanent shearing deformations in the planes of maximum shear.

(*c*) Local concentrations of stress do not contribute appreciably to failure.

2 *Alternating Stress.*

(*d*) When the stress is alternating, the failure is predicted by a material constant, the endurance limit, which can be considered fully defined and measurable, for practical purposes.

(*e*) Failure is due to the maximum alternating shear stress, and consists in ruptures in the plane of maximum shear.

(*f*) Local concentrations of stress contribute to the failure, so that a stress larger than the average stress, but smaller than the maximum stress, is determining the failure.

To the above can be added the results of somewhat less comprehensive experiments for cases of variable stress, that is, when the stress consists of a constant component and a variable component. Here the results are less definite but the tendency is obvious.[11]

Referring to Fig. 5, we assume a test piece in uniform tension subjected to a stress cycle, consisting of the constant component S_0 and the variable component S_v. If S_0 and S_v are varied, and the failure for each combination is observed, it has been found that the stresses contributing to failure are related in the manner shown in Fig. 9. Thus for each value of the constant component S_0, there is a certain value of the variable component S_v which produces failure. The line of failure L varies for different materials, but in general it seems to go through the points corresponding to S_e, the endurance limit, and S_u, the ultimate strength. It appears, then, that most steels have a certain ability to stand alternating stresses even at the yield point. This fact is connected with the difference in the mechanism of failure.

As far as tests on ductile materials have been made, it seems that the above facts hold for compression as well as tension.

The Fundamental Case of Stress Cycle

Assuming the above principles to be established, we next proceed to interpret them for practical application.

If we consider first the case of only one principal stress, such as a test piece in tension and compression, we modify the stress

[10] Attention is here called to the fact that in the Applied Mechanics Division of the A.S.M.E. this subject has been proposed as the principal subject for one or several meetings in 1931.

See also W. Kuntze, "Kerbzähigkeit und statische Kennziffern," *Stahl und Eisen*, March 21, 1929, p. 394. This paper is an abstract of a more complete discussion in *Arch. Eisenhüttenwesen*, 2 (1928–1929), pp. 583–593 (Group E, No. 51).

[11] S. Timoshenko and J. M. Lessells, "Applied Elasticity," p. 477.

H. F. Moore, "An Investigation of the Fatigue of Metals," University of Illinois Eng. Sta. Bulletin No. 136.

B. P. Haigh, "Alternating Stress Tests of a Sample of Mild Steel," Report Brit. Association, 1915.

J. H. Smith, "Some Experiments on Fatigue of Metals," *Journal Iron and Steel Inst.*, Part II, 1910.

cycle in accordance with point f, so that the variable component is increased from S_v to kS_v. (See Fig. 5.) The constant k should correctly have a value somewhat less than the factor of stress concentration,[12] but we shall err on the safe side by using for k the factor of stress concentration as far as this is known.

As our next step we remove the uncertainty in the line of failure, Fig. 9, by replacing it with a straight line connecting the endurance limit S_e with the yield point S_y. (See Fig. 10.) This again introduces an error, but as far as experiments indicate, it is also on the safe side. Furthermore, the yield point is already established as a point of failure by (a), so that the end points of the assumed line of failure are definite.

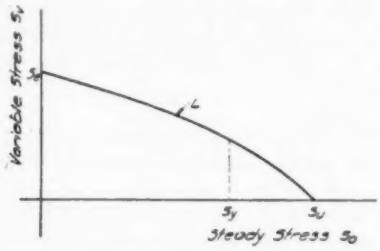


FIG. 9 EXPERIMENTAL LINE OF FAILURE

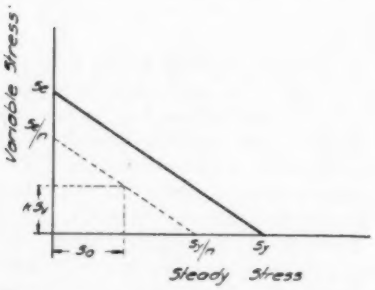


FIG. 10 ASSUMED LINE OF FAILURE

By factor of safety n we define the ratio of the stress which produces failure, to the actual stress. Thus, for a constant stress S_0 we have $n = S_y/S_0$, and for the variable stress kS_v we have $n = S_e/kS_v$. In other words, if the endurance limit and the yield point were reduced to one nth of their actual values, failure would occur under the stress cycle S_0 and kS_v. As a result of this we find that all stress combinations which fall on the straight line S_e/n–S_y/n are characterized by a factor of safety of n. By this reasoning we have extended the definition of factor of safety to any stress combination.

This result may be given a simple mathematical form. If we determine the value of kS_v from the geometrical relations in Fig. 10, we find that

$$kS_v = \frac{\frac{S_e}{n}}{\frac{S_y}{n}}\left(\frac{S_y}{n} - S_0\right)$$

or

$$\frac{1}{n} = \frac{S_0}{S_y} + \frac{kS_y}{S_e} = \frac{1}{n_0} + \frac{1}{n_v} \quad \ldots\ldots\ldots\ldots [31]$$

The expressions n_0 and n_v are known quantities: n_0 is the factor of safety when S_0 is acting alone; n_v is the factor of safety when kS_v is acting alone.

[12] Ragnar Liljeblad, "An Investigation of the Fatigue of Metals Due to Locally Concentrated Stresses." Svenska Ingeniörsvetenskapsakademiens Handlingar No. 47.

The idea underlying the usual definition of the factor of safety is that the stress can be increased in the ratio n:1 before failure occurs. Referring to Fig. 10 we find that the definition expressed by Equation [31] presents a parallel to this idea; both stresses can be increased in the ratio n:1 before failure occurs.

Now, if we define the factor of utilization by $u = 1/n$, Equation [31] becomes

$$u = u_0 + u_v \quad \ldots\ldots\ldots\ldots [32]$$

and this equation expresses the fact that the total utilization of the material is equal to the sum of the utilization due to the stress components.

It might be said, therefore, that the simplification of the line of failure into a straight line is equivalent to bringing the effect of the two stresses S_0 and S_v within the law of superposition.

This is a very desirable simplification, but it is important to bear in mind that it has no other justification than that of experimental results. For example, certain materials have a line of failure which is a curve resembling an ellipse with the axes at S_e and S_y. If this were actually the case, the line of failure might be given the equation

$$\left(\frac{S_0}{S_y}\right)^2 + \left(\frac{kS_v}{S_e}\right)^2 = 1 \quad \ldots\ldots\ldots\ldots [33]$$

Now, if S_e and S_y are reduced to uS_e and uS_y, corresponding to a utilization of the material of u, we obtain a curve of the corresponding stresses S_0 and S_v having the equation

$$\left(\frac{S}{S_0}\right)^2 + \left(\frac{S_v}{S_e}\right)^2 = u^2 \quad \ldots\ldots\ldots\ldots [34]$$

so that in this case

$$u = \sqrt{u_0^2 + u_v^2} \quad \ldots\ldots\ldots\ldots [35]$$

For certain materials this may, in fact, be a more correct definition of u, but it does not lend itself so well to extensions to more complicated cases of stress applications, as does the definition expressed by [32].

THE GENERAL CASE OF THREE PRINCIPAL STRESSES

Having thus established the fundamental consequences of these principles for the simple problem of a bar under uniform stress, the next object is to extend this definition to more complicated cases of stress applications. For this purpose we make use of the result expressed by e, which simply extends the maximum-shear theory to the case of variable stress.

Consider, then, a cube in the material subjected to three components of stress, each having variable and constant parts as indicated in Fig. 6. The $\pm$ sign expresses the fact that the component is alternating, and the constants k_1, k_2, and k_3 are the respective factors of stress concentration. We shall assume that the relative phase positions of the three stresses are unknown so that it is necessary to select the worst combination.

The problem is to find a plane in which the factor of safety is lowest. It is obvious at once that the plane must be inclined 45 deg. to the principal axis, but there is uncertainty as to which diagonal plane of the cube it should be. This can only be established by trial. Assuming it to be the shaded plane in Fig. 6, we find that the shearing stress in this plane is given by

$$\frac{S_1 \pm k_1 S_{v1} - (S_3 \pm k_3 S_{v3})}{2}$$

This shearing stress has a constant component

$$\frac{S_1 - S_3}{2}$$

and a variable component

$$\pm \frac{k_1 S_{v1} \mp k_2 S_{v2}}{2}$$

The factor of safety for this plane is thus expressed by

$$\frac{1}{n} = \frac{S_1 - S_2}{S_y} + \frac{k_1 S_{v1} + k_2 S_{v2}}{S_e} \quad \ldots\ldots [36]$$

and this relation is sufficient to determine the working stresses for this general case.

The General Case of Tension and Shear

The case of tension and shear is of more immediate interest because this is the case upon which working stresses in shafts must be based.

Referring to Fig. 7, we consider a cube of which four sides only are under stress. It is subjected to the tension stress $S_1 \pm k_1 S_{v1}$ and a shearing stress $S_s \pm k_s S_{vs}$.

Again the problem is to find a plane in which the factor of safety is the smallest or the factor of utilization the greatest. Here we know without trial that the plane must be parallel to the surfaces upon which the stresses are acting. Assume that its normal is inclined an angle θ to the direction of S_1. The problem is then reduced to a problem of minimum; θ must be determined in such a manner that the factor of safety for this particular plane is a minimum.

The shearing stress in the plane θ is obtained by projecting all the forces on the plane. This gives

$$S_{s\theta} = \frac{S_1 \pm k_1 S_{v1}}{2} \sin 2\theta + (S_s \pm k_s S_{vs}) \cos 2\theta \ldots [37]$$

This shearing stress consists of a steady component

$$\frac{S_1}{2} \sin 2\theta + S_s \cos 2\theta$$

and a variable component

$$\pm \left(\frac{k_1 S_{v1}}{2} \sin 2\theta \mp k_s S_{vs} \cos 2\theta \right)$$

To establish the factor of safety for the plane θ we have merely to apply Equation [31] with these components. This gives

$$\frac{1}{n} = \frac{S \sin 2\theta + 2S_s \cos 2\theta}{S_y} + \frac{k_1 S_{v1} \sin 2\theta + 2k_s S_{vs} \cos 2\theta}{S_e} \quad [38]$$

when the most unfavorable sign of S_{vs} is selected.

The only operation that remains is to vary θ in such a manner that n becomes a minimum. This is the case when

$$\frac{d\left(\frac{1}{n}\right)}{d\theta} = 0 \quad \ldots\ldots [39]$$

After rearranging, this equation gives the following value of θ:

$$2\theta = \tan^{-1} \frac{S_1 + k_1 S_{v1} \frac{S_y}{S_e}}{2\left(S_s + k_s S_{vs} \frac{S_y}{S_e}\right)} = \tan^{-1} \frac{K_1}{K_2} \frac{S_1}{2S_s} \ldots [40]$$

where

$$K_1 = 1 + k_1 \frac{S_{v1}}{S_1} \frac{S_y}{S_e} \qquad K_2 = 1 + k_s \frac{S_{vs}}{S_s} \frac{S_y}{S_e} \ldots [41]$$

An investigation into the sign of the second derivative reveals the fact, moreover, that this value of θ really gives a minimum.

It is interesting to note that the location of the plane of failure is a function of not only the values of the stresses, as is usually the case, but also of the values of S_e and S_y.

The particular value of n for this plane is obtained by introducing this value of θ into the above equation. This gives

$$\frac{1}{n} = \frac{\sqrt{(K_1 S_1)^2 + (2K_2 S_s)^2}}{S_y} \quad \ldots\ldots [42]$$

The experimental results available on this subject[13] indicate that the results obtained by Equation [42] are considerably on the safe side. From Equation [42] it is evident that the equivalent working stress is given by

$$\sqrt{(K_1 S_1)^2 + (2K_2 S_s)^2}$$

This result is identical with the empirical formula proposed in the Code. We can thus state that the existence of a variable stress component S_0 in addition to the already existing constant component S_0 is satisfactorily taken care of by "weighting" the constant component with the factor

$$1 + k \frac{S_v}{S_0} \frac{S_y}{S_e}$$

The author's proposition for modification of the Code now consists in removing the empirical nature of the constants K, and actually express them as functions of the variables of which they are built up.

It might be argued that neither the constants of stress concentration nor the endurance limit are always known for shaft designs. The only answer that can be given to this objection is that wherever these quantities cannot be estimated, the shaft design must remain guesswork as far as the variable stresses are concerned.

In reality both the constants of stress concentration and the endurance limits for a variety of materials have been determined by several large firms interested in shaft design.

In conclusion, this may be illustrated by a concrete example. In the Westinghouse Electric & Mfg. Co. certain classes of shafts are being designed on the following basis:

(*a*) The bending stress S_1 is completely alternating and the shaft fillets are standardized so as to give a factor of stress concentration $k_1 < 1.7$.

(*b*) The torsion stress is assumed to consist of a constant component S_s plus a variable component $0.1\ S_s$. The factor of stress concentration for torsion has been found to differ very little from that for bending, for standard fillets.

Under these assumptions the constants K_1 and K_2 assume the following values for a few of the most important shaft steels of this form:

Steel	Grade	K_1	K_2
A	Axle steel, normalized	2.40	1.24
B	Forged steel.....	2.50	1.25
C	Ni steel.....	2.30	1.23

The selection of the factor of safety for the different applications of these steels is a matter which is left to the individual judgment of the responsible designer. All that the above reasoning accomplishes is to give, for each case considered, the meaning of the term "factor of safety."

[13] F. C. Lea, "Combined Torsional and Repeated Bending Stresses," *Engineering*, Aug. 20, 1926.

Discussion

R. V. Baud.[14] The writer agrees with the author that the first step in drawing up a code on working stresses will consist of establishing clear definitions of the terms and symbols to be used.

The author makes suggestions in this respect, but he deserves some criticism in that some of them are not clear. In defining the factor of safety, Equation [1] for instance, he uses the expression "actual stress S." Presumably "actual stress" means a computed average stress, since in some of the equations given in the paper S is multiplied by a factor of stress concentration k, which is defined as the ratio between the maximum stress and a computed average stress. The latter stress does not really exist except in very simple cases such as uniform one-axial tension, etc. Therefore, the computed average stress is not an "actual stress" as a rule, but a hypothetical stress and ought to be defined as such. There the trouble begins, however, because it will be difficult to give a definition for this hypothetical stress to hold generally, and consequently designers may interpret it in a different manner. This possibility evidently should be excluded. Any improvement in the foregoing would involve the discussion of the subject matter of Appendixes Nos. 1 and 2, which is beyond the scope of this discussion.

The definition of the "factor of utilization" as proposed by the author would be a useful one provided the numerator and denominator of Equation [2] are each given within a few per cent more or less. In the great majority of engineering problems, however, this is not the case, on account of the large expense involved in making all the numerous tests required in order to obtain the values of numerator and denominator within the accuracy mentioned. The writer therefore does not see any constructive improvement in introducing this new definition, at least so long as so many engineering problems include numerous unknown quantities. He agrees, however, that the engineering profession should attempt to give the factor of safety a well-defined meaning.

A. Nadai.[15] The writer would like to comment on this interesting paper regarding some of the principles which may have a further relation to the limiting factors determining the forces acting on a structure. Although the author himself states that besides stress, different factors have to be taken in account, such as rigidity, wear, etc., if the rules defining "working stresses" shall be established, it seems that the author of the paper gives a preference to one of these factors—that is, to the term "working stress"—and to be more exact, he thinks that failure mostly depends rather on the reaching of a certain stress defined by material properties than on other limiting factors. In the writer's opinion the author limits the subject a little too much to one of the possible and fundamental quantities which affect failure in a structure or which determine the limiting values of the forces acting on a given structure. Even if one does not consider dynamical causes, such as wear or the consequences of vibrations, numerous cases can be mentioned in which the limiting values of the forces acting on a construction do not depend on the actual value of the maximum stress, but on some other mechanical quantities. One is, for example, deformation. In many cases the magnitude of an elastic transverse deformation (deflection of a beam) determines rather the limiting value of a force than a stress. In slender beams, for example, the greatest bending stress may be unimportant, while the greatest deflection may attain values which will be considered as too high. A factor of safety based on stress cannot express the relation of the forces which must hold if the deflection of the beam is to remain between given limits.

Another criterion of importance to which the engineer should devote his attention if the conditions of failure in general are discussed, is the elastic instability of the equilibrium of the forces acting on parts of structures. One of the best-known examples is the buckling of straight bars under axial loads. In more complicated systems many analogous cases of instability are possible and of practical importance. A load may become critical—that is, may cause buckling—and consequently the bending stresses may increase rapidly without an appreciable change of the load. For such types of failure the condition which has to be considered again does not contain quantities depending on limiting stress components, such as yield or breaking stresses.

G. M. Eaton.[16] The points to which the writer would like to call attention in connection with the paper have no direct bearing on the major issue presented by the author. They all bear on the engineer's quantitative factor of safety or of utilization. They are, however, of sufficient importance that no opportunity of bringing them to attention should be permitted to pass.

The first point is internal or residual stress. This is a vital factor, and one requiring constant attention. It becomes of increasing importance as the size of the actual piece under consideration increases. This is due to the fact that a great mass of testing information is based on small test pieces from which residual stress has been, in a great measure, eliminated by the act of machining the test piece.

The next point is a realization of the effect of surface conditions. If the article which is being designed has a finish inferior to the finish of the test piece, it will have physical characteristics which from this cause alone will appear inferior to those appearing from the destruction of the test piece. This is particularly true in connection with fatigue-test results. If the surface of the article is the surface left by hot-forging or rolling operations, surface decarbonization is a factor of vital importance.

Again, the location in the ingot of the piece from which the contemplated article is to be made is important. We know of no concerns who give consideration to the location of steel in the ingot. We believe that a very great advance in the art of steel application could be made, particularly by large companies, if they would purchase the product of entire ingots, having each bar, billet, etc. marked with its location in the ingot in a manner similar to that followed in the production of railroad rails. They could then apply the steel most favorably located in the ingot to the service where results of failure would be most serious, degrading the parts known to be inferior to less vital service. This is a radical procedure, and yet one is more and more impressed as time passes that it is a very logical procedure.

Particularly in heat-treated parts, residual stress resulting from thermal treatments is not all relieved by drawing operations, and there are many cases where it is entirely possible to so adjust the rate of cooling in the quench that the residual stress remaining after the drawing operation is of a friendly nature rather than of a hostile one.

Where what may be termed "pot luck" is accepted—the engineer simply calling for a quench and draw—it is entirely possible, and in fact very frequently happens, that the residual stress is hostile in that it has used up internally much of the fundamental ability of the steel to resist stress. There are many cases where it has been possible to quench a part which persistently fails in tension in service in such a manner that the entire loca-

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tion of failure is in compression when the piece starts to work. This thought, of course, is old in principle, but is new in many designs.

A. M. WAHL.[17] It should be noted that, as the author points out, the definition of factor of safety, n = ratio of stress which will cause failure to actual stress, is somewhat unsatisfactory. In fact, the value of n depends on the kind of stress involved. For example, the value of stress which will cause failure will usually be smaller if only variable stresses are acting than if only static stresses are acting. Assuming that the material is subjected to a given actual maximum stress, it is plain that the ratio of the stress which will cause failure to the actual stress will depend on whether the stresses which act to cause failure are static or variable.

Assuming that the actual line of failure is as represented by the line $S_e S_y$ in Fig. 10, what the factor of safety n, as defined by the author, really means is that both the variable stress and the constant stress may be increased in the same ratio n before failure will occur. In other words, if we have a factor of safety n as defined, we can increase the variable stress to a value n times its actual value at the same time that we increase the constant stress to a value n times its actual value before failure will occur.

Regarding Mr. Eaton's remarks on the effect of surface finish, it seems to the writer that this factor is very important. Recently fatigue tests have been made in England on spring-steel plates in the unmachined condition. It was found that the actual endurance limit was one-half to one-third of that estimated on the bases of fatigue tests of machined bars of spring steel. The writer knows of several other instances where this effect has been noted.

Therefore it seems that the value of n as defined by the author should be determined wherever possible on the basis of fatigue tests made with the surfaces as in the actual condition. If such tests are not available, then a factor of safety n of a value sufficiently large to cover all uncertainties must again be used.

S. TIMOSHENKO.[18] The problem of choosing an adequate factor of safety in design is of the utmost practical importance. If this factor is taken too low, making the working stress too high, the structure may prove weak in service. On the other hand, if the working stresses are too low, the structure becomes unnecessarily heavy and uneconomical. The factor of safety depends on the accuracy with which we know the external loads acting on a structure, on the homogeneity of the material, and on the accuracy of the formula which gives the maximum stress. In the paper it is assumed that the forces are established on the basis of experience with past practice, and that the mechanical properties of the material are known. Methods are then considered for determining the effect of various kinds of stress conditions on the choice of working stresses. This knowledge enables us to design a structure in such a manner as to have the same factor of safety in all parts of the structure. It is obvious that this latter requirement must always be fulfilled if the design is to be economical, because the ultimate strength of a structure is determined by the strength of the weakest part. As a basis for comparing various stress conditions, the yield point together with the maximum-shear theory is chosen for the condition of constant stresses, and the endurance limit together with the maximum-shear theory[19] for the case of variable stress. These methods of comparison on the basis of stress conditions are supported by experiment, and it seems logical to introduce them into design practice. The rules of the Westinghouse Company represent one of the first attempts in this direction to develop a logical system of working stresses in machine design.

It will be of great practical importance to have, in the future, a discussion of the effect of these rules on the design. The choosing of the straight line in Fig. 10 as a basis for determining the factor of safety in the case of variable stresses simplifies the problem and enables a designer to obtain working stresses for the more complicated cases, such as combined bending and torsion, in which both components are usually variable.

In the paper it is assumed that the dimensions are determined by strength considerations only. There are sometimes additional requirements which must be considered in design. There are cases in which a limiting deflection is prescribed and where this must be taken as a basis for calculating the dimensions. The rigidity is especially important for cases in which vibration of the system is to be considered. Sometimes there are requirements regarding the maximum deflection of beams or girders. Shafts must sometimes satisfy requirements regarding the angle of twist per unit length.

Sometimes the stresses are not proportional to the external loads, and in these cases it is logical to determine the factor of safety with respect to the load producing failure rather than to the stress-producing failure. We have examples of such conditions in discussing simultaneous action of bending and direct loads.

In the case of slender bars and thin plates submitted to the action of compression, the stability of the structure may be the controlling factor, and the working stress must be chosen on the basis of the critical stress rather than the yield point of the material.

From the foregoing discussion it may be seen that the choice of working stresses is a very complicated problem. In establishing the factor of safety, the designer must always be guided by past experiences. The methods outlined in the paper giving a comparison of the working stresses for various stress conditions are not intended to replace the use of past experience, but may be helpful in interpreting this experience. They may be useful also in making comparisons of different designs and in comparing the strength of existing structures.

The study of actual failures and the investigation of the causes of these failures in the light of such theory, as presented by the author, form a very useful method for acquiring a better knowledge of the strength of our structures. Combining such an analysis of failures with theoretical investigations of the stress distribution in various cases and with laboratory investigations of the strength of materials under various stress conditions, will enable us to accumulate a more reliable knowledge of the actual strength of structures. When we have such knowledge, the present specifications for working stresses in the various branches of engineering can be considerably improved. This will, without doubt, result in an economy of material and in greater reliability for structures and machines.

HAROLD C. CLAUSEN.[20] The writer would like to draw attention to the question of working stresses in connection with the section "Fundamental Laws of Failure," paragraph (*d*), which says: "When the stress is alternating, the failure is predicted by a material constant, the endurance limit, which can be considered fully defined and measurable, for practical purposes."

Now in engineering practice such as occurs in railroad work, we have many examples where the stress is not subject to a complete reversal and where it is repeated many times, varying from a tension to zero or a compression or zero, or it may vary

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[19] The writer agrees with using this theory in design, although the statement that failure due to alternating stress consists of ruptures in the plane of maximum shear is not correct.

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from a high value of tension or compression to a low value of tension or compression. In these cases it will be necessary to multiply the endurance limit by some factor. This factor has been put forward in the shape of a formula in two recent papers: (1) "What Happens When Steel Gets Tired," by D. D. Ewing, *Electric Railway Journal*, August, 1929, and (2) "Materials in Machine Construction," by H. F. Moore, *Mechanical Engineering*, October, 1929. The writer would be glad to learn what value the author of the paper suggests.

A further factor may be necessary to take account of the nature of the way the piece is manufactured: for instance, whether the piece is a forging and is sheared on some part of its surface by a trimming die, or is a piece completely machined and polished. There is also the question of size and of the inclusion of foreign matter. These latter points have been brought up very forcibly by Mr. Eaton, and the writer thinks they are of primary importance.

A still further factor may be necessary to take into account the fact that there may be some periods of rest between repeated loadings.

H. F. Moore.[21] The author's suggestion that the term "factor of safety" be discarded and that its reciprocal be used under the name "factor of utilization," is very interesting and is worthy of consideration. Of course, even if the new term seems to be the better one, it will have to overcome a great deal of inertia of common usage before it can come into use.

In considering the sample codes which the author quotes (without definite recommendation for adoption), the opinion of the writer would be quite different as to (1) their practical value and reliability and (2) the rigid correctness of the theories of failure which they rather dogmatically state to be true.

As practical codes for determining working stresses both appendixes seem good, with the possible exception of the statement in Appendix No. 1: "When at least one of the principal stresses is a tension stress, and the maximum compression stress (if there is one) does not exceed the ultimate strength in tension S_{ut}, *only the largest tension stress contributes to failure*,[22] and the material may be said to fail according to the maximum-strength theory."

This statement the writer believes to be lacking in confirmatory experimental evidence and to be on the "danger" side of the maximum-shear theory, the maximum-strain theory, and the maximum strain-energy theory. The principle implied in both appendixes of codifying the relative magnitudes of factor of safety for different stress combinations, while leaving the determination of the actual numerical value to the judgment of the designing engineer, seems thoroughly sound.

However, the writer thinks it doubtful whether it has been rigidly proved that the failure of ductile materials is always due to shearing stress, as is stated in Appendix No. 2, or whether under certain combinations of tensile and compressive stresses at right angles the maximum-stress theory holds rigidly, as seems to be indicated in Appendix No. 1.

It is doubtful whether any theory of failure based on the theory of elasticity, with its obviously inaccurate assumptions of homogeneity and indefinite divisibility of material, will be found to be rigidly true. It would seem that the rather dogmatic statements concerning the truth of the shear theory for ductile materials and of the truth of the maximum-stress theory for certain stress combinations in brittle materials might well be modified slightly, without detracting at all from the usefulness of the codes developed.

In connection with theories of failure, attention may be called to the theory advanced by Prof. Parker Haigh, the British metallurgist,[23] which holds that the criterion of failure of a material is the amount of energy stored up per cubic inch at any region of a stressed machine or structural part. This theory seems to come nearer agreeing with test data for the whole range of possible combinations of stress than does any other single theory of failure.

John C. Wattleworth.[24] In the author's statement regarding the two subdivisions of the strength calculation of machine parts, a third subdivision might be added, namely, a determination of the amount of the factor of safety. Much of the value of the immense amount of theoretical and experimental work on the determination of stresses and mode of failure is invalidated by lack of definite thought on this point. In other words, analytic or semi-analytic methods of evaluating the conditions peculiar to the service of a machine part into a factor of safety are desirable. It is fully realized that much of empiricism must exist in this direction.

Regarding the author's remarks on the probable insistence on specifications for working stresses being given definitely for the various subtitles in his classification, Fig. 1, and standard materials, the writer can sympathize heartily with the author's statements on this point. An experience in the teaching of machine design inevitably involves questions by members of the classes asking for recommendations for factors of safety. Some of the texts on this subject attempt to give such values, based on the classification the author suggests. It is the writer's feeling that this practice should be discouraged; and the writer has evolved a semi-analytic method, which at least for his own purposes appears to be highly satisfactory. This is given at the end of the discussion. In this connection it may be remarked that the writer started using this method six years ago in his classes and that he is fully aware of several similar analyses which have appeared from time to time. He, however, cannot claim indebtedness to any of the other similar methods for his treatment of the subject.

The author's phrase "factor of utilization" is preferable to the term "factor of safety." It is recognized, as the author states, that no fundamental advantage other than clarity of expression can result; but the advantages of the train of thought provoked by the phrase are so great that it seems highly desirable that it should attain widespread use.

An extremely useful purpose is served by the author's arbitrary classification of materials as ductile or brittle, with the limiting value of 5 per cent permanent elongation as the division line.

Regarding the material constants on which working stress is based, although every one is agreed that from the standpoint of deformation of a machine part no stress should exceed the yield point, it would seem that long-established practice might better be deferred to, and that even in the case of ductile materials the ultimate strength might properly be employed as the material constant from which to derive working stress.

Apart from the thought that an abandonment of a long-used constant might retard the adoption of the proposed rules, several practical considerations might be brought forward to support the claim. First, the correlation of endurance data with ultimate strength is much better than with yield point. If attention were focused on the latter, some sacrifice in the matter of safe values of endurance limit might result. Second, from the standpoint of the phrase "utilization factor" the ratio (yield point—ultimate strength) expresses the thought that the utilization of the material is enhanced by (*a*) the properties of the material itself or

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[22] The italics are by the writer of the discussion.

[23] See report of Section G of the British Association for the Advancement of Science, 1919, p. 486; 1921, p. 324.

[24] Assistant Professor of Machine Design, Case School of Applied Science, Cleveland, Ohio. Mem. A.S.M.E.

(*b*) the properties with which the material may be endowed by treatment. Such a factor, or rather its reciprocal, is in fact used in the writer's procedure for the derivation of factor of safety.

Regarding the treatment of the problem in Appendix No. 2, under "The Fundamental Case of Stress Cycle," several questions arise. First, is the simplification, that is, the transition shown in Figs. 9 and 10, warranted? The writer feels that on this point the arguments against the abandonment of the ultimate strength may be brought forward and that sufficient simplification will be accomplished by merely straightening the line S_e–S_u of Fig. 9. Such a procedure is in good agreement with older investigations on the effect of range of stress, as opposed to completely reversed stress. The only proviso necessary is that the active part of the diagram be restricted to that portion to the left of a vertical line passing through S_y, located on the axis of the abscissas.

A second point is as to the general form of the treatment. In the older work on the subject the term "range of stress" occupied a prominent place in the discussion, and the conclusion reached was that as the range decreased the maximum stress might increase. The numerous diagrams proposed all bore witness to this point of view.

The treatment outlined in the author's paper, that of superimposing a variable stress upon a constant stress, seems preferable. This is because the method presents clearly the picture of the effect of the degree of utilization possible by the one stress caused by the presence of the other.

The procedure in determining factors of safety which the writer has been using is given herewith, employing the author's symbols. It is hoped that it may be considered as a supplement to the author's paper on the point of assistance in the derivation of numerical values of factors of safety. No numerical values are given, for the very obvious reasons already cited and in the author's paper. On points of difference which have arisen, no great difficulty should exist.

1 The selection of a working stress involves much uncertainty which is generally met by empirical overall factors of safety. It is proposed to substitute for such factors of safety a semi-analytic method of selection. The central thought is that of expressing quantitatively the effect of each of the elements of the total factor of safety. Numerical values cannot be given as each case carries with it its own special considerations.

As ordinarily used, the factor of safety is employed to derive a working stress by the expression:

$$S_w = S_u + n$$

where S_w = working stress
S_u = ultimate stress
n = factor of safety.

Let us regard n as the product of several factors, $n_1, n_2, \ldots . n_n$ and write

$$n = n_1 \times n_2 = n_3 \ldots . n_n$$

where each of the factors n_1, n_2, etc., is intended to be given a quantitative value expressing the effect of a single consideration upon the working stress.

2 For static loads the following are suggested:

Let n_1 be the ratio $S_u + S_y$, where S_u = ultimate strength, and S_y = yield point.

In the case of brittle materials, an arbitrary value must be assumed corresponding to the yield point in ductile materials. It is of course obvious that no consideration can be given to working stresses beyond the yield point from the standpoint of permanent deformation of machine parts.

Let n_2 be a uniformity factor, which shall take account of variations of the properties to be expected in an individual piece from the average, for a given material. This may be expressed as

$$n_2 = \frac{P + t}{P}$$

where P = property in question
t = tolerance to be expected in the property.

Let n_3 be a factor which will take into consideration the well-known fact that a variation of properties may be expected on account of size. Thus, a test on which a determination of properties is made may come from a small section where favorable conditions may prevail in the matter of forging or other working. This is particularly true in the case of pieces which are to be heat-treated. Generally test specimens of such pieces are more responsive to heat treatment because of better penetration than larger pieces subject to the same treatment.

Let n_4 be taken as an insurance factor. Assuming the preceding factors as having been applied, the presumption is then that the stress in a piece so designed will, in an unfavorable case of uniformity, be on the verge of the elastic limit. Quite obviously even apart from the standpoint of deformation on the part of structure under consideration, cases where such high stresses would be satisfactory are very rare. Hence most cases require the use of excess material in order to reduce the stress in the part to a point where the designer feels secure. This excess material may be considered in the light of a premium which is paid for the sake of insurance against failure.

This factor is one which is dependent either on the designer's own judgment or on precedent. As a suggestion, several cases may be mentioned, not so much in an effort to provide numerical values for these factors as to crystallize the thought on this. These may be:

(*a*) Cases where failure involves danger to life
(*b*) Cases where failure involves risk to valuable property
(*c*) Cases where failure will interrupt extensive production processes
(*d*) Cases where the operation of the machine only is interrupted by failure.

A stress concentration factor may be called n_5. The elementary formulas for stress do not take into account local stresses of high value. In the case of static loads considered here, stress concentrations are not of much consequence for ductile materials, but are a source of danger for brittle materials. It is recommended that the designer classify such sources of local stress concentration, working out factors which he may apply to his design calculations.

To cover the case of anticipated wastage or corrosion of a machine element in service, n_6 may be introduced. This is largely a matter for the judgment of the designer.

Then n_7 may finally be termed the factor of ignorance. Even with the excellent tools at our disposal, in the form of ever-extended methods of calculations, cases occasionally arise in which the computation of stress is either impossible or in which the designer feels the computation is at best only an approximation.

In this event, whether the suggested procedure is followed or not, the designer's instinct or judgment must lead him somehow to evaluate this uncertainty into a factor which shall take the situation into account to his satisfaction.

3 For the case of repetitive loads, fatigue stresses must be considered. There is just as much reason for the application of a factor of safety to the endurance limit in this case as to the ultimate strength in the case of static loads. The factor of safety in the case of fatigue stresses, however, is formed somewhat differently.

The factor n_1 has now no meaning and may be omitted. The starting point for this work is then the endurance limit, and the

other factors $n_2 \ldots . n_7$ are effective. The stress concentration factor n_8 is to be applied in its full effect for both ductile and brittle materials.

Where stress reversal is not complete, a direction computation of the working stress from the endurance limit for reversed stresses may be rather severe. In such cases recourse may be made to one of the diagrams or formulas proposed for maximum strength expressed as a function of range of stress. To this value, then, rather than to the endurance limit for completely reversed stresses, the factor of safety may be referred.

4 Many cases arise where stiffness is of great importance, so much so that it is probable that whatever proportions will produce a design possessing the required rigidity will certainly produce sufficient strength. In such problems it is necessary to compare the stresses required from the standpoint of safety with those required by rigidity.

5 Assuming for the present that the designer has assigned values to each of the foregoing factors for a particular case, he is in a position to obtain a working stress. Let us examine the result of this analysis.

It is perfectly true that he may, in his judgment, eliminate certain of the factors mentioned, or, for the particular case under consideration, introduce new factors. In any event probably the chief arguments which will be raised against his selection by the foregoing method are that many of the factors have depended on his judgment entirely, and this being the case he might as well have applied his judgment to the whole factor of safety at once, with at least as satisfactory results.

This may be granted, but an answer to the objection is, first, that he may still apply an overall judgment, so to speak, to his result; and, second, that by using the process outlined he has considered separately the tangible and the more or less intangible elements, and he has allowed himself the opportunity of evaluating separately, and each on its own merits, every one of the considerations on which the experienced designer's judgment is the only available guide. It would seem a reasonable conclusion that the foregoing method should produce a factor of safety more rationally determined than the ordinary method.

M. Stone.[25] Having been associated with the author and others in codifying our knowledge of the behavior of materials under various working conditions, the writer would like to add to and elaborate on some of the ideas expressed in the paper.

The author's opening statement divides the subject of properly designing a mechanical structure for strength into two parts: (1) calculation of stresses and (2) significance of stress as regards failure. It is essentially the second part that this subject properly deals with. It is often in men's minds that, let us say, mild carbon steel fails at 60,000 lb. per sq. in., and that safe stress values should not exceed 25,000 lb. per sq. in. in a structure. These ideas are based on experiments made by testing such steels in tension, and form a basis for most design work. However, when such ideas are contrasted with the facts that stresses in roller bearings approach 200,000 lb. per sq. in., and in ball bearings 400,000 lb. per sq. in., along the areas of contact, and that gear teeth operate regularly close to 100,000 lb. per sq. in. at the pitch circle, the significance of the stress is quite apparent. A close investigation into the reasons for the persistence of such stress values without destruction reveals the fact that steels under three-dimensional stress may be stressed to values approaching the cohesive forces of the molecules before rupture. Joffé, in a classical experiment, lowered the temperature of a steel ball slowly to the temperature of liquid air, and on quickly drawing it out into atmospheric temperature, by dint of the rapid heating of the external surface, exceeded the yield point of the material, placing the center portion into very high three-dimensional tension.

Subsequent cutting and examination of the material showed no failure at stress values far in excess of ordinary ultimate strengths.

Very often the stress at the edge of a hole, where a pin is pulling against one side, exceeds the yield point of the material, and such a condition is considered satisfactory design practice. The appreciation of the extreme localization of stress and inability to spread are the guiding principles in such applications.

In a dovetail, the mechanical element used to fasten an electrical field pole to the armature body, stress values in excess of the yield point are tolerated at the fillets because it is realized that concentrations of stress do not enhance the possibility of failure under static stressing.

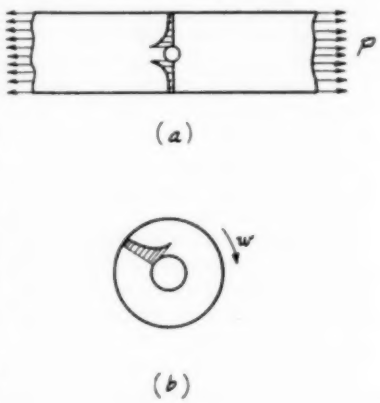


Fig. 11

The examples that have been discussed here served to emphasize the complexity of general ideas regarding applications of materials, but in the same sense must be regarded as strong arguments for a simplification of working-stress rules.

The last point, in particular, gives evidence of a certain element of difficulty in generalization. Although it is indeed a worthy goal to be able to say stress concentrations have no meaning for static applications but must be fully considered under varying stress, the question is immediately raised, when can a stress distribution be called a "concentration of stress?" This point may be illustrated by comparing the stress distributions shown in Fig. 11 (*a*) and (*b*). It is the dictate of safe designing to judge (*a*) a condition of stress concentration, while in (*b*) the design must be limited by the peak stress at the inner bore. This point could be further developed, but is introduced here only as one of the difficulties encountered in generalizing.

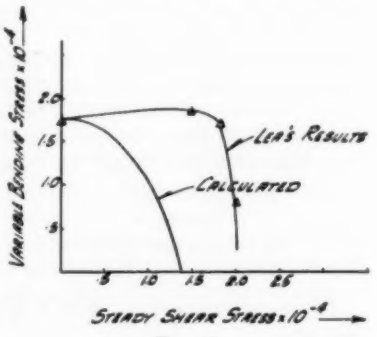


Fig. 12

In Appendix No. 1 the author has reproduced rules and arguments as to interpretation of materials under various complex stress applications. It must be realized that all such relations have as their experimental basis the failure of materials under combined steady and variable stresses in one plane only; and the failure under stress combinations on various planes and with shear have been worked out on the basis of stress combinations that follow from the theory of elasticity. This, it must be admitted, is all that can be offered in a rational treatment of the complex subject until further tests are made of the failure of materials under combined stresses, and it is only when such rather profound material researches are carried out that more definite ideas may be had.

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Professor Lea[26] in England has already made a beginning in determining the failure of shaft material under steady torsion (shear stress) and alternating bending (direct stress). His tests show rather startling results, and although not investigated for confirmation by other engineers as yet, are worth discussing further. In Fig. 12 it is seen that comparatively large values of steady shear stress may be superimposed on an already existing bending stress without precipitating failure, until a certain critical value is reached. In this same figure has been drawn the curve of failure as predicted from the discussion in Appendix No. 1. It will be noticed that the rules proposed are distinctly on the side of safety.

Probably the most important result given in the paper is that discussed under "The Fundamental Case of Stress Cycle." This has been reported on earlier[27,28] and still retains certain alternative interpretations. As the author has given it, it will be appreciated that the same factor of safety has been chosen in variable stress relative to the endurance limit as for static stress relative to the yield point. However, when it is considered that endurance limit means actual rupture while yield point means only an increase in strain, it is conceivable that a larger factor of safety might well be applied to the variable stress. Equation [31] then becomes

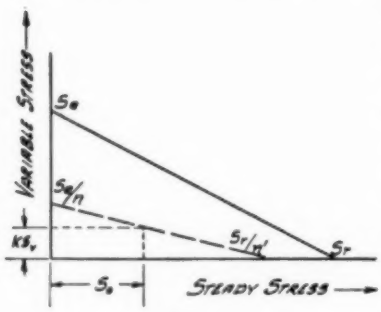


FIG. 13

$$\frac{S_s}{S_y} + \frac{N'}{N}\frac{KS_y}{S_e} = \frac{1}{N}$$

when N = static factor of safety.
N' = fatigue factor of safety ($N' > N$)

This modifies Fig. 10 to that shown in Fig. 13. However, a certain simplicity of use is lost if this modification is introduced, and it must be left to the designer to estimate the exigencies of his case. The intrinsic value of the arguments under this paragraph are very valuable and far-reaching in so far as they simplify the use of a great mass of experimental data for design purposes.

In closing, it must still be realized that in addition to the best interpretation of material properties and load stresses, there always exists some uncertainty as to the conditions that may occur in the future life of a structure. If, for instance, in a three-bearing set, the foundation were to "set" under an outer pedestal, comparatively high bending stresses would be set up in the shaft. Non-uniform temperature distributions in a structure are conducive to additional, usually non-predictable stresses. For such exigencies, actual factors of safety reaching as high as 3 must still be retained, depending on experience with particular structures.

However, the general reaction to the author's suggestion regarding organized action on "working stresses" must be a positive one, and it is well established that with the beginning that has been made and the future work that is indicated much benefit will result.

[26] F. C. Lea, "Combined Torsional and Repeated Bending Stresses," *Engineering*, Aug. 20, 1926.

[27] S. L. Henderson and C. R. Soderberg, "Recent Improvements in Turbine Generators," Trans. A.I.E.E., 1928.

[28] M. Stone, "Stress Analysis in Electrical Rotating Machinery," Trans. A.S.M.E., 1928.

AUTHOR'S CLOSURE

Mr. Baud's criticism with respect to the vague meaning of the term "actual stress" is justified, but the author would like to point out that the definition of factor of safety put forward in the paper is not new and has always been afflicted with this ambiguity, nor will anything short of a complete solution of the entire problem disperse it. To be able to define the factor of safety (or factor of utilization) in a general statement means that there should always exist a single stress which, when increased to a certain limit, causes failure in the sense that this term is understood. Such a stress can be said to exist only in simple stress applications such as straight tension, bending, torsion, and a few cases of combined stresses. For ductile materials under constant loads the stress that comes nearest to this in significance is the stress figured by ordinary elementary formulas. Local concentrations of stress apparently have appreciable influence only in the case of variable stresses and in brittle materials. Under these circumstances it seems defensible to base the definition of factor of safety upon some such average stress, which must be defined separately for each case. Obviously, Equation [1] presupposes the existence of such a stress, and although this may be debatable in many cases, it is justified by the many important simple ones where experience shows the supposition to be essentially correct.

With respect to the remark by Dr. Nadai that too much significance has been given to stress among numerous other design factors, the author wishes to point out that the paper is limited intentionally to a discussion of stresses as causes of failure. This has not been done with the implication that the term "working stress" should be given general preference over other factors such as deflection, vibration properties, stability, etc. These are matters to be decided in each case by the responsible designer. The statements of the paper apply to such cases where stresses are the dominating elements.

The number of cases where stress is the deciding factor is so great, however, that the term "working stress" might well be given a preferred place in the list of factors which have to do with failure. In the electrical industry in particular, where expensive materials are applied to very large units, the rational utilization of these materials is a problem of real economical importance.

Mr. Eaton's remarks on internal stress, surface condition, and location in the ingot are to the point and illustrate the difficulties that obstruct the path toward quantitative evaluation of the factor of safety. All three subjects must necessarily influence the designer's judgment as to the utilization of the properties of strength, but it seems that the method of defining this utilization quantitatively need be influenced by the internal stresses only. Where surface conditions and location in the ingot are of deciding importance, the tests must be made in such a manner that the material constants, upon which the utilization is ultimately based, reflect the influence of these factors.

Referring to the question propounded by Mr. Clausen on the working stress for combinations of constant and variable stress, it seems that the rule given by Equation [16] might be used as a guide.

The author agrees with the statements of Dr. Timoshenko and Professor Moore that certain more or less doubtful phases of the theory of failure have been somewhat too dogmatically stated to be true. It certainly has not been proved that fatigue failures always occur in the plane of maximum shear, nor is the maximum-shear theory unassailable. Least of all can there be said to exist a definite experimental justification for the treatment of brittle materials given in Appendix No. 1. Until more knowledge is collected on the subject, however, it is difficult to obtain equally simple and safe rules which more accu-

rately follow the general trend of the experimental results. The author is prepared to admit that the wording of a possible code on the subject may have to take the form of more or less arbitrary assumptions, justified by references to recognized experimental results. Naturally, a code must not mean stagnation of progress; it should be merely a medium through which the latest results are interpreted in a simple form for the practical engineer.

Professor Wattleworth's proposal to base working stresses on ultimate strength rather than yield point seems to the author to do injustice to the case of constant stress. Here the yield point is a true limiting stress for the most important ductile materials, and represents 100 per cent utilization of the properties of strength. It is true that in some cases endurance data may be correlated with ultimate strength rather than yield point, but this does not appear to the author a sufficient reason, particularly since many cases are on record where the line of failure of Fig. 9 does head for the yield point. Well-established customs should be left alone as far as possible, but it seems that the idea of basing failure under constant stress in ductile materials upon the ultimate strength should form an exception to this rule. The fundamental conception of utilizing a certain percentage of the possibilities of the materials is otherwise lost. The quantitative method of arriving at the factor of safety proposed by Professor Wattleworth seems very sound, particularly because it forces the designer to take stock of the various factors in operation. Intuitive judgment is certainly strengthened by quantitative reasoning of this general type.

Mr. Stone's proposal to "weight" the endurance limit in Equation [31] by a separate factor of safety is equivalent to reducing the endurance limit a certain arbitrary amount. The fundamental soundness of Mr. Stone's reasoning is admitted, but it is questioned whether the same result might not be obtained more easily by agreeing on some constant fraction of the endurance limit to be used in the equation.

APM-52-3

Stresses and Deflections in Flat Circular Plates With Central Holes

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Simplified formulas are derived for calculating maximum stress and maximum deflection of flat circular plates having central holes. Eight different loading and edge conditions are considered. These formulas are presented in a usable form for the practical designer. Approximate solutions are given for purposes of checking the formulas. Using the principle of superposition, a simple method of superimposing fundamental cases is developed for calculating cases other than those considered in the paper. Experimental determinations of stress and deflection for certain cases indicate that the formulas are sufficiently accurate for practical work where the edge conditions are determinate. For indeterminate edge conditions, as for example so-called built-in edges, the tests indicate that, while there is good agreement in the case of stress, the measured deflection is considerably greater than the theoretical.

FLAT circular plates with central holes are structures which are widely used in engineering work. To name a few of the more common applications, there may be mentioned Kingsbury thrust-bearing plates, diaphragms of all kinds as used in telephones, loud-speakers, steam turbines, diffusers, piston heads, certain types of cylinder head, etc.

For calculating the strength or stiffness of these structures, the ordinary flat-plate theory,[3] usually known as the Grashof theory, may be applied. The application of this flat-plate theory by the use of formulas commonly given in textbooks will, when applied to flat circular plates, be found in most cases to entrail long and laborious computation, with much chance for error and such as to make practical men and designers skeptical of the final results. In reviewing the literature on flat-plate theory, the authors were unable to find any simple formulas that may be applied easily and quickly to such commonly used cases as flat circular plates with central holes and having various loading and edge conditions.

The reason, therefore, for undertaking this investigation was to obtain, if possible, simple formulas for these cases of flat plates, such that practical men and designers could use the results of flat-plate theory without long and tedious computation. It was also desired to develop, if possible, a more simple procedure for calculating specific cases of flat plates other than those considered in this paper. This was done by the use of the principle of superposition of fundamental cases. This method, as finally developed, will require less calculation than the ordinary method. Finally, it was desired to obtain experimental verification of at least part of the formulas given in this paper. To this end the paper is divided into four major parts:

Part I Simplified Formulas for Eight Common Cases of Flat Plates With Holes.

Part II Approximate Solutions of the Plate Problem.

Part III The Principle of Superposition of Fundamental Cases, With Applications.

Part IV Experiments and Comparison With Theory.

PART I SIMPLIFIED FORMULAS FOR EIGHT COMMON CASES OF FLAT PLATES WITH HOLES

In order to obtain a formula for the maximum stress and deflection of a flat circular plate with a central hole, symmetrically loaded, it is necessary to integrate the general differential equation[4] for the deflection of a symmetrically loaded flat circular plate, and so determine the integration constants as to satisfy the edge conditions of the actual plate. These values of maximum stress and maximum deflection in the plate may thus be obtained in the form of comparatively simple formulas, as given farther on.[5] In order to simplify the formulas as much as possible in this work, the value of Poisson's ratio $1/m$ was assumed equal to 0.3 (the commonly accepted value for steel). However, it is believed that the results will be reasonably accurate even where Poisson's ratio is considerably different from 0.3.

The following nomenclature is used in all cases:

a = outer radius of plate = $^1/_2$ outer diameter
b = inner radius of plate = $^1/_2$ inner diameter
α = ratio $\frac{a}{b} = \frac{\text{outer radius}}{\text{inner radius}}$
P = total load uniformly distributed along edge of plate (for those cases having this condition)
p = pressure per square inch of plate surface (for those cases having a uniformly distributed pressure acting on the plate)
t = thickness of plate, inches
E = modulus of elasticity of the material, lb. per sq. in.
S = maximum stress in the plate, lb. per sq. in.
y = deflection of one edge with respect to the other, inches.

Cases Considered With Simplified Formulas

Eight different cases which commonly occur in practice were worked up. These cases are represented schematically in Fig. 1. The final formulas for maximum stress and maximum deflection as derived for each case are also given. It will be seen that in all cases the maximum stress is given by a formula of the type $S = \frac{Kpa^2}{t^2}$ or $\frac{KP}{t^2}$, depending on whether the applied load is uniformly distributed over the surface or concentrated along the

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[3] The fundamental differential equation for calculating the strength of flat circular plates symmetrically loaded is derived in several textbooks, including "Applied Elasticity," by Timoshenko and Lessells, p. 275, and also "Strength of Materials," by Case, p. 489. Very complete studies, not only on circular plates, but also on rectangular plates, are given in "Elastische Platten," by A. Nadai. Some tests and theory of circular plates without holes are reported by Max Ensslin, Dingler's Polytechnishes Journal, vol. 318, 1903. The general theory of bending of plates is also given in a paper by Westergaard and Slater, Proc. Am. Concrete Inst., 1921.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] See, for example, "Applied Elasticity," p. 274, mentioned in footnote 3, for the derivation of this equation.

[5] An illustration of this method is given in Appendix 1, which includes the derivation of the formulas for Case 1, cited farther on.

edge. Likewise, the deflection y as shown is given by a formula of the type $y = \frac{Mpa^4}{Et^3}$ or $y = \frac{MPa^2}{Et^3}$. The constant K in the stress formula depends on the ratio α of outside to inside radius of the plate and may be taken from the curves of Fig. 2. Thus for case 1 the formula for maximum stress is $S = \frac{K_1 P}{t^2}$ where K_1 depends on the ratio $\alpha = \frac{a}{b}$ for the plate considered, and is taken from the curve marked K_1 in Fig. 2; likewise the constants M in the deflection formula depend on the ratio α of the radii and may be taken from the curves of Fig. 3. Thus for case 3 the deflection is $M_3 \frac{pa^4}{Et^3}$, where M_3 again depends on the ratio $\alpha = a/b$ and is taken from the curve marked M_3 in Fig. 3.

For values of α near to unity the curves do not, in most cases, give the values of K or M accurately. In such cases a radial strip of the plate may be considered as a beam with end conditions and loading as in the actual plate. As shown farther on, this method is accurate for values of α near to unity.

Limitations of Foregoing Formulas

It should be noted that the flat-plate theory, by means of which the foregoing formulas were derived, assumes that the thickness of the plate is small relative to the diameter and that deflections are small relative to thickness. This means that deflections due to shear as well as stresses set up by stretching the middle surface of the plate were neglected. It is, however, believed that where the deflection is not over one-half the thickness and where the thickness is less than one-third the difference between inside and outside diameters for simply supported edges and less than one-sixth the difference between inside and outside diameters where one or more edges are built in, the foregoing formulas hold with sufficient accuracy for practical work. It should, however, be noted that a truly built-in edge does not exist in practice, and for this reason the formulas for deflection of plates with built-in edges must be considered as approximations only, although it is probable the stress formulas for such cases are more accurate. Where the thickness is greater than one-third difference in diameters for simply supported edges or one-sixth the difference in diameters for built-in edges the deflection due to shear may be taken into account by adding on to the deflection as found by the foregoing plate formulas the following deflections,[6] G being the shear modulus of elasticity:

$$y_s = \frac{0.239P \log_e \alpha}{tG}$$ (for plates with load P uniformly distributed along edges, cases 1, 6, and 8)

$$y_s = \frac{0.375pa^2}{tG}\left[1 - \frac{1}{\alpha^2} - \frac{2 \log_e \alpha}{\alpha^2}\right]$$ (for plates with uniform load p on surface and supported on outer edge, cases 4 and 7)

$$y_s = \frac{0.375pa^2}{tG}\left[2 \log_e \alpha - 1 + \frac{1}{\alpha^2}\right]$$ (for plates with uniform load p on surface and supported on inner edge, cases 2, 3, and 5)

Moreover, as mentioned, for values of α near to unity the curves of Figs. 2 or 3 do not, in most cases, give the values of K or M accurately. In such cases the procedure recommended may be used.

Case 1 Loaded along inner edge; simply supported at outer edge; P = total load on edge.



Case 2 Uniformly loaded; simply supported at inner edge; p = uniform load on surface (lb. per sq. in.).



Case 3 Uniformly loaded; built-in at inner edge; p = uniform load on surface (lb. per sq. in.).



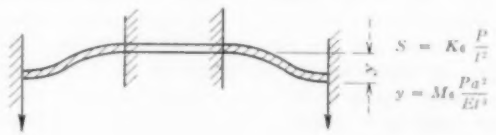
Case 4 Uniformly loaded; simply supported at outer edge; inner edge prevented from rotating; p = uniform load on surface (lb. per sq. in.).



Case 5 Uniformly loaded; built-in at inner edge; outer edge prevented from rotating; p = uniform load on surface (lb. per sq. in.).



Case 6 Loaded along outer edge; built-in at inner edge; outer edge prevented from rotating; P = total load on edge.



Case 7 Uniformly loaded; simply supported at outer edge; p = uniform load on surface (lb. per sq. in.).



Case 8 Loaded along outer edge; built-in at inner edge; P = total load on edge.



Fig. 1 Various Cases of Loading and Edge Conditions Considered
(a, outer radius; b, inner radius; t, thickness in inches.)

Practical Example

As an example of the application of these simplified formulas, consider a loud-speaker diaphragm of 0.017-in. steel sheet with

[6] See Appendix 2 for method of derivation of these equations.

an outer radius $a = 1.12$ in. and an inner radius $b = 0.5$ in. The inner edge is clamped, and the load applied by electromagnetic forces is 5.8 lb., uniformly distributed along the outer edge. Required: the maximum stress and deflection of the diaphragm.

It will be noticed that these are the edge and loading conditions of case 8. In this case $\alpha = a/b = 2.25$. From the curve of Fig. 2 we find for $\alpha = a/b = 2.25$ a value $K_8 = 0.9$. Hence the maximum stress (from the formula given opposite case 8, Fig. 1) is:

$$S = K_8 \frac{P}{t^2} = \frac{0.9 \times 5.8}{(0.017)^2} = 18{,}000 \text{ lb. per sq. in.}$$

Also from the curve of Fig. 3 we find for $\alpha = a/b = 2.25$ a value $M_8 = 0.12$. Hence the maximum deflection (from the formula opposite case 8, Fig. 1) is:

$$y = M_8 \frac{Pa^2}{Et^3} = \frac{0.12 \times 5.8 \times (1.12)^2}{30 \times 10^6 \times (0.017)^3} = 0.006 \text{ in.}$$

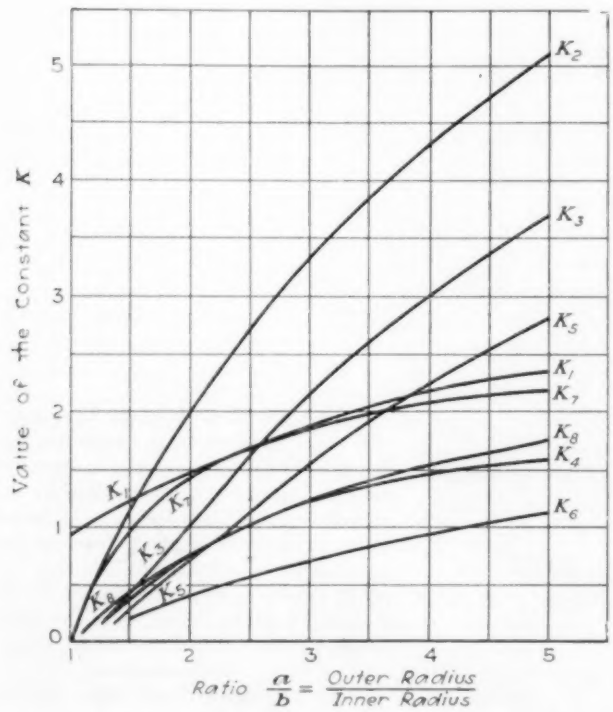


FIG. 2 CURVES FOR FINDING K IN THE STRESS FORMULA FOR FLAT PLATES $S = K\frac{P}{t^2}$ OR $S = K\frac{pa^2}{t^2}$

In this a modulus $E = 30 \times 10^6$ lb. per sq. in. is assumed.

The foregoing example should make clear the procedure in calculating similar cases.

ALGEBRAIC VALUES OF THE CONSTANTS K AND M

The values of the constants K_1, K_2, ... K_8 and M_1, M_2, ... M_8, from which the curves of Figs. 2 and 3 were plotted, were found, as mentioned, from the ordinary flat-plate theory, taking Poisson's ratio $1/m = 0.3$. The final results in algebraic form were as follows:

$$K_1 = 0.3343 + \frac{1.242\alpha^2 \log_e \alpha}{\alpha^2 - 1}$$

$$K_2 = \frac{3.9\alpha^2 \log_e \alpha + \frac{0.525}{\alpha^2} + 0.9 - 1.425\alpha^2}{\alpha^2 - 1}$$

$$K_3 = \frac{3.9\alpha^2 \log_e \alpha + \frac{0.525}{\alpha^2} - 1.425\alpha^2 + 0.9}{0.7 + 1.3\alpha^2}$$

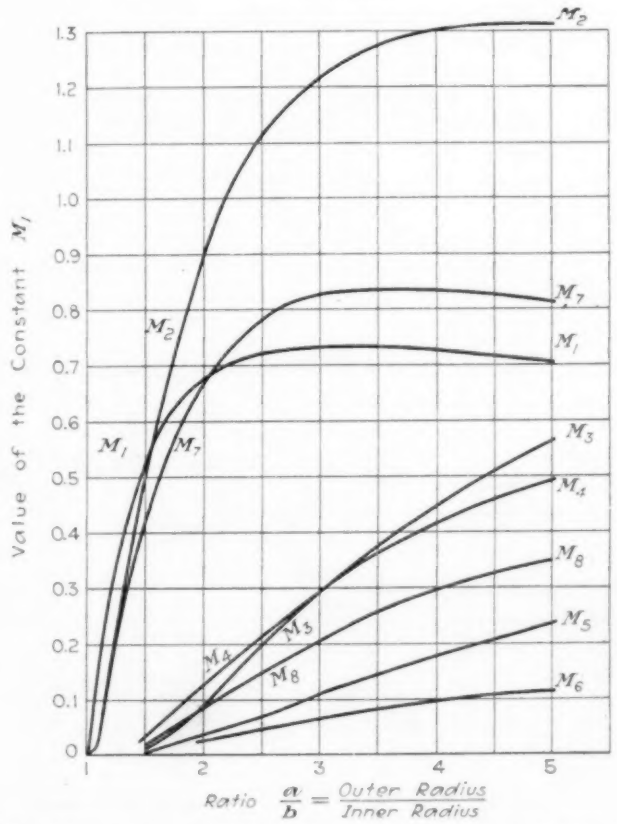


FIG. 3 CURVES FOR FINDING M IN THE DEFLECTION FORMULA FOR FLAT PLATES $y = M\frac{Pa^2}{Et^3}$ OR $y = M\frac{Pa^4}{Et^3}$

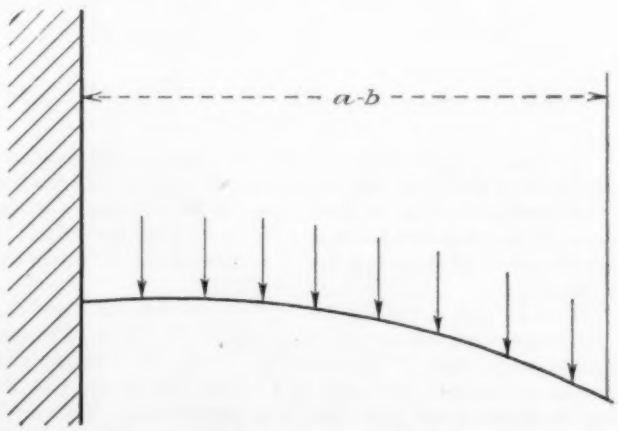


FIG. 4 LOAD ACTING ON CANTILEVER STRIP

$$K_4 = \frac{\frac{0.525}{\alpha^2} - 3.9\log_e \alpha - 3 + 2.475\alpha^2}{1.3\alpha^2 + 0.7}$$

$$K_5 = \frac{3\alpha^2 \log_e\alpha + 3 - 2.25\alpha^2 - \dfrac{0.75}{\alpha^2}}{\alpha^2 - 1}$$

$$K_6 = \frac{0.9549\alpha^2 \log_e\alpha}{\alpha^2 - 1} - 0.4775$$

$$K_7 = \frac{-3.9 \log_e\alpha + 2.475\alpha^2 - 3 + \dfrac{0.525}{\alpha^2}}{\alpha^2 - 1}$$

$$K_8 = \frac{1.241\alpha^2 \log_e\alpha + 0.3342\alpha^2 - 0.3342}{1.3\alpha^2 + 0.7}$$

$$M_1 = 0.5514\left(\frac{\alpha^2 - 1}{\alpha^2}\right) + \frac{1.614\,(\log_e\alpha)^2}{\alpha^2 - 1}$$

$$M_2 = \frac{-1.037\alpha^2 + 2.770 - \dfrac{2.428}{\alpha^2} + \dfrac{0.695}{\alpha^4} - 5.07\,(\log_e\alpha)^2 + 3.22\log_e\alpha + \dfrac{3.22\log_e\alpha}{\alpha^2}}{\alpha^2 - 1}$$

$$M_3 = \frac{0.529 - 1.348\alpha^2 + \dfrac{0.938}{\alpha^2} - \dfrac{0.119}{\alpha^4} + 3.21\log_e\alpha + 3.55\,(\log_e\alpha)^2 + \dfrac{0.888\log_e\alpha}{\alpha^2}}{1.3\alpha^2 + 0.7}$$

$$M_4 = \frac{0.905\alpha^2 - 3.498 + \dfrac{2.235}{\alpha^2} + \dfrac{0.358}{\alpha^4} + \dfrac{6.35\log_e\alpha}{\alpha^2} - 2.25\log_e\alpha + \dfrac{3.55\,(\log_e\alpha)^2}{\alpha^2}}{1.3\alpha^2 + 0.7}$$

$$M_5 = \frac{-0.512\alpha^2 + 1.195 - \dfrac{0.854}{\alpha^2} + \dfrac{0.171}{\alpha^4} + 2.73\,(\log_e\alpha)^2 - 0.683\log_e\alpha + \dfrac{0.683\log_e\alpha}{\alpha^2}}{\alpha^2 - 1}$$

$$M_6 = \frac{0.217\alpha^2 - 0.434 + \dfrac{0.217}{\alpha^2} - 0.868\,(\log_e\alpha)^2}{\alpha^2 - 1}$$

$$M_7 = \frac{-2.425\alpha^2 + 2.768 - \dfrac{1.037}{\alpha^2} + 0.694\alpha^4 + 3.22\alpha^2\log_e\alpha - 3.22\log_e\alpha - 5.07(\log_e\alpha)^2}{\alpha^2(\alpha^2 - 1)}$$

$$M_8 = \frac{1.129\,(\log_e\alpha)^2 + 1.738\log_e\alpha - 0.7169\alpha^2 + 0.5648 + \dfrac{0.1521}{\alpha^2}}{1.3\alpha^2 + 0.7}$$

PART II APPROXIMATE SOLUTIONS OF THE PLATE PROBLEM

In order to check up the numerical values of the constants K and M of Part I, an approximate solution to the plate problem was developed. The method consisted in assuming a radial strip of the plate as a beam with end conditions and loading as in the actual plate, and on this basis determining the stress and deflection.

Thus for case 3, a radial strip of the plate was considered as a cantilever subjected to a uniformly distributed load. For case 4, the radial strip is assumed a beam supported at one end, uniformly loaded, and acted on by a bending moment at the other of such value as to keep it in equilibrium. For case 5, the elementary radial strip is taken as one-half a beam with ends built in and loaded with a uniform load. The radial strip for case 6 is assumed one-half a beam with ends built in and loaded by a load in the middle, while for case 8, it is assumed a cantilever beam with load in the end. In the other cases the approximate solution is obtained on the basis that a radial section of the plate remains undistorted during bending,[7] deflection consisting entirely of a rotation of the section. It will be noted that these assumptions are accurate only for ratios of α which approach unity. The values of the constants K' and M' corresponding to the constants K and M were determined in this way; they were then compared with those obtained by the exact theory.

As an example of the application of this method, take case 3. In this case we may imagine a cantilever strip cut out from the plate by two radial planes normal to the plate and subtending a small angle $\delta\theta$ (Fig. 4). The total load acting on this elementary cantilever will be

$$L = \frac{\delta\theta\pi p\,(a^2 - b^2)}{2\pi}$$

Assuming this load to act at the middle point of the cantilever, the bending moment will be

$$M = \frac{L\,(a - b)}{2}$$

from which the stress at the built-in end will be:

$$S = \frac{6L\,(a - b)}{2\delta\theta b t^2} = \frac{1.5pa^2\,(\alpha^2 - 1)\,(\alpha - 1)}{t^2\,\alpha^2}$$

or

$$S = K'_3\frac{pa^2}{t^2}$$

where

$$K'_3 = \frac{1.5(\alpha^2 - 1)\,(\alpha - 1)}{\alpha^2}$$

If the curve for K'_3 be plotted as a function of α it will be found to practically coincide at small values of α with the curve for K_3 given in Fig. 2, determined by the more exact flat-plate theory. At larger values of α the discrepancy between the exact and the approximate solution becomes more and more pronounced, which is what we should expect, since the beam action of elementary strips, on which the approximate solution is based, is no more predominant.

In a similar way a value of M'_3 may be found which may be compared with the value of M_3 given by the exact solution (Fig. 3).

Values of K'_1, K'_2, K'_3, etc., and M'_1, M'_2, M'_3, etc., determined in this way for the various cases 1 to 8, are given in Table 1, for various values of $\alpha = a/b$. For comparison, the more exact values of the K's and the M's, K_1 to K_8, and M_1 to M_8 are given.

An examination of this table shows that for values of α near to unity the agreement between the approximate and the exact solutions is very good. This of course would be expected, since as α approaches unity a radial strip of the plate approaches nearer and nearer to the conditions assumed in the approximate solutions. At larger values of α the agreement is surprisingly good in some cases, while in others it is very poor.

[7] For details of this method, see S. Timoshenko's discussion of paper by Waters and Taylor, "The Strength of Pipe Flanges," *Mechanical Engineering*, 1927, p. 1343.

TABLE 1 VALUES OF K, K', M, AND M', FOR CASES 1 TO 8

α =	1	1.25	1.5	2	3	4	5
K_1	0.955	1.10	1.26	1.48	1.88	2.17	2.34
K_1'	0.955	1.07	1.18	1.38	1.74	2.07	2.37
K_2	0	0.66	1.19	2.04	3.34	4.30	5.10
K_2'	0	0.63	1.10	1.81	2.83	3.67	4.38
K_3	0	0.135	0.410	1.04	2.15	2.99	3.69
K_3'	0	0.135	0.416	1.12	2.67	.....	.....
K_4	0	0.122	0.336	0.74	1.21	1.45	1.59
K_4'	0	0.135	0.416	.....	.....	.....	.....
K_5	0	0.090	0.273	0.71	1.54	2.23	2.80
K_5'	0	0.090	0.278	0.75	1.78	2.81	.....
K_6	0	0.115	0.220	0.405	0.703	0.933	1.13
K_6'	0	0.119	0.239	0.477	0.955	.....	.....
K_7'	0	0.592	0.976	1.440	1.880	2.080	2.19
K_7	0	0.587	0.981	1.530	2.290	2.930	3.50
K_8	0	0.227	0.428	0.753	1.205	1.514	1.745
K_8'	0	0.248	0.477	0.955	1.910	2.860	3.820
M_1	0	0.341	0.519	0.672	0.734	0.724	0.704
M_1'	0	0.343	0.524	0.689	0.773	0.775	0.760
M_2	0	0.202	0.491	0.902	1.220	1.300	1.310
M_2'	0	0.194	0.487	0.902	1.260	1.369	1.400
M_3	0	0.00231	0.0183	0.0938	0.2925	0.448	0.564
M_3'	0	0.00218	0.0168	0.0853	0.2683	0.432	0.559
M_4	0	0.00343	0.0313	0.1250	0.2910	0.417	0.492
M_4'	0	0.00364	0.0281	0.1420	0.4470	.....	.....
M_5	0	0.00077	0.00618	0.0329	0.1096	0.1792	0.2338
M_5'	0	0.00073	0.00562	0.0284	0.0894	0.1440	0.1864
M_6	0	0.00129	0.00637	0.0237	0.0619	0.0923	0.114
M_6'	0	0.00129	0.00644	0.0241	0.0644	0.0975	0.124
M_7	0	0.18410	0.41390	0.6640	0.8237	0.8296	0.813
M_7'	0	0.20200	0.42600	0.7210	0.8990	0.9130	0.891
M_8	0	0.00510	0.02490	0.0877	0.2090	0.2930	0.350
M_8'	0	0.00515	0.02570	0.0965	0.2575	0.3890	0.494

The approximate solution thus offers a check on the correctness of the calculations since at values of α near to unity the agreement between the value of a constant K or M and the corresponding value of K' or M' must be good or a mistake in calculation has been made.

PART III THE PRINCIPLE OF SUPERPOSITION OF FUNDAMENTAL CASES, WITH APPLICATIONS

In order to do away with the necessity of solving the fundamental differential equation with different boundary conditions for each different case, each case can be resolved into its component simple loadings, and the formulas for the simple loadings combined to give the desired result. These simple, or fundamental, cases are five in number, and almost any case of flat plates with central holes may be solved by combining two or more of these cases in the manner which will be shown later. These five cases are shown schematically in Fig. 5, as A, B, C, D, and E.

In order to be able to combine these cases, it is necessary to find six quantities for each case. These are:

1 Slope of plate surface at outer edge $(\phi)_a$
2 Slope of plate surface at inner edge $(\phi)_b$
3 Deflection of plate at outer edge $(W)_a$
4 Deflection of plate at inner edge $(W)_b$
5 Radial bending moment at inner edge $(M_1)_b$
6 Tangential bending moment at inner edge $(M_2)_b$

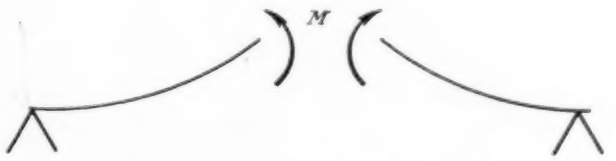
Having these values, the cases which must be superimposed to give the desired result are determined by inspection. Then, if the slope at one edge is zero, the formulas for the slope of the component cases at this edge are added algebraically,[8] equated to zero, and solved for one load in terms of the others. These results are then substituted in the summations of deflections and moments which give the deflection and bending moment for the case being investigated. If one edge is simply supported or free, then the radial bending moment M_1 at this edge must equal zero, and the equation may be derived with this as a start.

There are two methods of applying this principle of superposition. The first consists of combining the complete equation algebraically, and arriving at complete formulas for the desired case. The second method consists of arbitrarily taking certain values of the ratio of inner to outer radius α, finding the numerical value for the coefficients of each equation, and then combining them. Both of these methods are illustrated later.

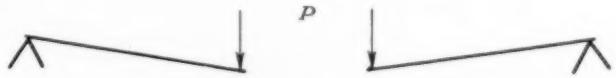
The formulas derived by the method of Appendix 1 for each of the fundamental cases are given, with the numerical values of the coefficients for various values[9] of α, in Tables 2, 3, and 4.



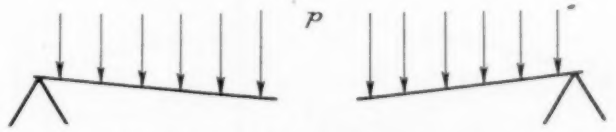
CASE A Loaded by a bending moment of M lb. in. per in. along outer edge; simply supported along inner edge.



CASE B Loaded by a bending moment of M lb. in. per in. along inner edge; simply supported along outer edge.



CASE C Loaded along inner edge; simply supported along outer edge; P = total load on edge.



CASE D Uniformly loaded on surface; simply supported on outer edge; p = uniform load on surface, lb. per sq. in.



CASE E Uniformly loaded on surface; simply supported on inner edge; p = uniform load on surface, lb. per sq. in.

FIG. 5 THE FIVE FUNDAMENTAL CASES

TABLE 2 SLOPE, DEFLECTION, AND STRESS IN PLATES

	Case A	Case B	Case C	Case D	Case E
$(\phi)_a$	$(n_A)_a \frac{Ma}{Et^3}$	$(n_B)_a \frac{Ma}{Et^3}$	$(n_C)_a \frac{Pa}{Et^3}$	$(n_D)_a \frac{pa^3}{Et^3}$	$(n_E)_a \frac{pa^3}{Et^3}$
$(\phi)_b$	$(n_A)_b \frac{Ma}{Et^3}$	$(n_B)_b \frac{Ma}{Et^3}$	$(n_C)_b \frac{Pa}{Et^3}$	$(n_D)_b \frac{pa^3}{Et^3}$	$(n_E)_b \frac{pa^3}{Et^3}$
$(W)_a$	$(m_A)_a \frac{Ma^2}{Et^3}$	0	0	0	$(m_E)_a \frac{pa^4}{Et^3}$
$(W)_b$	0	$(m_B)_b \frac{Ma^2}{Et^3}$	$(m_C)_b \frac{Pa^2}{Et^3}$	$(m_D)_b \frac{pa^4}{Et^3}$	0
$(M_1)_b$	0	M	0	0	0
$(M_2)_b$	$(S_A)_2 M$	$(S_B)_2 M$	$(S_C)_2 P$	$(S_D)_2 pa^2$	$(S_E)_2 pa^2$

With the formulas and figures given in these tables, we are now ready to obtain almost any desired case by superposition. As an example, a case will be taken up which has already been solved analytically, so as to have a check on both methods.

[8] It is understood that the proper sign is taken for the various quantities.

[9] In these tables the sign of the various quantities is disregarded. In combining the cases the proper sign may be determined by inspection.

TABLE 3 VALUES OF COEFFICIENTS FOR TABLE 2

Case A

$$(n_A)_a = \frac{8.4\alpha^2 + 15.6}{\alpha^2 - 1}$$

$$(n_A)_b = \frac{24\,\alpha}{\alpha^2 - 1}$$

$$(m_A)_a = \frac{4.2(\alpha^2 - 1) + 15.6 \log \alpha}{\alpha^2 - 1}$$

$$(m_A)_b = 0$$

$$(S_A)_1 = 0$$

$$(S_A)_2 = \frac{2\alpha^2}{\alpha^2 - 1}$$

Case B

$$(n_B)_a = \frac{24}{\alpha^2 - 1}$$

$$(n_B)_b = \frac{8.4 + 15.6\,\alpha^2}{\alpha(\alpha^2 - 1)}$$

$$(m_B)_a = 0$$

$$(m_B)_b = \frac{4.2\left(1 - \frac{1}{\alpha^2}\right) + 15.6 \log \alpha}{\alpha^2 - 1}$$

$$(S_B)_1 = 1$$

$$(S_B)_2 = \frac{\alpha^2 + 1}{\alpha^2 - 1}$$

Case C

$$(n_C)_a = \frac{2.483 \log \alpha + 0.6685(\alpha^2 - 1)}{\alpha^2 - 1}$$

$$(n_C)_b = \frac{2.483\alpha^2 \log \alpha + 0.6685(\alpha^2 - 1)}{\alpha(\alpha^2 - 1)}$$

$$(m_C)_a = 0$$

$$(m_C)_b = \frac{0.5515\left(\alpha^2 + \frac{1}{\alpha^2}\right) - 1.1029 + 1.614(\log \alpha)}{\alpha^2 - 1}$$

$$(S_C)_1 = 0$$

$$(S_C)_2 = \frac{0.2069\alpha^2 \log \alpha + 0.0557\,\alpha^2 - 0.0557}{\alpha^2 - 1}$$

Case D

$$(n_D)_a = \frac{1.05\alpha^2 + 1.8 - \frac{7.8 \log \alpha + 2.85}{\alpha^2}}{\alpha^2 - 1}$$

$$(n_D)_b = \frac{4.95\alpha^2 - 6 - 7.8 \log \alpha + \frac{1.05}{\alpha^2}}{\alpha(\alpha^2 - 1)}$$

$$(m_D)_a = 0$$

$$(m_E)_a = \frac{-1.0369\alpha^2 + 2.7694 - \frac{2.4281}{\alpha^2} + \frac{0.6956}{\alpha^4} - 5.07(\log \alpha)^2 + 3.217 \log \alpha - \frac{3.217 \log \alpha}{\alpha^2}}{\alpha^2 - 1}$$

$$(m_D)_b = \frac{-2.428\alpha^2 + 2.769 - \frac{1.039}{\alpha^2} + 0.694\alpha^4 + 3.22(\alpha^2 \log \alpha - \log \alpha) - 5.07(\log \alpha)^2}{\alpha^2(\alpha^2 - 1)}$$

$$(S_D)_1 = 0$$

$$(S_D)_2 = \frac{-0.65 \log \alpha + 0.4125\alpha^2 - 0.5 + \frac{0.0875}{\alpha^2}}{\alpha^2 - 1}$$

Case E

$$(n_E)_a = \frac{7.8 \log \alpha + 1.05\alpha^2 - 6 + \frac{4.95}{\alpha^2}}{\alpha^2 - 1}$$

$$(n_E)_b = \frac{7.8\alpha^2 \log \alpha - 2.85\alpha^2 + 1.8 + \frac{1.05}{\alpha^2}}{\alpha(\alpha^2 - 1)}$$

$$(m_E)_b = 0$$

$$(S_E)_1 = 0$$

$$(S_E)_2 = \frac{0.65\alpha^2 \log \alpha - 0.2375\alpha^2 + 0.15 + \frac{0.0875}{\alpha^2}}{\alpha^2 - 1}$$

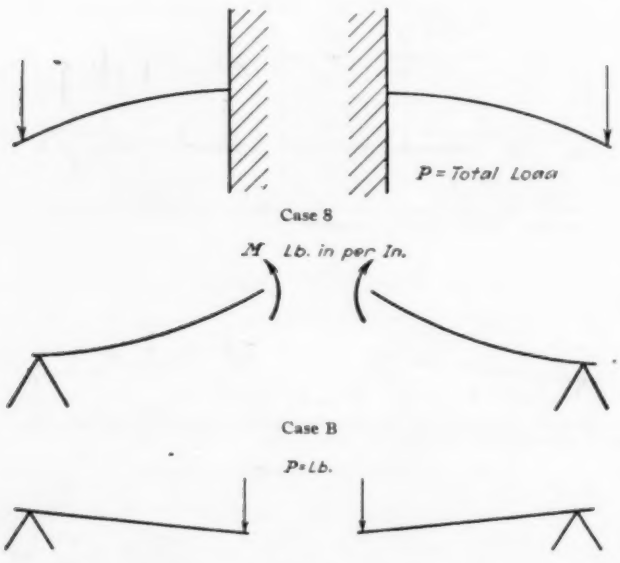


FIG. 6 A CASE OBTAINED BY SUPERPOSITION. CASE 8 OF PART I IS SHOWN TO BE A COMBINATION OF CASES B AND C

Case 8 of Part I, which is the case of a loud-speaker diaphragm, is chosen for this. This is the case of a plate fixed at the inner edge and loaded by a load distributed along the outer edge. A sketch of this case and of the component simple cases is shown in Fig. 6, this case being considered as a combination of cases B and C.

The stress and deflection formulas will be determined by two methods:

Method 1. Since the slope due to the radial bending moment $(M_1)_8$ at the inner edge (case B) is such as to annihilate that due to the applied load P (case C), we have:

$$(\phi_B)_b = (\phi_C)_b$$

TABLE 4 NUMERICAL VALUE OF COEFFICIENTS OF TABLE 2

CASE A

α	$(n_A)_a$	$(n_A)_b$	$(m_A)_a$	$(m_A)_b$	$(S_A)_1$	$(S_A)_2$
1.1	122.6862	125.71	11.280	0	0	11.524
1.25	51.0667	53.33	10.389			5.555
1.5	27.6000	28.80	9.260			3.600
2	16.4000	16.00	7.804			2.667
3	11.4000	9.00	6.342			2.250
4	10.0000	6.40	5.642			2.133
5	9.4000	5.00	5.246			2.083

CASE B

α	$(n_B)_a$	$(n_B)_b$	$(m_B)_a$	$(m_B)_b$	$(S_B)_1$	$(S_B)_2$
1.1	114.286	118.078	0	10.551	1	10.524
1.25	42.666	46.613		8.877		4.556
1.5	19.200	23.200		6.927		2.600
2	8.000	11.800		4.654		1.667
3	3.000	6.200		2.609		1.250
4	1.600	4.300		1.704		1.133
5	1.000	3.320		1.214		1.083

CASE C

α	$(n_C)_a$	$(n_C)_b$	$(m_C)_a$	$(m_C)_b$	$(S_C)_1$	$(S_C)_2$
1.1	1.7953	1.8472	0	0.1660	0	0.1693
1.25	1.6534	1.7659		0.3416		0.1857
1.5	1.4738	1.6537		0.5187		0.2067
2	1.2421	1.4815		0.6721		0.2469
3	1.0094	1.2457		0.7337		0.3114
4	0.8979	1.0850		0.7238		0.3616
5	0.8349	0.9658		0.7035		0.4024

CASE D

α	$(n_D)_a$	$(n_D)_b$	$(m_D)_a$	$(m_D)_b$	$(S_D)_1$	$(S_D)_2$
1.1	0.4797	0.4929	0	0.04419	0	0.04518
1.25	0.8937	0.9470		0.1838		0.09865
1.5	1.1922	1.3022		0.4130		0.1628
2	1.3120	1.4427		0.6819		0.2404
3	1.2477	1.2541		0.8237		0.3135
4	1.1831	1.0409		0.8296		0.3470
5	1.1418	0.8770		0.8132		0.3654

CASE E

α	$(n_E)_a$	$(n_E)_b$	$(m_E)_a$	$(m_E)_b$	$(S_E)_1$	$(S_E)_2$
1.1	0.4992	0.5143	0.04605	0	0	0.04715
1.25	0.9709	1.0502	0.2023			0.1094
1.5	1.3881	1.5840	0.4913			0.1980
2	1.6147	2.0481	0.9016			0.3414
3	1.5711	2.2246	1.2252			0.5561
4	1.4615	2.1546	1.3022			0.7182
5	1.3748	2.0358	1.3084			0.8483

Substituting in this equation the proper values of $(\phi_B)_b$ and $(\phi_C)_b$, as determined from Tables 2 and 3, and solving, the radial bending moment at the inner edge becomes:

$$(M_1)_b = P\left[\frac{0.2069\alpha^2 \log\alpha + 0.0557\alpha^2 - 0.0557}{0.7 + 1.3\alpha^2}\right]$$

Substituting this in the formula for deflection for case B, we get

$$(W_B)_b = \frac{Pa^2}{Et^3}\left[\frac{20.86(\log_e\alpha)(\alpha^2 - 1) + 38.71\alpha^2(\log_e\alpha)^2 + 2.805\left(\alpha^2 - 2 + \frac{1}{\alpha^2}\right)}{(8.4 + 15.6\alpha^2)(\alpha^2 - 1)}\right]$$

The actual deflection in this case is found by subtracting the deflection $(W_B)_b$, due to the bending moment $(M_1)_b$, from that due to the load P, thus:

$$W_b = (W_C)_b - (W_B)_b$$

Substituting and solving for W_b:

$$W_b = \frac{Pa^2}{Et^3}\left[\frac{1.129(\log_e\alpha)^2 + 1.738\log_e\alpha - 0.7169\alpha^2 + 0.5648 + \frac{0.1521}{\alpha^2}}{1.3\alpha^2 + 0.7}\right]$$

This corresponds to the value of M_b given in Part I.

The maximum bending moment in this case is the radial bending moment at the inner edge, which, as found, is:

$$(M_1)_b = P\left[\frac{0.2069\alpha^2 \log_e\alpha + 0.05570\alpha^2 - 0.05570}{1.3\alpha^2 + 0.7}\right]$$

and the maximum stress is:

$$S_b = \frac{6(M_1)_b}{t^2} = \frac{P}{t^2}\left[\frac{1.241\alpha^2 \log_e\alpha + 0.3342\alpha^2 - 0.3342}{1.3\alpha^2 + 0.7}\right]$$

This formula corresponds to the value of K_b as given in Part I.

Method 2. As before, we have:

$$(\phi_B)_b = (\phi_C)_b$$

Using the values given in Table 2 in this equation:

$$+ (n_B)_b \frac{Ma}{Et^3} = (n_C)_b \frac{Pa}{Et^3}$$

Since M in this case is $(M_1)_b$, we have by solving this equation:

$$(M_1)_b = + \frac{(n_C)_b}{(n_B)_b} P$$

Substituting in the formula for the deflection $(W_B)_b$ of case B (Table 2):

$$(W_B)_b = (m_B)_b \frac{(n_C)_b}{(n_B)_b} \frac{Pa^2}{Et^3}$$

As before:

$$W_b = (W_C)_b - (W_B)_b$$

$$W_b = -(M_B)_b \frac{(n_C)_b}{(n_B)} \frac{Pa^2}{Et^3} + (m_C)_b \frac{Pa^2}{Et^3}$$

$$= \frac{Pa^2}{Et^3}\left[-(m_B)_b \frac{(n_C)_b}{(n_B)_b} + (m_C)_b\right]$$

$$= \frac{Pa^2}{Et^3}(M_b) \text{ where } M_b = (m_C)_b - (m_B)_b \frac{(n_C)_b}{(n_B)_b}$$

and

$$S_b = \frac{6(M_1)_b}{t^2} = \frac{6(n_C)_b}{(n_B)_b} \frac{P}{t^2}$$

Then from Table 4 we get the values of the coefficients for various values of α and combine them. For example, for $\alpha = 1.25$, we have:

$$(n_B)_b = 46.613$$
$$(n_C)_b = 1.7659$$
$$(m_B)_b = 8.877$$
$$(m_C)_b = 0.3416$$

then the deflection is:

$$W = \left[-8.877 \times \frac{1.7659}{96.613} + 0.3416\right]\frac{Pa^2}{Et^3} = 0.0051\frac{Pa^2}{Et^3}$$

$$S = 6 \times \frac{1.7659}{46.613} P = 0.227P$$

Repeating for various values of α, Table 5 is obtained.

TABLE 5

	1.1	1.25	1.5	2	3	4	5
K_b.....	0.094	0.227	0.428	0.753	1.205	1.514	1.743
M_b.....	0.00045	0.0051	0.0249	0.0877	0.209	0.293	0.350

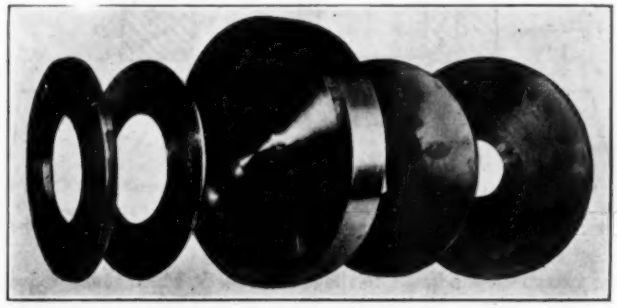


FIG. 7 CIRCULAR PLATES TESTED

By plotting curves from these values of K and M, the coefficients for any intermediate value of α may be obtained by interpolation. It will be noted that these values of K_b and M_b found by this method coincide with those given in Table 1, thus affording a check on the numerical calculations, as well as the method.

In a similar way, other cases of flat plates may be solved. It will be found as a rule that the method of combining the numerical coefficients is the simpler to carry out in practice. The use of this method will in many cases be found to simplify greatly the calculation of flat circular plates with holes and having loading and edge conditions different from those given in Part I.

PART IV EXPERIMENTS AND COMPARISON WITH THEORY

While all formulas derived for each case were checked by means of approximate equations, it was considered desirable to make measurements of stresses and deflections on actual plates for confirmation of the theory. Although most of the cases are very difficult of exact duplication for test purposes, it was believed to be sufficient if one or two of the cases were checked by experiment, since the general method of attack is the same for all cases.

Cases 1 and 6 were chosen as being those whose loading and edge conditions could best be simulated by test. Four plates

were made to represent the conditions of case 1, with different thicknesses and values of ratio of inner to outer radii, and one representing the conditions of case 6. The plates as photographed are shown in Fig. 7, and the dimensions are given in Table 6.

TABLE 6

Plate	a	b	t	α
1	3.873	2.000	0.125	1.91
2	3.813	1.031	0.125	3.70
3	3.813	2.000	0.375	1.91
4	3.813	1.031	0.375	3.70
5	4.000	1.000	0.125	4.00

Method of Test

Plates 1 to 4, inclusive, were tested as shown in the schematic sketch of Fig. 8. Load was applied in a testing machine through

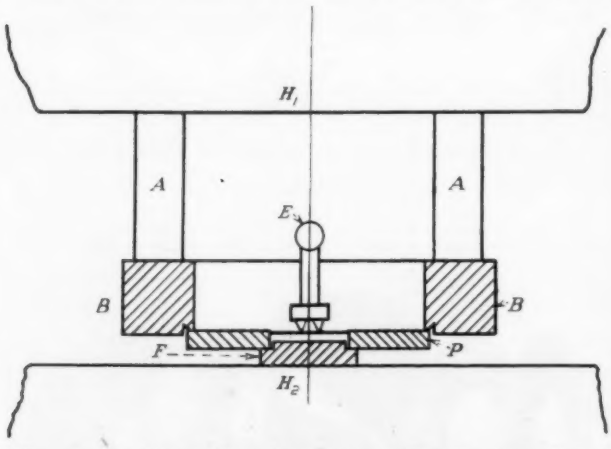


Fig. 8 Diagram of Test Arrangement Representing Case 1 (A, Two Steel Cylinders; B, Heavy Ring; P, Plate; E, Extensometer; F, Supporting Cylinder; H_1, Upper Head of Testing Machine; H_2, Lower Head of Testing Machine.)

the two steel cylinders A to the heavy ring B, which thus applied a load uniformly around the outer circumference. The plate is supported by the cylinder F, resting on the lower head of the machine. Sufficient clearance was allowed between the edges of the plate and the cylinder F and ring B, respectively, to allow a slight lateral movement of the edges. Also the supporting surfaces were slightly beveled, as shown considerably exaggerated, in order that the point of load application be made very definite. The extensometer E was attached along the inner edge where the maximum stress occurs. A photograph of the test as set up in an Amsler testing machine is shown in Fig. 9. The cylinders A, the ring B, and the extensometer E are here clearly shown. The plate is also visible below the ring B. In Fig. 10 the ring is shown resting on the plate with the extensometer in position.

Plate 5, representing case 6, was tested as shown schematically in Fig. 11. The plate is shown in section, being machined integral with two heavy rings; in this way the condition of built-in edges of case 6 was simulated. The position of the extensometer is also shown, being as near as possible to the inner edge.

In all cases strain readings on a 1-cm. gage length were taken on opposite sides of the test plates to be sure of central load application. No trouble, however, was experienced because of non-central application of load. Deflections were measured by dial gages attached so as to record the relative travel of the heads of the machine.

Results of Tests

The results of the tests on plates 1 to 4, tested as in Fig. 9, are shown on the curves of Figs. 12 to 19, inclusive. Those on plate 5 representing case 6 are shown on the curves of Figs. 20 and 21. In all cases the stresses were determined from the measured strain, the average value taken on diametrically opposite positions being used. This was done to eliminate the effect

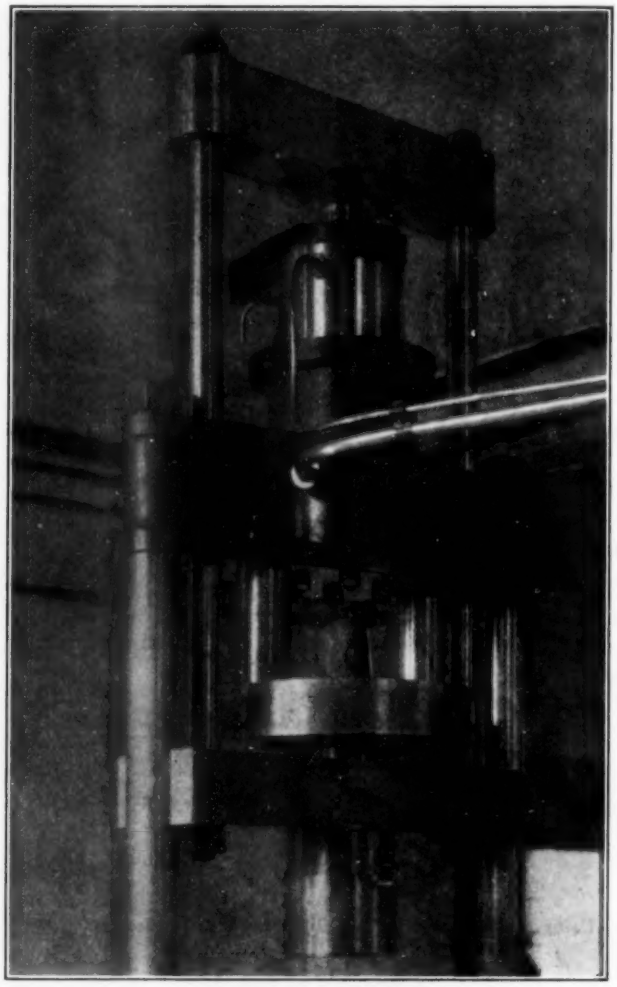


Fig. 9 Photograph of Test Arrangement Representing Case 1

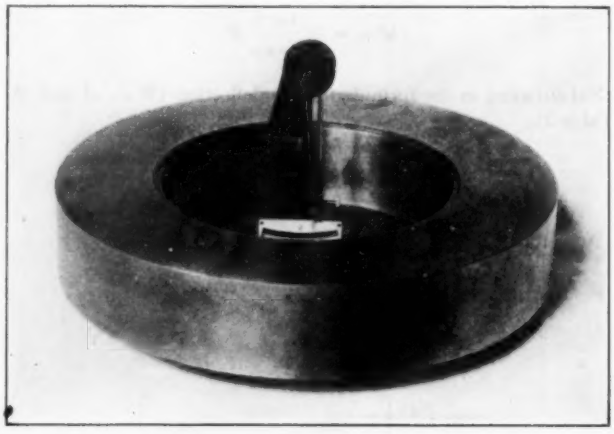


Fig. 10 Extensometer and Ring in Position on Plate

of any slight eccentricity of load. A modulus of elasticity $E = 30 \times 10^6$ lb. per sq. in. for steel was taken. In case two stresses were acting at right angles the stress was determined from the well-known formulas for two-dimensional stress conditions:[10]

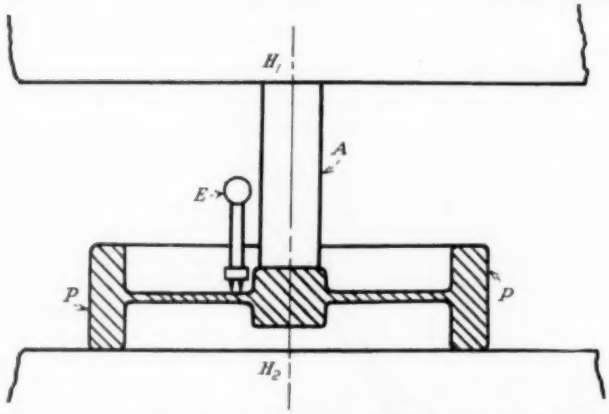


FIG. 11 DIAGRAM OF TEST ARRANGEMENT REPRESENTING CASE 6 (*A*, Cylinder; *P*, Plate; *E*, Extensometer; H_1, Upper Head, and H_2, Lower Head of Testing Machine.)

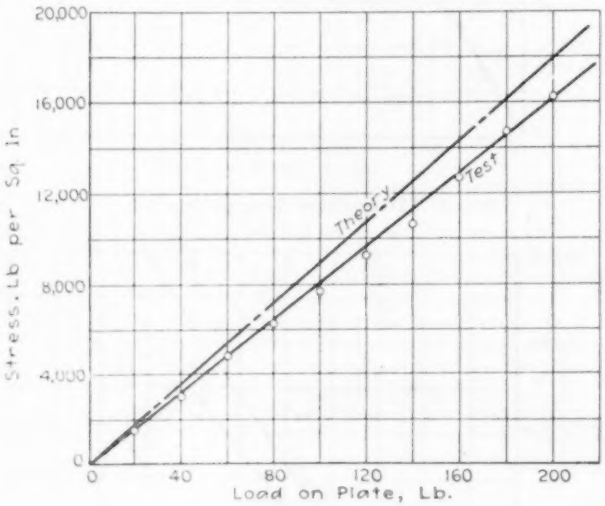


FIG. 12 STRESS TEST ON PLATE 1

$$S_r = E\left[\frac{e_r + \frac{1}{m}e_t}{1 - \frac{1}{m^2}}\right]$$

$$S_t = E\left[\frac{e_t + \frac{1}{m}e_r}{1 - \frac{1}{m^2}}\right]$$

where

E = modulus of elasticity (assumed 30×10^6 lb. per sq. in.)
e_r = measured radial strain, inches per inch
e_t = measured tangential strain, inches per inch
S_t = tangential stress, lb. per sq. in.
S_r = radial stress, lb. per sq. in.
$\frac{1}{m}$ = Poisson's ratio.

[10] See, for example, "Applied Elasticity," Timoshenko and Lessells, p. 12.

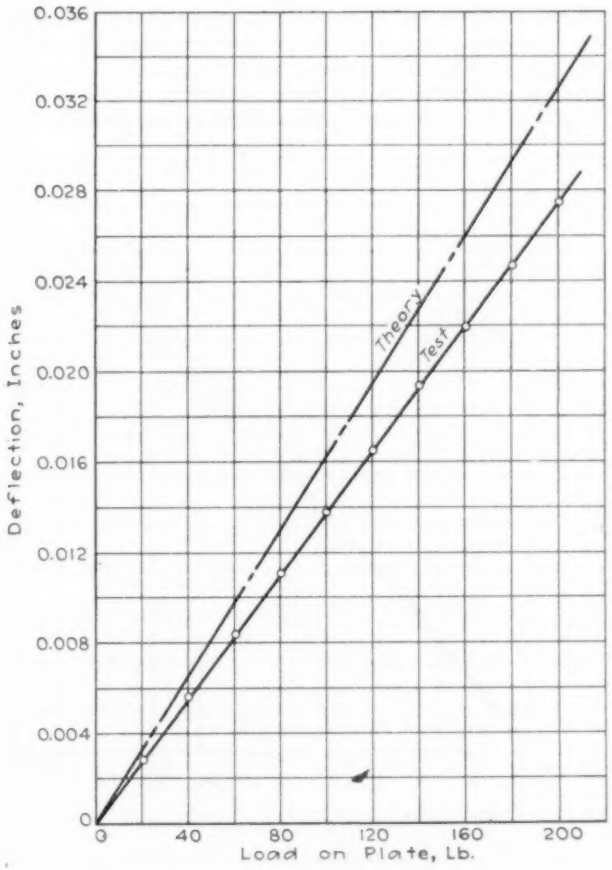


FIG. 13 DEFLECTION TEST ON PLATE 1

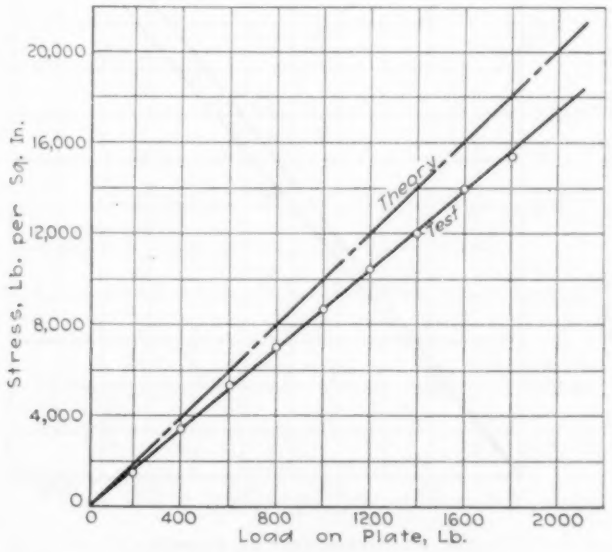


FIG. 14 STRESS TEST ON PLATE 2

For plates 1 to 4, the stress and deflection were computed using the stress and deflection formulas of case 1 (Figs. 2 and 3), a slight correction being necessary due to the fact that the point of application of the load was not exactly along the circumference, but displaced slightly inward. It may be seen that for these plates the agreement between test and theory was fairly good,

an error of 15 per cent being the maximum. In addition, the stress found for each test was below that calculated, thus indicating that the formulas are on the safe side.

For the case of plate 5 (case 6) the stress was measured about $^3/_8$ in. from the inner edge and the general formulas for

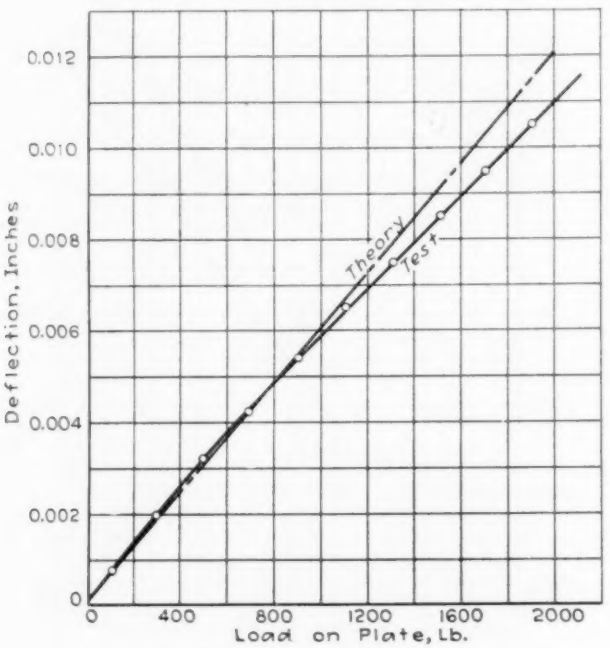


FIG. 15 DEFLECTION TEST ON PLATE 2

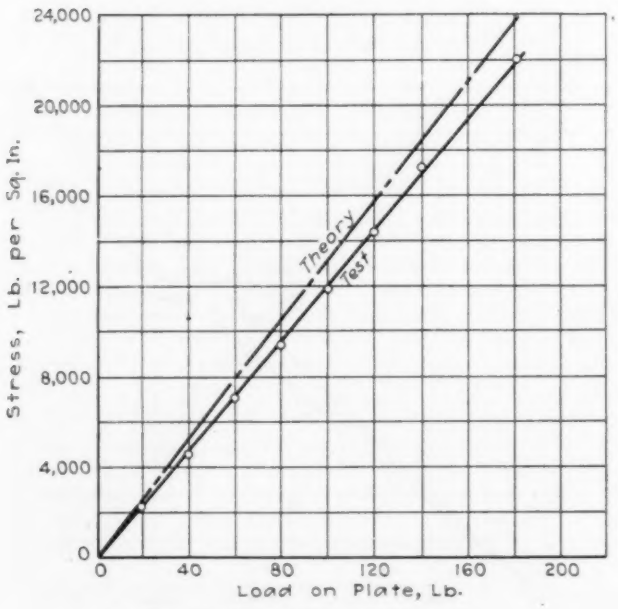


FIG. 16 STRESS TEST ON PLATE 3

slope and deflection along a radius used to obtain the theoretical stress curve (Fig. 21). It may be seen that the stress found by test (shown in full lines) agrees very well with that found theoretically. In the case of deflections, however, the actual deflection, as shown on the curve of Fig. 21, was much greater than that calculated using the formulas of case 6. It is reasonable to assume that the portion *AB* of the test curve (Fig. 21) was due to taking up slack in the apparatus, so that the true load deflection curve has the same slope as the portion *BC* of the test curve. On this basis we have a corrected test curve as shown. Even with this correction the test curve is considerably above the theoretical curve (shown with the dot-and-dash line). A

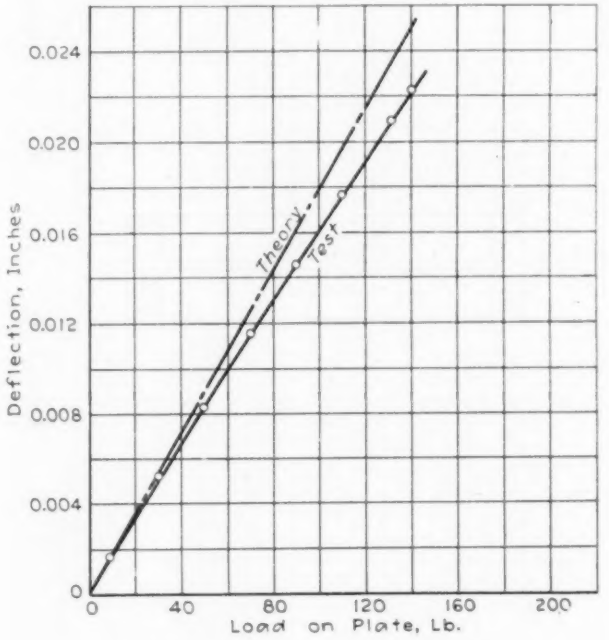


FIG. 17 DEFLECTION TEST ON PLATE 3

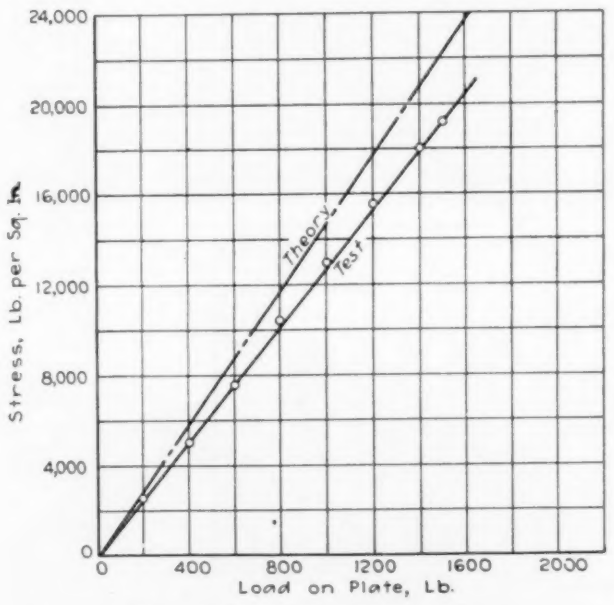


FIG. 18 STRESS TEST ON PLATE 4

correction to the theoretical curve was made by taking into account the bending of the rim *A* (Fig. 11), but this correction was only a few per cent and is therefore not shown. It may therefore be concluded that the fixity of the edges in this case was far from perfect (as assumed in the derivation of the formulas), and hence the true deflection is considerably greater than the theoretical. The matter of fixity of the edges is not so im-

portant for stress, and hence the theoretical and test stress curves agree quite well. In this respect the plate is similar to a cantilever beam fixed at one end and loaded at the other. If the fixity of the one end is imperfect, the deflection will be considerably greater, although the stress will be about the same as that given by the cantilever formula.

Taking the tests as a whole, it appears that a fairly good check between theory and test is obtained where the edges of the plate are simply supported; but where the edges are built in, the theoretical results, while approximately correct for stress, give values much too low for deflection. In all cases the calculated stresses were on the safe side. The refinement of these tests was also not great; more refined tests would probably show a somewhat better agreement with theory than was the case with these tests, which were made mainly to show that the formulas for certain cases were sufficiently accurate for practical engineering work.

Conclusions

The tests therefore indicate that the simplified formulas given in this paper are sufficiently correct for practical purposes

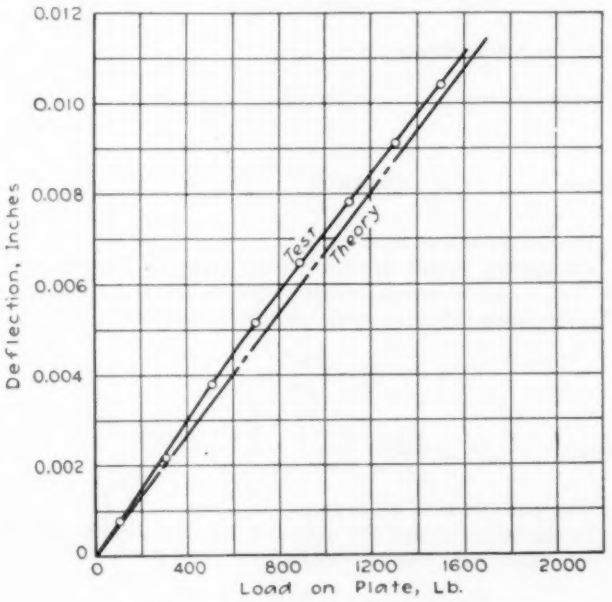


Fig. 19 Deflection Test on Plate 4

for the cases where the edge conditions are determinate, such as free or simply supported edges. For the case of so-called built-in edges the tests indicate that, while the formulas are probably satisfactory for computing stress, the actual deflection in such cases may be much larger than that calculated, because of imperfect fixity of the edges.

These simplified formulas take no account of shear or of the stretching of the middle surface of the plate. Hence, it is believed that they will be reasonably accurate for practical work for a plate with simply supported or free edges only where the thickness is less than about one-third the difference in diameters, and deflections are less than one-half the thickness. Where one or more edges are built-in, the tentative limit is a thickness of one-sixth the difference in diameters. It should be noted, as the tests show, that the formulas for built-in edges are somewhat inaccurate anyway because of imperfect fixity of the edges.

The limits of thickness equal to one-third and one-sixth the difference in diameters are, as mentioned, merely tentative; an idea of the error involved because of neglecting shear may be obtained by using the formulas for deflection due to shear.

In practical work where the fixity of the edges is indeterminate, the safest procedure would be to assume the worst edge conditions and calculate on this basis. If the worst conditions are

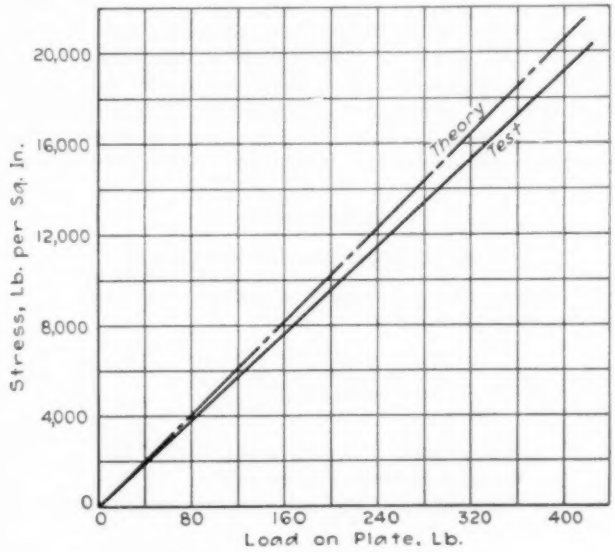


Fig. 20 Stress Test on Plate 5

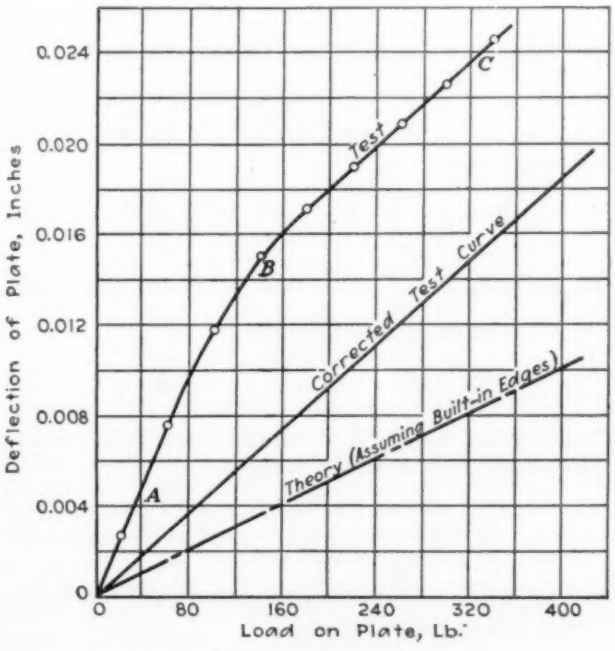


Fig. 21 Deflection Test on Plate 5

not known, the stresses may be computed by assuming various edge conditions, and the maximum stress thus obtained taken as a basis.

It should be noted that this paper is mainly confined to presenting the results of elastic-flat-plate theory, as applied to circular plates, in a simple and usable form for the designer. Therefore, no account was taken of the effect of overstraining the plate material or the effect of plastic flow, which may make possible a higher elastic limit for the plate and therefore a higher

working stress. A complete treatment of the plate problem should of course include such questions, as well as the question of working stress, but these are problems beyond the scope of this paper.

ACKNOWLEDGMENTS

The authors wish to express their indebtedness to Dr. S. Timoshenko, professor of Engineering Mechanics, University of Michigan, and to Dr. A. Nádai, of the Research Department of the Westinghouse Company, for many valuable suggestions made during the preparation of this paper. They are also indebted to Messrs. G. Runke and W. Cory for checking the numerical work involved in the formulas for the various cases.

Appendix 1

DERIVATION OF FORMULAS FOR CASE 1

The differential equation for the slope ϕ of a circular plate symmetrically loaded is:[11]

$$\frac{d^2\phi}{dx^2} + \frac{1}{x}\frac{d\phi}{dx} - \frac{\phi}{x^2} = -\frac{Q}{D} \quad \ldots\ldots\ldots\ldots [1]$$

In this

ϕ = slope at radius x from axis of plate
x = radius at which slope is considered
Q = shearing force per unit circumferential length at radius x
D = flexural rigidity of the plate $= \dfrac{Et^3}{12\left(1 - \dfrac{1}{m^2}\right)}$

For case 1 where a load P is uniformly distributed along the edges, $Q = \dfrac{P}{2\pi x}$ (dividing the total shearing load by the total circumferential length). Using this value in Equation [1] and integrating, we get

$$\phi = -\frac{Px}{8\pi D}[2\log_e x - 1] + \frac{C_1 x}{2} + \frac{C_2}{x} \quad \ldots\ldots [2]$$

Since the slope $\phi = \dfrac{dW}{dx}$, where W is the deflection at the radius x, we may integrate [2] to get W, thus:

$$W = \frac{Px^2}{8\pi D}(\log_e x - 1) - \frac{C_1 x^2}{4} + C_2 \log_e x + C_3 \ldots [2a]$$

The following boundary conditions for case 1 enable the determination of the integration constants C_1, C_2, C_3:

$$\left.\begin{array}{l} \text{When } x = a,\ (M_1)_{x=a} = 0 \\ \text{When } x = b,\ (M_1)_{x=b} = 0 \\ \text{When } x = a,\ W = 0 \end{array}\right\} \quad \ldots\ldots\ldots [3]$$

In this M_1 is the radial bending moment per unit length. From the well-known plate theory[12]

$$M_1 = D\left(\frac{d\phi}{dx} + \frac{1}{m}\frac{\phi}{x}\right) \quad \ldots\ldots\ldots\ldots [4]$$

Therefore, at $x = a$, $\left(\dfrac{d\phi}{dx} + \dfrac{1}{m}\dfrac{\phi}{x}\right)_{x=a} = 0$

at $x = b$, $\left(\dfrac{d\phi}{dx} + \dfrac{1}{m}\dfrac{\phi}{x}\right)_{x=b} = 0$

Using these conditions, we get:

$$\left.\begin{array}{l} C_1 = \dfrac{P}{5.2\pi D}\left\{\dfrac{2.6\,(a^2\log_e a - b^2\log_e b)}{a^2 - b^2} + 0.7\right\} \\ C_2 = \dfrac{2.6P}{5.6\pi D}\left[\dfrac{a^2b^2}{a^2 - b^2}\log_e \alpha\right] \\ C_3 = -\dfrac{Pa^2}{8\pi D}(\log_e a - 1) + \dfrac{C_1 a^2}{4} + C_2 \log_e a \end{array}\right\} \quad \ldots\ldots [5]$$

The tangential bending moment is (from the well-known theory):

$$M_2 = D\left(\frac{\phi}{x} + \frac{1}{m}\frac{d\phi}{dx}\right) \quad \ldots\ldots\ldots\ldots [6]$$

Putting the constants given by Equation [5] in [2], differentiating, and substituting in [6], we find a value of M_2 in terms of x. The maximum value of the bending moment M_2 occurs when $x = b$ and is:

$$(M_2)_{x=b} = \frac{0.7P}{4\pi}\left[1 + \frac{3.714\,\alpha^2\log_e \alpha}{\alpha^2 - 1}\right]$$

The maximum stress will be

$$S = \frac{6[M_2]_{x=b}}{t^2} = \frac{P}{t^2}K_1$$

where $$K_1 = 0.3343 + \frac{1.242\alpha^2\log_e \alpha}{\alpha^2 - 1}$$

This is the value of K_1, as given.

Putting the values of the constants given by Equation [5] in [2a], we get a formula for the deflection W at any radius x. The maximum deflection occurs when $x = b$ and is:

$$y = \frac{Pa^2}{Et^3}M_1$$

where $$M_1 = 0.5514\frac{\alpha^2 - 1}{\alpha^2} + \frac{1.614\,(\log_e \alpha)^2}{\alpha^2 - 1}$$

This is the value of M_1, as given.

In this same manner the values of the constants K and M for the other cases may be determined.

Appendix 2

DEFLECTION OF PLATE DUE TO SHEAR

As an example of the method of taking into account the deflection due to direct shear, consider a plate with load concentrated along the edges (cases 1, 6, and 8). We may imagine an elementary radial strip cut out of the plate, as was done in the approximate solution. This strip, bounded by two radial planes passing through the axis of the plate and subtending an angle $\delta\theta$, will be subjected to shearing forces of magnitude:

$$Q = \frac{P\delta\theta}{2\pi}$$

We now consider a small element of this strip at a distance x from the center of the plate and having a length dx. Since the cross-section of this strip is rectangular and of area $A = \delta\theta x t$, the deflection due to shear of this element will be:[13]

[11] See "Applied Elasticity," p. 276.
[12] See "Applied Elasticity," p. 274.
[13] This formula is the ordinary one for shearing deflection in a bar of rectangular cross-section. See "Applied Elasticity," p. 94.

$$dy_s = \frac{1.5Qdx}{AG} = \frac{1.5Pdx}{2\pi xtG},$$

G being the modulus of rigidity. The total deflection of this elementary strip will then be:

$$y_s = \int_b^a dy_s = \frac{0.239P \log_e \alpha}{tG}$$

which is the formula given for cases 1, 6, and 8.

Other formulas for shearing deflection are derived similarly.

Discussion

E. O. Waters.[14] The critically minded reader, after glancing over this paper, might raise the objection that the material here presented is not original. The eight cases illustrated in Fig. 1, for example, have all been discussed, and appropriate formulas derived, in a series of articles by Max Ensslin in *Dingler's Polytechnisches Journal*, vol. 319 (1904), in which several other cases are considered which are not mentioned in the present paper. By way of rebuttal to any such objection, the writer feels that the authors deserve great commendation for their method of approach to the problem and the system used for presenting the stress and deflection formulas. Any such arrangement is a boon to the designer, and places the calculation of flat rings on a par with that of beams, shafts, and other common machine elements.

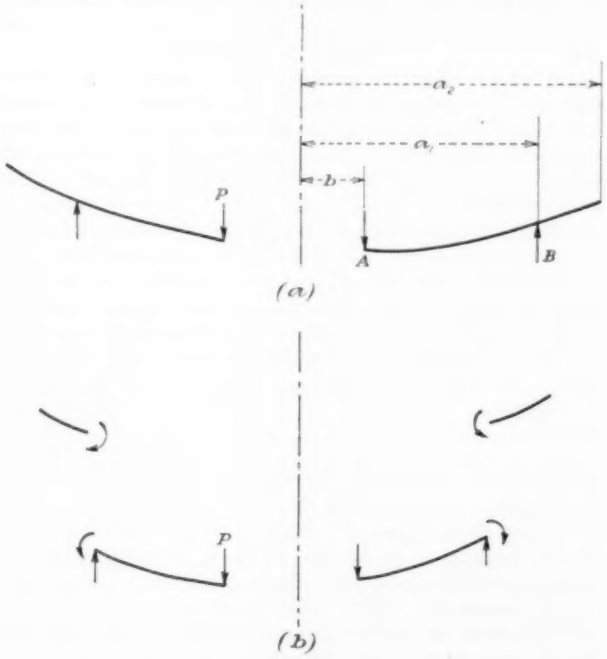


Fig. 22

The experimental work shows that the formulas are sound, at least as far as simply supported rings are concerned. This checks with the writer's experience with pipe flanges and proves that the precise formulas should be used, rather than the approximate ones, except for values of α close to unity. With regard to the built-in rings, it is apparent that there was imperfect fixity at the edges, as stated by the authors. Furthermore, even though the extensometer readings showed stresses corresponding closely with the theoretical values, it should be remembered that the readings were taken at an appreciable distance away from the edge, and there may well have been much higher stresses at the fillets. It is impossible to go into this quantitatively without a knowledge of the fillet radius. The writer believes that it would be worth while to make further tests of some of the built-in cases, with various different fillet radii, and derive if possible some relation between the size of fillet, thickness of plate, and degree of fixity as shown by a comparison of computed and measured deflections.

The authors state that an attempt was made to reconcile the theoretical and test deflections on plate 5 by taking into account the bending of the rim, but without success. The writer would

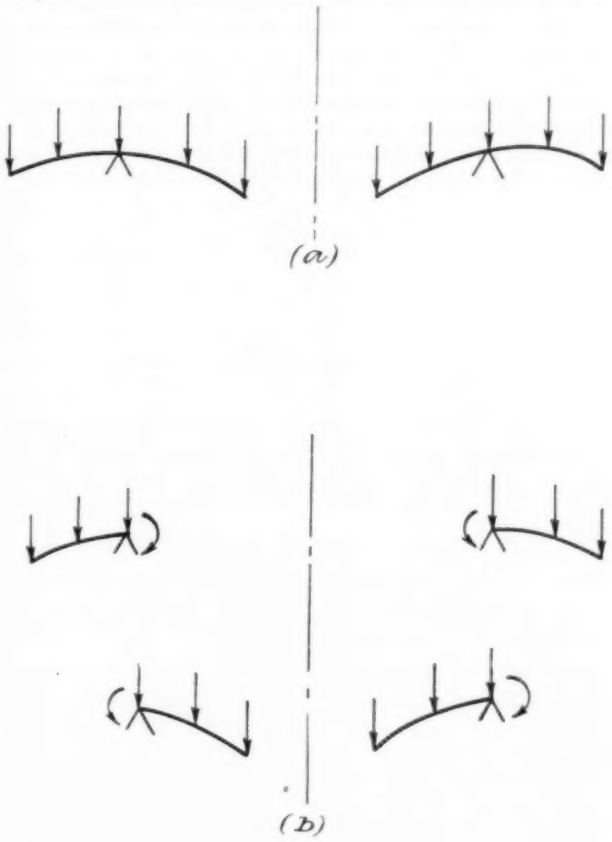


Fig. 23

suggest that, in this case, the outer radius of the plate be taken at the center of the rim; this will increase α and a to 4.47 (scaling dimensions from Fig. 11) and change the theoretical deflection from 0.0025 in. per 100 lb., as per Fig. 21, to 0.0035 in. per 100 lb. This will account for half the discrepancy between the theoretical and the test curves, and bending of the rim will remove part of the remaining discrepancy. In his analyses of pipe flanges with hubs, the writer made this assumption and obtained a very good agreement between the calculated and the measured deflections. It is of course an approximation, and would be quite inaccurate if there were much difference between the thickness of the plate and that of the rim.

The method of superposition, treated in Part III, opens up an unlimited variety of possible combinations of loading. In fact, the five fundamental cases (A to E) are really of much more importance than the eight cases considered in the first part of the paper, and the writer would recommend that the coefficients of Table 2 be charted against α, in a series of curves corresponding to those in Figs. 2 and 3, so that problems involving combinations of the fundamental cases may be solved rapidly and accurately with a minimum of slide-rule manipulations.

[14] Associate Professor of Mechanical Engineering, Yale University, New Haven, Conn. Assoc-Mem. A.S.M.E.

The superposition principle is based upon the theorem that stress and deflection due to several loads applied simultaneously are equal to the sum of the stresses and deflections due to these loads when applied separately. A somewhat similar theorem, which may be called the principle of juxtaposition, states that the stress and deflection in a member are equal to the stress and deflection of a group of connected members, provided that the connected members are geometrically identical with the single member and that the moments, slopes, and deflections are mutually equal at the points of connection. By means of this principle innumerable other combinations may be built up from the five fundamental cases. Take the loading in Fig. 22*b*, herewith, as an example. This arrangement is the same as a ring loaded along the inner edge, simply supported along the outer edge, and loaded by a bending moment along the outer edge (cases A and C) fastened to another ring loaded by a bending moment along the inner edge (case B), the slopes, moments, and deflection sat point B being equal. (See Fig. 22*b*.) From Table 2:

$$(nC)_a \frac{Pa_1}{Et^2} - (nA)_a \frac{Ma_1}{Et^2} = (nB)_b \frac{Ma_2}{Et^3}$$

$$(M_2)_b = (S_c)_2 P - (S_A)_2 M$$

Furthermore,

$$(S_2)_b = \frac{6(M_2)_b}{t^2}$$

Obviously, M and $(M_2)_b$ may be eliminated from the foregoing, and a solution obtained for $(S_2)_b$, the tangential stress at point A, Fig. 22*a*, in terms of P, a_1, a_2, b, and t. In like manner, the radial stress and the deflection at the inner edge may be computed.

Another possibility is shown in Fig. 23, *a* and *b*. In this instance, it is seen that cases D and E are involved, together with two fundamental cases (load consisting of a bending moment at the inner or outer supporting edge) similar to cases A and B.

A. NADAI.[15] The paper contains a valuable application of the bending theory of flat plates. As the authors have mentioned, the bending theory of flat circular plates (with or without a concentric circular hole) was established a long time ago and can be considered in its main parts as perfected. It may be found in old and new textbooks on strength of materials or on applied elasticity. The main reason why engineers who have to calculate the dimensions of plates or similar elements of construction which require a two-dimensional analysis do not apply the results of the theory of elasticity seems to be that the results of this theory have often not been presented in a manner to enable an engineer to apply the results directly to a practical case and without an extensive additional analysis. It seems therefore to the writer that besides the value of the bending theory of flat plates which consists in establishing more or less general expressions for the functions expressing the stresses or the deflections of flat plates, a special value of the theory is to give to the designer the possibility to apply directly the results of a more exact analysis to special applications. It should be emphasized that this is often possible, and in many cases of practical importance, if the engineer will make a study of how to apply the general results of the theory to special cases. Many results of the theory for circular or rectangular plates can be expressed in formulas of such a form as:

$$\text{Maximum bending stress of plate } s = \alpha \cdot \frac{pa^2}{t^2} \quad \text{or} \quad s = \beta \cdot \frac{P}{t^2}$$

In this formula, p or P indicates the load acting on the plate (in case of a uniform pressure, p; in the case of a single force, P); a, a principal dimension of the plate; t, the thickness. All that the theory should furnish to the designer is, in many cases, only the numerical value of the parameter α or β. It is therefore very valuable that the authors have endeavored to calculate the values of these parameters for some of the most frequently occurring cases of bending of circular plates and that they have tabulated and plotted them in the form of curves showing how they depend, in the case of a circular plate with a central hole, for example, on the ratio a/b of the outer to the inner radius of the plate.

It would be valuable and useful if engineers would apply more the theory of bending of plates and of two-dimensional systems of stresses from this point of view. It will help to find the numerical values of such parameters as α or β, which are of a practical importance when stresses in plates are to be calculated, and will allow other engineers to use the results without elaborate, numerical, or analytical work.

Some time ago efforts in other directions had been made which may be mentioned perhaps in connection with these remarks. In th se efforts it was tried to replace the more complicated analysis of two-dimensional stress problems by solving the same cases approximately as one-dimensional problems. For example, efforts have been made to apply to the case of a circular plate loaded in the center a similar method as used in the theory of one-dimensional bending of a straight bar. It is not thought today that this was the right way to find more exact expressions for the stresses. It is felt that by such a proceeding the beautiful symmetry in many of the given problems would be destroyed. Also, in this way, the character of a singularity in the functions, which possibly has to be considered in a problem, would remain concealed. Compare, for example, the two cases of the bending of a straight bar and of a circular plate—the one simply supported at its ends and loaded at the middle, and the other simply supported on its circumference and loaded by a single force at its center. There is a definite difference between the two cases. In the first, the maximum stress always remains a finite quantity and is given by the ratio of the maximum bending moment at the middle point of the bar to the section modulus. On the other hand, in a circular plate loaded in the middle point by a single force P, the bending stress at the center of the plate becomes, as is well known, infinite. It would be hopeless to try to replace the part in the solution containing the singularity of the deflected middle surface

$$w = cx^2 lnx$$

by some other analytical expression derived from an approximate method to calculate bending stresses. Although this part of the complete solution (see Equation [2a]) contributes a term to it which gives in a solid disk infinite stress at the center, it is of the greatest practical importance to retain the corresponding term in the solution, as it would otherwise not be possible to establish a formula for the greatest stress in the case that the single force P is distributed over a finite area; for example, over a small circle. A further knowledge and discussion of such special solutions as the aforementioned seem to be very valuable to engineers who have to calculate stresses in plates with loads concentrated in small parts of the surface of the plate.

Regarding the results of the bending tests of the authors with plates with a circular hole, other experimenters and the writer have found that the deflections of circular plates loaded in their center by a single force and simply supported on their circumference are always found to be greater than the theory would predict. Contrary to this, the authors found the calculated deflections (with the exception of one case) always greater than the observed deflections. This discrepancy seems to be a consequence of the fact that the authors have used comparatively rigid

[15] Mechanics Section, Research Department, Westinghouse Electric & Manufacturing Co., East Pittsburgh, Pa.

plates (with a great ratio a/b) and that they have loaded these plates along two concentric circles lying on the surface of the plate. Through this method of supporting the plate additional moments are introduced through the friction which acts in radial direction when the plate is loaded in axial direction and bends out of its plane. It can easily be seen that these radial bending moments tend to diminish the deflection of the plate. Correspondingly, the authors found smaller deflections than predicted for the same case by the bending theory of plates.

The authors' work has contributed a valuable extension to the well-known theory of the bending of circular plates, and it should therefore be appreciated by engineers that they have worked out the results of the theory in a way which allows the application of the results of it directly to the practical cases of design.

S. Timoshenko.[16] The theory of bending of symmetrically loaded circular plates was developed a hundred years ago by Poisson,[17] but it has not found a broad application in engineering because the final results have not been given in a simple form for numerical calculations. The authors have done a very useful work in presenting the formulas for deflection and for maximum stress in an extremely simple form and in giving graphs for the numerical factors in the formulas. The formulas were checked by making derivations in two different ways, thus excluding the possibility of errors.

Regarding the experiments described in the paper, the writer would like to state that it is always difficult to realize simply supported edges which are free to move in the plane of the plate. Considering this, the results obtained by the authors for this type of support seem satisfactory. The case of fixed edges seems to be simpler to realize. Although it is impossible to have an absolutely fixed edge, the writer thinks that in the case represented in Fig. 11 the condition of a fixed edge is realized with sufficient accuracy. This makes it difficult to explain the large discrepancy between theory and experiment shown by Fig. 21. The small deformations spreading into the heavy boundary ring cannot be responsible for such a large discrepancy, and the writer considers it desirable to repeat the experiment in such a way that the deflections of the plate are measured directly and the deformation in the testing machine is completely eliminated. This can easily be done with the instruments mentioned in the paper.

Authors' Closure

Regarding the remarks made by Drs. Timoshenko and Nadai relative to the difficulty of loading plates with simply supported edges, so that the edges are free to move in the plane of the plate, the authors agree with this. It is very probable that the small discrepancies obtained between test and theory, in the case of plates with simply supported edges, may be largely explained if the friction between the edges is taken into account. Since it was beyond the scope of this investigation to carry out elaborate tests where this friction was eliminated, the authors made the simpler tests, which show that the plate formulas for these cases are sufficiently accurate for most practical work.

Regarding the tests on plate 5, having built-in edges, the authors have repeated the tests on this plate as suggested by Dr. Timoshenko, using a dial gage mounted on a frame in such a way that only the deflections of the plate itself were measured. In this way the deformations of the testing machine itself were eliminated. The load-deflection curve determined in this way came about 10 per cent below the curve determined by measuring the travel of the head of the testing machine.

[16] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[17] Paris, Mém. de l'Acad., vol. 8, 1929.

The curve marked "corrected test curve," as shown in Fig. 21, gives a deflection of 0.0148 in. at 320-lb. load. Repeating the test by measuring the head travel of the machine after the irregularities had been smoothed out, the deflection was 0.0141 in. at 320-lb. load. The test was then repeated, with the dial gage mounted on a frame in such a way as to eliminate the distortion of the testing machine, and this gave a deflection of 0.0127 in. at 320-lb. load, or about 10 per cent less than that found before.

This indicates that there was some distortion in the testing machine amounting to about 10 per cent, so that the true deflections in this case were about 10 per cent less than those found by measuring the head travel. However, this 10 per cent error in measuring the deflection does not appear to invalidate the conclusions drawn in the paper regarding the imperfect fixity of the edges, since the theoretical deflection at 320-lb. load, assuming perfectly built-in edges, was only 0.008 in. Comparing this with the accurate test value of 0.0127 in., as previously found, it is seen that the true deflection is around 60 per cent greater than the calculated, thus indicating imperfect fixity of the edges.

It is possible that a partial explanation of this lack of fixity is the fact that the fillets for plate 5, Fig. 11, were made of very small radii, so that they were practically sharp corners. The authors had another plate made having a large fillet and with the same ratio of outer to inner radius. This plate gave a deflection only about 15 per cent greater than the theoretical deflection, calculated from the formulas for case 6. This tends to indicate that the size of fillet does have a considerable effect on the fixity of the edge, as suggested also by Professor Waters in the discussion.

In order to show that the flange on plate 5 was sufficiently large, another plate was made up identical with plate 5, but with a much larger flange, and with the same size fillets. This plate when tested gave results practically the same as those obtained on plate 5. This therefore indicates that the flange on plate 5 was sufficiently large, and hence the discrepancy between test and theory may not be accounted for on the basis of distortion of the flange.

The whole subject of the effect of ratio of outer to inner diameter, fillet radii, thickness, etc., on the fixity of the edges is quite involved, and, as Professor Waters remarks, this subject could properly form the basis of another investigation. It would certainly be desirable to develop a method whereby allowance for the imperfect fixity of the edges could be carried out. One such method might be to assume the plate built-in at some point inside the rim, say at a distance from the edge equal to some constant times the thickness. Such an empirical method, as well as the method suggested by Professor Waters for plate 5, would, however, involve a large amount of experimental work before a sufficiently comprehensive solution could be found. This, however, is beyond the scope of this paper.

In regard to Professor Waters' statement relative to the articles by Ensslin, it should be noted that the tests carried out by Ensslin were either on plates without holes or on plates having holes with diameters less than one-tenth the diameter of the plate. For such cases, the holes were so small that, in the authors' opinion, for practical purposes these might be considered almost as plates without holes. Also, no tests were reported by Ensslin on plates with built-in edges. Furthermore, Ensslin was unable to make any strain measurements on his plates, because of the lack of appropriate instruments, and hence the formulas for stress in the plates could not be verified by direct strain measurements. This, however, was possible in the present paper.

APM-52-4

The Radially Tapered Disk Spring

By W. A. BRECHT[1] and A. M. WAHL,[2] EAST PITTSBURGH, PA.

In this paper disk springs, both flat and dished, having radially tapered cross-sections are considered. The advantages of such springs are discussed. Equations are developed by approximate and more exact methods for calculating the strength and flexibility of such springs. The theoretical work is checked up by numerous tests, including both strain and deflection measurements. It is concluded that springs of this type may be of advantage in certain applications.

IN ENGINEERING problems the need is frequently felt for a "frictionless" spring where space or other limitations do not permit of the use of springs of conventional design such as the helical spring. For one such application, flat or dished circular disks of spring steel, of uniform thickness, having concentric circular holes, and loaded uniformly at the inner and outer edges, were considered. Such disks could be used singly, or in stacks with suitable spacers to secure the desired flexibility. In computing deflections and stresses in these disks an approximate method suggested by Dr. S. Timoshenko was first used. In this method the deformation is considered to consist purely of a rotation of the section, cross-bending being neglected. This was found to give results practically identical with those given by the elastic-flat-plate theory.[3] Referring to Fig. 1,

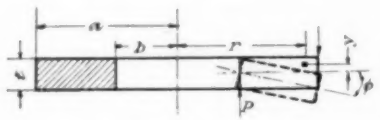


FIG. 1 PLATE OF CONSTANT THICKNESS

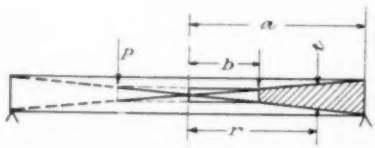


FIG. 2 DISK SPRINGS OF VARIABLE THICKNESS
($t = 2kr$, where k = thickness constant.)

P' = force per unit length of inner circle

ϕ = angle of rotation of the disk section under load.

The radial displacement of any element will equal $y\phi$, and the unit elongation, $y\phi/r$, from which

$$\text{Stress} = E\frac{y\phi}{r} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

The stress in each element of the disk will be a tangential tensile or compressive stress, depending on whether the element moves radially inward or outward when the section is rotated under load. From [1] it is seen that the stress will be a maximum at the inner corners, where $y = \pm t/2$ and $r = b$.

From an inspection of [1] it will be seen that if the disk thickness t, instead of being made constant, is varied so that t/r is a constant, the stress will be independent of r and (for small values of ϕ) uniform over the entire disk surface under the above assumptions. Such a disk (Fig. 2) would be tapered radially, so that its thickness at any point would be a linear function of the radius of that point.

The advantage of the radially tapered section over a disk of uniform thickness lies in the very materially increased flexibility that can be secured where space is limited in the direction of load application. A series of radially tapered disks, such as shown in Fig. 3, can often be advantageously substituted for helical springs, especially where loads are high and relatively small deflections are required. In such cases the advantages of the radially tapered disk spring may be summarized as follows:

1 The disk spring is adjustable in height and flexibility by adding or taking off disks.

2 It will withstand lateral loading as well as axial loading.

3 It can be so designed that no disk will be overloaded. Failure of one disk will not cause the complete loss of flexibility, nor will it increase the load on the remaining disks. A failure of one disk would not require replacement of the whole spring.

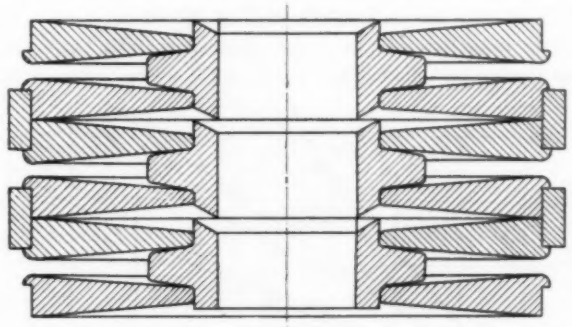


FIG. 3 STACK OF RADIALLY TAPERED DISKS

4 Helical springs of large size are difficult to heat treat uniformly—a disadvantage that can be overcome with the disk spring.

5 Disk springs can be designed to occupy very little space in the direction of load application.

Helical springs are the cheaper to manufacture unless production of the disks is high enough to warrant making forging dies. It may be noted that radially tapered disks have been successfully die forged from 0.90 to 1.00 carbon spring steel so that no machining was required on the disk faces.

I—THE DESIGN OF DISK SPRINGS

Formulas Derived by Approximate Theory

Formulas have been developed for deflection and stress of radially tapered disks using both the approximate theory (Appendix No. 1) and the more exact flat-plate theory (Appendix No. 2). The approximate theory is developed both for flat and dished disks. Since the flexibility decreases rapidly with dishing, the formulas given here apply to flat disks of radially tapered section only. Fig. 4 shows load-deflection curves for three disks similar in every respect except the dish angle. The dotted lines show the curves calculated by the use of the formulas developed farther on. Deflections are 50 per cent larger for the flat disk than for the one having a dish angle of 0.1 radian, and

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[3] Discussion by S. Timoshenko of paper by Waters and Taylor, "The Strength of Pipe Flanges," *Mechanical Engineering*, vol. 49 (1927), p. 1343.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

150 per cent larger than for the one with a dish angle of 0.15 radian. Formulas developed from the approximate theory for flat disks are as follows:

$$P = \frac{4}{3}\pi E\phi k\left[\frac{\phi^2}{8}(a-b)^2 + \frac{k^2}{3}(a^2+ab+b^2)\right] \quad [2]$$

$$S = Ek\phi \quad [3]$$

$$\delta = \phi(a-b) \quad [4]$$

where a = outer radius of disk
b = inner radius of disk
α = ratio of a to b
t = disk thickness at radius r
t_b = disk thickness at radius b
k = thickness constant, where $t = 2kr$
ϕ = angle of rotation of disk section under load, radians
P = total axial load
δ = axial deflection of disk
S = maximum stress, and
E = modulus of elasticity.

Formula [2] may be modified as follows:

$$P = \frac{4.19\,Ek\delta}{a-b}\left[\frac{\delta^2}{8} + \frac{k^2}{3}(a^2+ab+b^2)\right] \quad [5]$$

Comparison of Approximate Theory With Elastic-Flat-Plate Theory

The preceding formulas were derived on the basis of the assumption that diametral sections of the springs rotate without

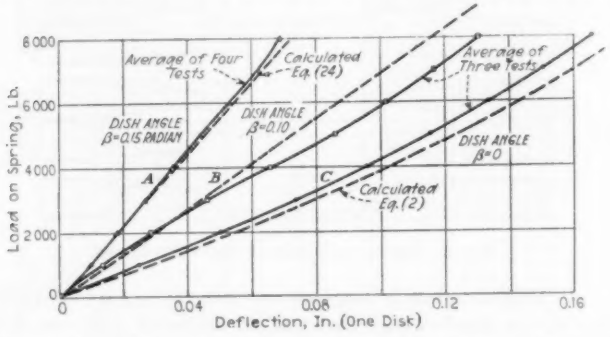


Fig. 4 Load-Deflection Curves on Three Disks Similar Except for Dish Angle

(a = 3.875 in.; b = 1.50 in.; k = 0.0445. Elastic limit not exceeded in any of these tests.)

distortion, deformation consisting principally of a rotation of these sections. This assumption, while very nearly correct for springs having ratios of outer to inner diameters less than about 2, is evidently not correct for springs with comparatively large ratios of diameters. In order to get an idea of the magnitude of the error involved by using this assumption, the elastic-flat-plate theory was applied to the case of a flat disk as shown in Fig. 2. (See Appendix No. 2.) The final results were as follows:

The maximum stress is

$$S = C\frac{P}{t_b^2} \quad [6]$$

where C is a factor depending on the ratio α of outer to inner radius and which may be taken from the curve of Fig. 5.

The maximum deflection is

$$\delta = M\frac{Pa^2}{Et_b^3} \quad [7]$$

where the factor M depends on the ratio α of outer to inner radius and may be taken from the curve of Fig. 6.

It should be noted that the elastic-flat-plate theory assumes small deflections. Therefore in order to compare the results of the approximate method as given by Equations [2] and [3] with those given by the more exact flat-plate theory, it is necessary to assume small deflections, i.e., that the angle ϕ in Equations [2] and [3] is small. The approximate Equations [2] and

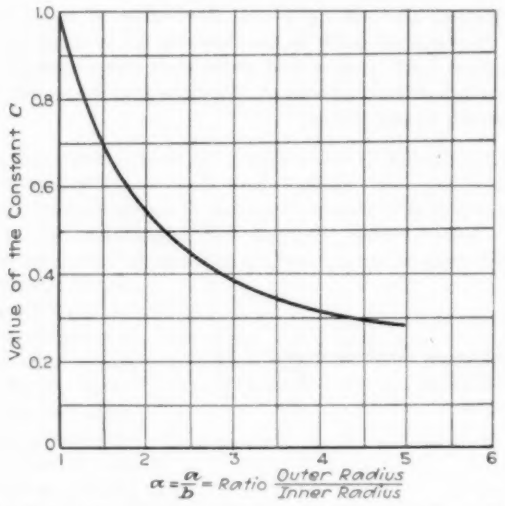


Fig. 5 Stress Constant for Radially Tapered Disk Springs

[3] (substituting for ϕ its value in terms of P) can then be written as follows:

$$S = C'\frac{P}{t_b^2} \quad [8]$$

where

$$C' = \frac{2.86}{\alpha^2+\alpha+1} \quad [8a]$$

and

$$\delta = M'\frac{Pa^2}{Et_b^2} \quad [9]$$

where

$$M' = \frac{5.73}{\alpha^2}\left[\frac{\alpha-1}{\alpha^2+\alpha+1}\right] \quad [9a]$$

Values of C, C', M, and M' are given in Table 1.

TABLE 1 VALUES OF THE CONSTANTS C, C', M, AND M'

For α =	1	1.25	1.5	2	3	4	5
C =	0.985	0.819	0.696	0.536	0.382	0.315	0.280
C' =	0.954	0.75	0.602	0.408	0.22	0.136	0.092
M =	0	0.243	0.271	0.213	0.110	0.062	0.039
M' =	0	0.242	0.268	0.205	0.098	0.0506	0.0296

An examination of this table shows that for values of the ratio α of outer to inner diameters less than 3 there is agreement between the constants M and M' within 10 per cent. This indicates that where α is less than 3, the approximate theory is sufficiently accurate for practical purposes. However, the constants C and C' agree within 10 per cent only for values of α less than about 1.4. Therefore, in computing the stress in disk springs it is usually necessary to use the more exact Equation [6], in order to obtain sufficiently accurate results for practical work.

It will be noted that the elastic-flat-plate theory assumes small deflections; consequently it does not take into account the effect of dishing of the plate as the deflection increases. This

effect tends to cause a stiffening of the plate with increasing load. The approximate theory, however, as shown by Equation [5], takes this effect into account. This equation may be rewritten as follows:

$$P = \frac{E\, t_b^3 \delta}{M' a^2}\left[1 + \frac{1.5\ \delta^2}{t_b^2(\alpha^2 + \alpha + 1)}\right] \ldots\ldots\ldots [10]$$

where M' is as given in Equation [9a].

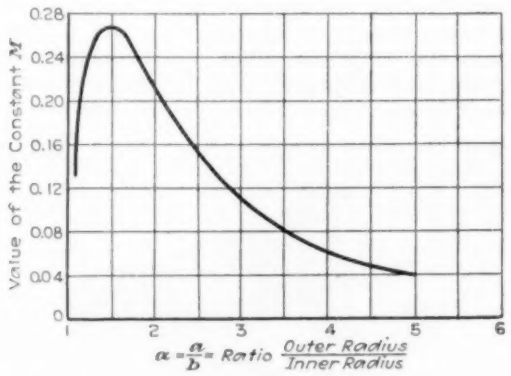


FIG. 6 DEFLECTION CONSTANT FOR RADIALLY TAPERED DISK SPRINGS

If δ is small, this equation reduces to Equation [9]. If now in this equation M' be arbitrarily replaced by M,

$$P = \frac{E\, t_b^3 \delta}{M a^2}\left[1 + \frac{1.5\ \delta^2}{t_b^2(\alpha^2 + \alpha + 1)}\right] \ldots\ldots\ldots [11]$$

For small deflections, i.e., where the second term in the brackets may be neglected, this equation agrees with that obtained by elastic-flat-plate theory. For large deflections the equation corrects for the effect of dish in the same proportion as is done in the approximate equation.

It should be noted that for ratios of diameters α less than 2, which includes most practical cases, results computed by this equation will agree with those calculated by Equation [5] within about 4 per cent in most cases. Hence for practical work Equation [5], in general, is sufficiently accurate.

CORRECTION FOR INWARD DISPLACEMENT OF LOAD

It should be noted that in actual practice the load on the spring does not act exactly at the edge as shown in Fig. 2, but is displaced inward somewhat as shown in Fig. 7. In calculating the actual stress or deflection, sufficient accuracy will be had if the actual load P is reduced in the ratio $d/(a - b)$ (Fig. 7). This takes into account the fact that these loads are displaced inwardly as shown.

PROPORTIONS OF SPRINGS FOR MAXIMUM FLEXIBILITY

It has been found that in designing a disk for a given load and a given stress the deflection varies with α, or the ratio of the outer radius a to the inner radius b. Fig. 8 shows how the deflections of a series of disks, all with the same outside diameter and designed for the same load, vary with α. The variation of α also produces a change in the thickness constant k. Deflections are a maximum when $\alpha = 2$. It has been found that this maximum always falls somewhere between $\alpha = 1.95$ and $\alpha = 2.05$. This value of α not only gives a maximum deflection, but results in disks of greater lateral stability and of better proportions for forging and heat treating than disks having larger ratios of outer to inner diameter.

II—TESTS AND COMPARISON WITH THEORY

DESCRIPTION OF TEST APPARATUS

In order to check up the theory developed in Part I by actual test, a test arrangement was designed as shown schematically in Fig. 9. In this arrangement, load was applied in a testing

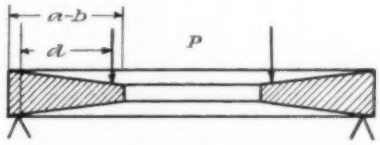


FIG. 7 APPLICATION OF LOAD ON ACTUAL DISK SPRING

machine through the two steel cylinders A to a heavy ring B, which applied the load uniformly around the outer circumference of the disk spring C. The disk spring was supported by the cylinder F resting on the lower head of the testing machine. Sufficient clearance was allowed between the edges of the disk spring and the cylinder F and ring B, respectively, to make certain there would be no binding during application of the load. In most of the tests the supporting edge of the ring was beveled as shown, greatly exaggerated, so that the point of load application was very definite. Special extensometers E placed along the inner edge of the spring made possible the measurement of maximum stress. At the same time it was possible to read the extensometers while the spring was loaded. In all cases strain measurements were made on diametrically opposite sides of the spring to determine whether or not the load was central. For measuring deflections a dial gage D was used.

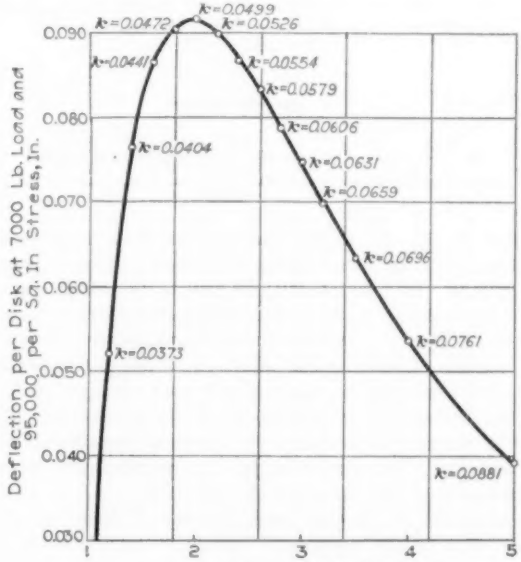


FIG. 8 DISK-SPRING DEFLECTION AS FUNCTION OF RATIO OF OUTER TO INNER RADIUS

(k = thickness constant where $t = 2kr$; a = outer radius = 4 in. in all cases.)

A photograph of the test arrangement in the machine is shown in Fig. 10. The disk spring C, the heavy ring B, the extensometer E, and the two cylinders A are here plainly visible.

TESTS ON FLAT DISK SPRINGS

A series of flat disk springs having various ratios of outer to inner diameter and various values of the thickness constant k

were tested. Data relative to these springs, including dimensions and method of fabrication, are given in Table 2.

TABLE 2 DATA ON SPRINGS TESTED

Spring designation	Outer radius a, inches	Inner radius b, inches	Outer thickness t_a, inches	Inner thickness t_b, inches	Method of making
A′	3.81	1.5	0.305	0.120	Machined, heat treated, and ground
C	3.81	1.5	0.339	0.133	Forged, heat treated, and ground
D	3.81	1.5	0.267	0.105	Machined, heat treated, and ground
I	3.81	2.54	0.220	0.330	Machined and heat treated
VI	3.81	1.91	0.193	0.388	Machined, heat treated, and ground
E	4.25	1.563	0.667	0.246	Machined, heat treated, and ground
F	3.87	1.50	0.344	0.133	Machined, heat treated, and ground

The results of the tests on these springs are shown graphically in the load-deflection and load-stress curves of Figs. 11–21 and

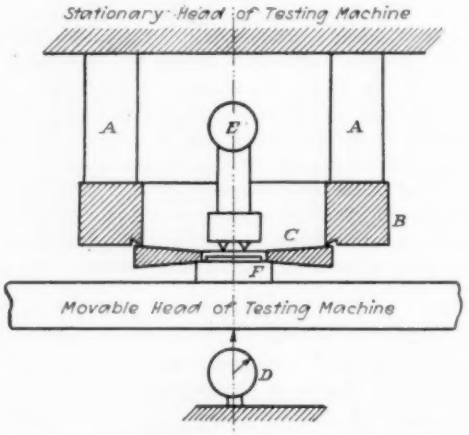


FIG. 9 ARRANGEMENT FOR TESTING DISK SPRINGS
(*A*, steel cylinder; *B*, ring; *C*, disk spring; *D*, dial gage; *E*, extensometer; *F*, cylinder.)

on curve *C*, Fig. 4. In all cases the test results are shown in full lines, while the theoretically calculated curves are shown in dotted lines. In calculating these curves allowance was made for the fact that the loads on the springs were displaced inwardly as shown in Fig. 7. All deflections were calculated on the basis of a modulus of elasticity $E = 30 \times 10^6$ lb. per sq. in., since tensile-test results on this material indicated a value close to this.

In calculating the theoretical curves for springs A′, C, D, I, and VI, the more exact theory as given by Equations [6] and [10] was used. In calculating the theoretical deflection curves for springs E and F, Equation [5] was used.

In all cases the stresses were determined from the strains, using a value of the modulus of elasticity equal to 30×10^6 lb. per sq. in. The strains were measured at the inside edge of the spring where the maximum stress is indicated theoretically. This was done especially since failure of these springs usually starts under fatigue loading at the inner edge.

Examination of the test curves shows that there is very good agreement between theory and test, in the case of stress, for the flat springs. For deflection, at the lower loads, theory and test agree quite well, but at higher loads the test curve in certain cases deviated somewhat from the theoretical. This deviation was caused by the line of application of the load tending to move inward in certain cases, i.e., the distance d (Fig. 7) becomes less. Consequently the spring becomes relatively stiffer at the higher loads. This effect was not so pronounced in springs C, A′, D, F, and I, where the location of the supporting edge was quite definite. However, in the case of springs VI and E the inner ring supporting the springs had a sloping shape similar to that shown on the sketch of Fig. 3. This accounts for the steepness of the deflection and stress curves at the higher loads in these cases.

Duplicate tests were made on part of these springs with practically the same results.

TESTS ON DISHED SPRINGS

Deflection tests were also made on dished disk springs of the type shown in Fig. 22. The results of the deflection tests on these disk springs are shown on curves *A* and *B*, Fig. 4. The dotted lines show the theoretical curves as computed using Equation [24], Appendix No. 1. It may be seen that the approximate Equation [24] gives satisfactory results for the case of deflection of these types of spring.

Experimental verification of the stress formulas for dished disk springs, given in Appendix No. 1, has not yet been obtained. Because of the assumptions made in the derivation, it is doubtful

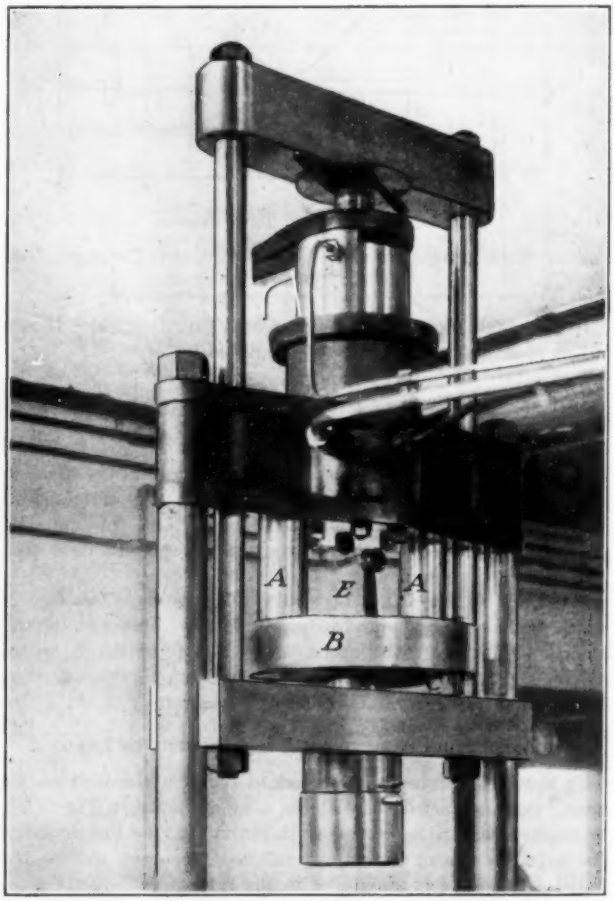


FIG. 10 VIEW OF TESTING ARRANGEMENT

if these formulas are very accurate; however, they at least give the designer an idea of the stress involved.

III—CONCLUSIONS

Since the stress distribution along a radial section of these disk springs is approximately constant for values of the ratio α of outer to inner diameter less than about 2, it may be said that these springs utilize the material about as efficiently as do transverse elliptic springs (which may be considered as bars under a constant moment). They therefore do not utilize the material quite as efficiently as helical springs, but, on the other hand, have certain advantages over helical springs which may make them preferable for certain applications.

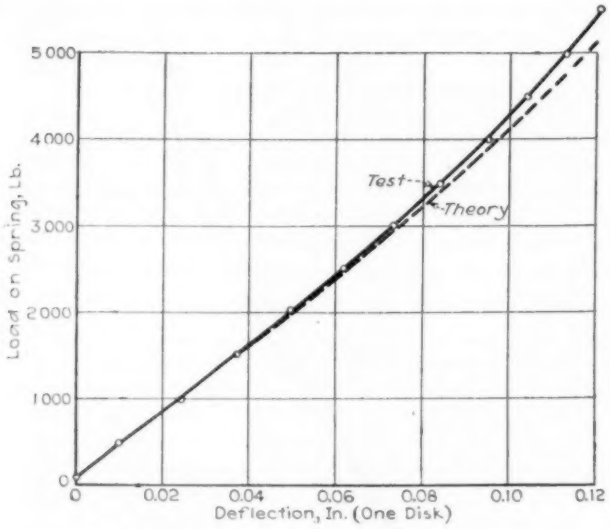


FIG. 11 DEFLECTION TEST ON DISK SPRING "C"
(a = 3.81 in.; b = 1.5 in.; t_b = 0.1335 in.; t_a = 0.339 in.)

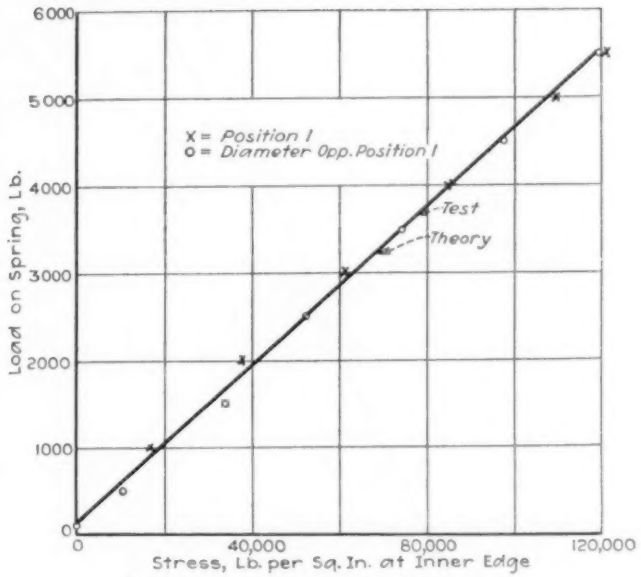


FIG. 12 STRESS TEST ON DISK SPRING "C"
(a = 3.81 in.; b = 1.5 in.; t_a = 0.339 in.; t_b = 0.1335 in.)

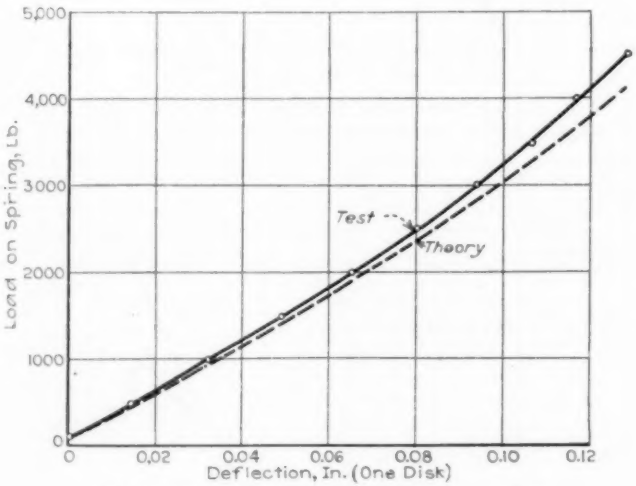


FIG. 13 DEFLECTION TEST ON DISK SPRING "A"
(a = 3.81 in.; b = 1.5 in.; t_b = 0.120 in.; t_a = 0.305 in.)

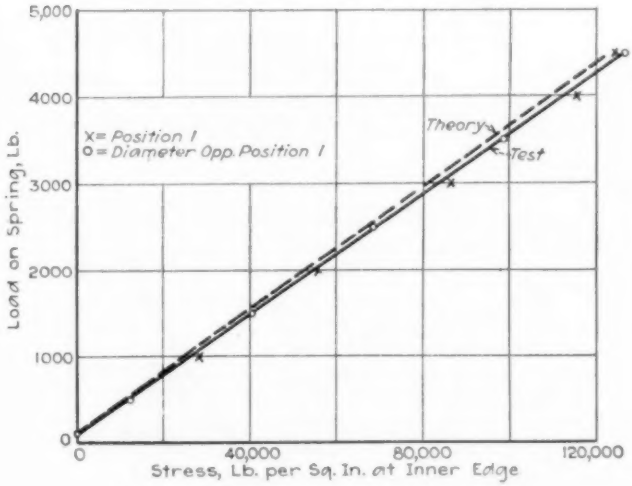


FIG. 14 STRESS TEST ON DISK SPRING "A"
(a = 3.81 in.; b = 1.5 in.; t_b = 0.120 in.; t_a = 0.305 in.)

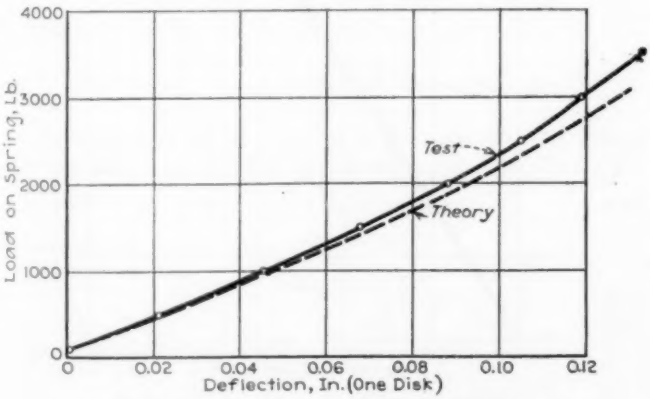


FIG. 15 DEFLECTION TEST ON DISK SPRING "D"
(a = 3.81 in.; b = 1.5 in.; t_b = 0.105 in.; t_a = 0.267 in.)

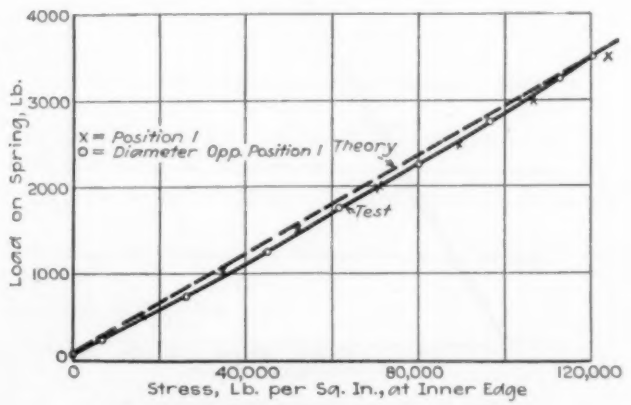


FIG. 16 STRESS TEST ON DISK SPRING "D"
(a = 3.81 in.; b = 1.5 in.; t_b = 0.105 in.; t_a = 0.267 in.)

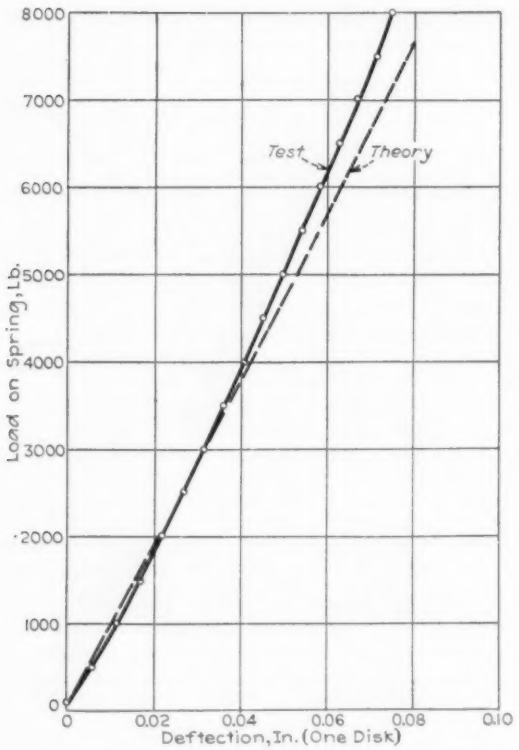


FIG. 17 DEFLECTION TEST ON DISK SPRING "I"
(a = 3.81 in.; b = 2.54 in.; t_b = 0.22 in.; t_a = 0.33 in.)

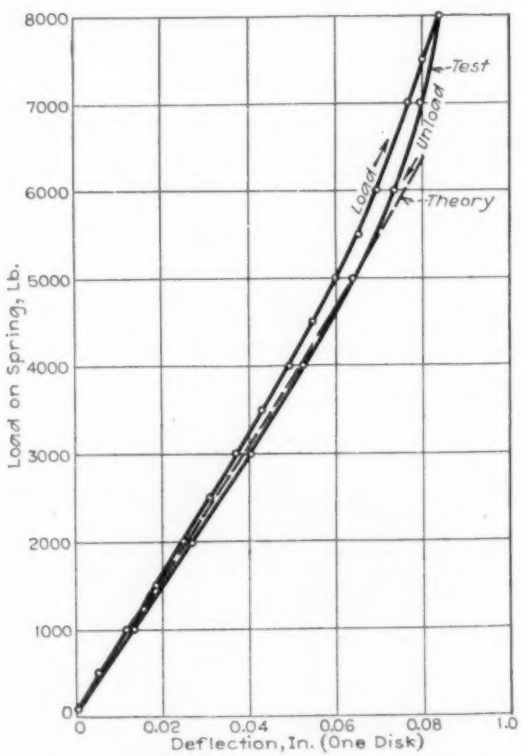


FIG. 19 DEFLECTION TEST ON DISK SPRING "VI"
(a = 3.81 in.; b = 1.91 in.; t_b = 0.193 in.; t_a = 0.388 in.)

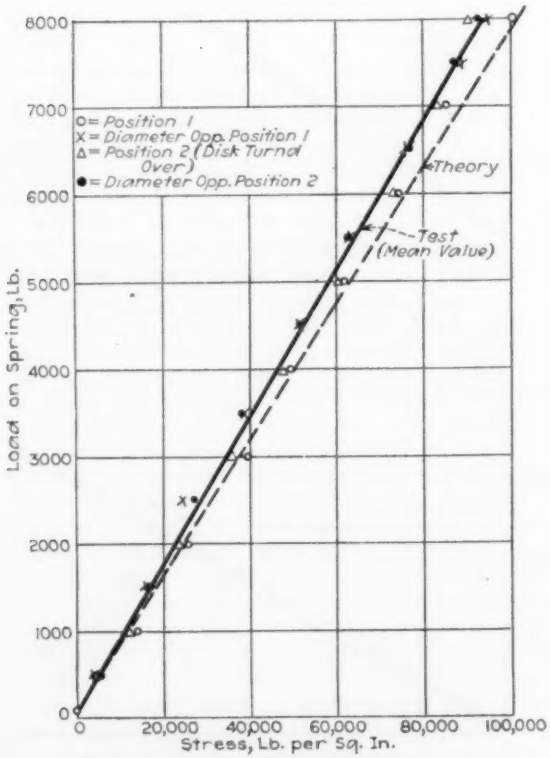


FIG. 18 STRESS TEST ON DISK SPRING "I"
(a = 3.81 in.; b = 2.54 in.; t_b = 0.22 in.; t_a = 0.33 in.)

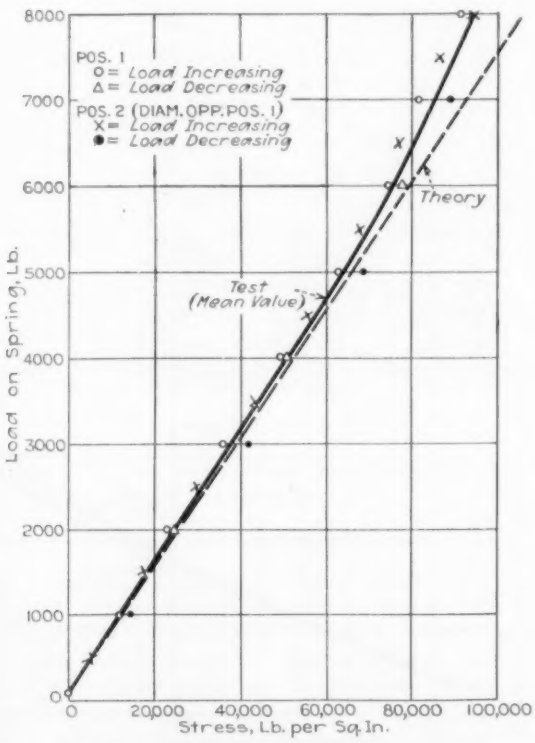


FIG. 20 STRESS TEST ON DISK SPRING "VI"
(a = 3.81 in.; b = 1.91 in.; t_b = 0.193 in.; t_a = 0.388 in.)

The flat tapered spring, because of its greater flexibility and better stress distribution, is, in general, preferable to the dished tapered spring or the Belleville type spring of constant thickness.

Test results, including both strain measurements and deflection tests, indicate that the formulas given by the elastic-flat-plate theory (Equations [6] and [11]) are sufficiently accurate for use in the practical design of flat disk springs of this type. Test results also indicate that the formulas given for the deflection of both flat springs by approximate theory (Equations [2] and [5]) are sufficiently accurate for most practical work, as is Equation [24] in the case of dished springs.

Acknowledgments

The authors wish to acknowledge their indebtedness to Messrs. C. Bethel and J. M. Lessells of the Westinghouse Company for encouragement received in this work. They are also indebted to Dr. S. Timoshenko of the University of Michigan and Dr. A. Nadai of the Westinghouse Company for valuable suggestions, and to Mr. M. Karelitz of the Westinghouse Company for preparing some of the springs for test.

Appendix No. 1

Derivation of Stress and Load-Deflection Formulas for Flat and Dished Radially Tapered Disks—Approximate Theory

IN ADDITION to the symbols used heretofore,

β = dish angle (= 0 for flat disks) in radians

c = radius at point of rotation of the disk section under load

$c - x$ = radius of any point.

The complete disk is shown in Fig 22(*a*). It is assumed in the approximate theory that cross-sections remain plane under load, and that their boundaries remain undistorted. Deformation thus consists of rotation of the section about some point O, at a radius c, through a small angle ϕ.

Consider element G [Figs. 22(*b*) and 22(*c*)]. Its initial length equals l_1.

$$l_1 = d\theta(c - x\cos\beta + y\sin\beta) \quad [12]$$

After deformation this length becomes $l_2 = d\theta[c - x\cos(\beta - \phi) + y\sin(\beta - \phi)]$ or

$$l_2 = d\theta[\,c - x\cos\beta\cos\phi - x\sin\beta\sin\phi + y\sin\beta\cos\phi - y\cos\beta\sin\phi] \quad [13]$$

Subtract [13] from [12] and let $\cos\beta = 1$, $\cos\phi = 1$, $\sin\beta = \beta$, and $\sin\phi = \phi$, since β and ϕ are small; then

$$l_1 - l_2 = d\theta[\phi(\beta x + y)] \quad [14]$$

This equals the elongation, and since $S = eE$,

$$S = \frac{E\phi(\beta x + y)}{c - x} \quad [15]$$

The moment of the forces acting on the element G [Fig. 22(*c*)] about an axis through O perpendicular to the paper is:

$$dM' = d\theta\, S\, dx\, dy\,(\beta x + y) \quad [16]$$

Substituting [15] in [16],

$$dM' = \frac{d\theta\, E\phi\,(\beta x + y)^2}{c - x}\, dx\, dy \quad [17]$$

Then the total moment will be

$$M' = \int_{c-a}^{c-b}\int_{-k(c-x)}^{k(c-x)} \frac{d\theta\, E\phi(\beta x + y)^2}{c - x}\, dy\, dx \quad [18]$$

Integrating and substituting limits,

$$M' = d\theta\, E\phi\,\frac{2k}{3}\,(a - b)\left\{\left(\beta^2 + \frac{k^2}{3}\right)\left[3c^2 - 3c\,(a + b) + a^2 + ab + b^2\right] - k^2c(2c - a - b) + k^2c^2\right\} \quad [19]$$

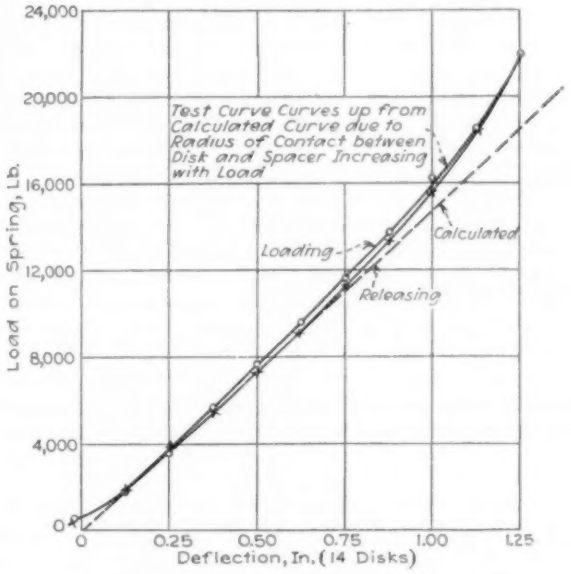


Fig. 21 Deflection Test on Disk Spring "E"
(a = 4.25 in.; b = 1.56 in.; t_b = 0.246 in.; t_a = 0.667 in.)

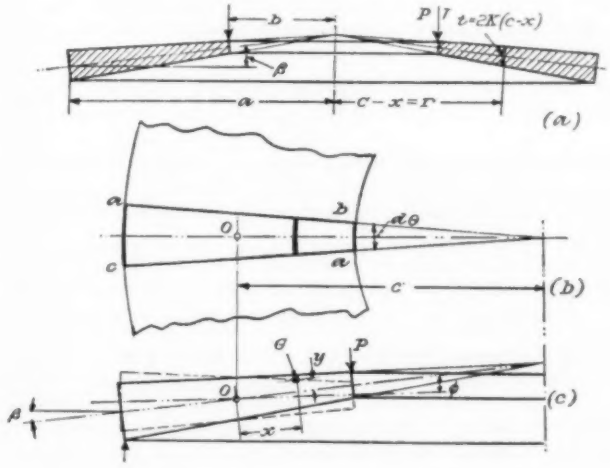


Fig. 22 Dished Disk Spring

The two unknowns are ϕ and c. To evaluate these, we know from statics [Fig. 22(*b*)], that

(*a*) The moment M' found in [18] equals the external moment; and

(*b*) The resultant of all forces acting on the element equals zero.

From (*a*), letting $\cos\beta = 1$,

$$M' = d\theta\,\frac{P(a - b)}{2\pi} \quad [20]$$

This substituted in [19] gives

$$P = \frac{4}{3}\pi E\phi k\left\{\left(\beta^2 + \frac{k^2}{3}\right)\left[3c^2 - 3c\,(a+b) + a^2 + ab + b^2\right] - k^2c\,(2c - a - b) + k^2c^2\right\} \quad [21]$$

Next considering (*b*),

$$d\theta \int_{c-a}^{c-b}\int_{-k(c-x)}^{k(c-x)} S\,dy\,dx = 0$$

or

$$d\theta \int_{c-a}^{c-b}\int_{-k(c-x)}^{k(c-x)} \frac{\beta x + y}{c - x}\,dy\,dx = 0 \quad [22]$$

Solving,

$$c = \frac{a+b}{2} \quad [23]$$

Substituting [23] in [21],

$$P = \frac{4}{3}\pi E\phi k\left[\frac{\beta^2}{4}(a-b)^2 + \frac{k^2}{3}(a^2 + ab + b^2)\right] \quad [24]$$

From Fig. 22(*c*), the axial deflection $\delta = \phi(a - b)$. From [15], when the dish angle $\beta = 0$ (flat disk),

$$S = \frac{E\phi y}{c - x}$$

or as $y = t/2 = k\,(c - x)$,

$$S = E\phi k \text{ (for flat disks)} \quad [25]$$

Equation [25] indicates that the stress is a constant in the outermost fibers, and maximum material economy obtains. In the case of dished plates the maximum stress will be at the upper, inner edge, or

$$S = E\phi\left(\beta\frac{a-b}{2b} + k\right) \quad [26]$$

This does not give maximum material economy, but gives much better economy than a disk of uniform thickness. The shape of a dished disk for maximum material economy would be $t = 2k\,(c - x) - 2\beta x$.

The above formulas are approximate because they are based on certain assumptions given at the beginning of Appendix No. 1. A somewhat different result can be obtained as follows:

$$l_1 - l_2 = d\theta[x\cos\beta\,(\cos\phi - 1) + x\sin\beta\sin\phi + y\sin\beta\,(1 - \cos\phi) + y\cos\beta\sin\phi] \quad [27]$$

But $(1 - \cos\phi) = 2\sin^2\phi/2$, and letting $\cos\beta = 1$, $\cos\phi = 1$, $\sin\beta = \beta$, $\sin\phi = \phi$, and $\sin^2\phi/2 = \phi^2/4$,

$$l_1 - l_2 = d\theta\left[x\phi\left(\beta - \frac{\phi}{2}\right) + y\phi\left(\frac{\beta\phi}{2} + 1\right)\right]$$

Neglecting $y\beta\phi^2/2$ since all factors are small,

$$l_1 - l_2 = d\theta\left\{\phi\left[\left(\beta - \frac{\phi}{2}\right)x + y\right]\right\}$$

Solving as before,

$$P = \frac{4}{3}\pi E\phi k\left[(\beta - \phi)\left(\beta - \frac{\phi}{2}\right)\frac{(a-b)^2}{4} + \frac{k^2}{3}(a^2 + ab + b^2)\right] \quad [28]$$

or, when $\beta = 0$ (flat disks),

$$P = \frac{4}{3}\pi E\phi k\left[\frac{\phi^2}{8}(a-b)^2 + \frac{k^2}{3}(a^2 + ab + b^2)\right] \quad [29]$$

and

$$S = E\phi\,\frac{\left(\beta - \frac{\phi}{2}\right)x + y}{c - x} \quad [30]$$

or for flat disks,

$$S = E\phi\left[\frac{\phi(a-b)}{4b} + k\right] \quad [31]$$

Appendix No. 2

DERIVATION OF MORE EXACT FORMULAS FOR DISK SPRINGS USING FLAT-PLATE THEORY

IN THIS appendix, formulas for maximum stress and deflection are derived for a flat circular plate with a central hole, loaded uniformly along the edges, and having a thickness which is directly proportional to the radius (see Fig. 2). In addition to the nomenclature previously given, let

ν = Poisson's ratio (= 0.3 for steel)
r = radius of element considered
$s = \sqrt{\frac{13}{4} - 3\nu}$
m_1 = radial bending moment per unit length at radius r
m_2 = tangential bending moment per unit length at radius r
w = deflection at radius r
$\phi = -\frac{dw}{dr}$ = slope in radial direction at radius r
D = flexural rigidity of the plate at radius r
$D = K_1r^3$

where $K_1 = \frac{2Ek^3}{3(1 - \nu^2)}$

The equilibrium of the forces acting on a small element of the plate at radius r gives the following equations:[4]

$$m_1 + \frac{dm_1}{dr}r - m_2 + \frac{P}{2\pi} = 0 \quad [32]$$

$$m_1 = D\left(\frac{d\phi}{dr} + \nu\frac{\phi}{r}\right) \quad [33]$$

$$m_2 = D\left(\frac{\phi}{r} + \nu\frac{d\phi}{dr}\right) \quad [34]$$

It should be noted that in this case the flexural rigidity is:

$$D = K_1r^3 \quad [35]$$

Differentiating [33] with respect to r gives

$$\frac{dm_1}{dr} = K_1r^3\left(\frac{d^2\phi}{dr^2} + \frac{\nu}{r}\frac{d\phi}{dr} - \frac{\nu\phi}{r^2}\right) + 3K_1r^2\left(\frac{d\phi}{dr} + \nu\frac{\phi}{r}\right) \quad [36]$$

Substituting these equations in [32] gives

$$\frac{d^2\phi}{dr^2} + \frac{4}{r}\frac{d\phi}{dr} + (3\nu - 1)\frac{\phi}{r^2} = -\frac{P}{2\pi K_1r^4}$$

The complete solution to this equation is[5]

[4] See, for example, "Applied Elasticity," by Timoshenko and Lessells, p. 274, for derivation of these equations.

[5] The authors are indebted to Dr. Nadai of the Research Department for suggesting the method of integration of this equation.

$$\phi = C_1 r^{-\frac{3}{2}+s} + C_2 r^{-\frac{3}{2}-s} - \frac{P}{2\pi K_1 (\nu - 1) r^2} \ldots [37]$$

The boundary conditions are:

$$\text{For } r = a,\ m_1 = 0, \text{ and } \left(\frac{d\phi}{dr} + \nu \frac{\phi}{r}\right)_{r=a} = 0$$

$$\text{For } r = b,\ m_1 = 0, \text{ and } \left(\frac{d\phi}{dr} + \nu \frac{\phi}{r}\right)_{r=b} = 0$$

Using these equations, the values of the constants C_1 and C_2 are obtained:

$$C_1 = \frac{P(2-\nu)}{6\pi K_1(1-\nu)(-\frac{3}{2}+s+\nu)}\left[\frac{a^{-\frac{1}{2}}b^{-s} - b^{-\frac{1}{2}}a^{-s}}{\alpha^s - \alpha^{-s}}\right]. [38]$$

$$C_2 = \frac{P(2-\nu)}{6\pi K_1(1-\nu)(-\frac{3}{2}-s+\nu)}\left[\frac{a^s b^{-\frac{1}{2}} - b^s a^{-\frac{1}{2}}}{\alpha^s - \alpha^{-s}}\right]. . [39]$$

The maximum stress in the plate will be

$$S = \frac{6(m_2)_{r=b}}{t_b^2} \ldots\ldots\ldots\ldots\ldots [40]$$

Using Equations [34], [37], [38], and [39] and the derivative of [37] in [40] and taking $r = b$, the maximum stress becomes

$$S = \frac{CP}{t_b^2} \ldots\ldots\ldots\ldots\ldots [41]$$

where

$$C = \frac{1}{\pi(1-\nu)}\left[\frac{\left(2-\nu\right)\left(1-\frac{3\nu}{2}+s\nu\right)\left(\alpha^{-\frac{1}{2}} - \alpha^{-s}\right)}{-\frac{3}{2}+s+\nu} - \frac{\left(2-\nu\right)\left(1-\frac{3\nu}{2}-s\nu\right)\left(\alpha^{s} - \alpha^{-\frac{1}{2}}\right)}{-\frac{3}{2}-s+\nu}}{\alpha^s - \alpha^{-s}} + 1 - 2\nu\right] \ldots [41a]$$

To get the deflection w of the plate the following equation is used:

$$\phi = -\frac{dw}{dr}$$

or

$$w = -\int \phi dr + C_3$$

Using the value of ϕ given by [37] and integrating, the deflection becomes

$$w = -\frac{C_1 r^{-\frac{1}{2}+s}}{-\frac{1}{2}+s} - \frac{C_2 r^{-\frac{1}{2}-s}}{-\frac{1}{2}-s} - \frac{P}{6\pi K_1(\nu-1)r} + C_3 . . [42]$$

For determining C_3, the condition that when $r = a$, $w = 0$ is used; hence

$$C_3 = \frac{C_1 a^{-\frac{1}{2}+s}}{-\frac{1}{2}+s} + \frac{C_2 a^{-\frac{1}{2}-s}}{-\frac{1}{2}-s} - \frac{P}{6\pi K_1(\nu-1)a}$$

Using this value of C_3 in [42] and taking $r = b$ (since the maximum deflection will occur where $r = b$), the maximum deflection becomes

$$\delta = M\frac{Pa^2}{Et_b^3} \ldots\ldots\ldots\ldots\ldots [43]$$

where

$$M = \frac{0.637(1+\nu)}{\alpha^3}\left[\frac{\nu-2}{\alpha^s - \alpha^{-s}}\left\{\frac{-\alpha^{1-s} - \alpha^s + 2\sqrt{\alpha}}{(-\frac{1}{2}+s+\nu)(-\frac{1}{2}+s)} + \frac{\alpha^{s+1} + \alpha^{-s} - 2\sqrt{\alpha}}{(-\frac{3}{2}-s+\nu)(-\frac{1}{2}-s)}\right\} + \alpha - 1\right] \ldots . [43a]$$

Discussion

DR. ING. A. NADAI.[6] The authors have not only contributed a comparatively simple method to the calculation of the deformations of flat disks, but also have found an exact solution of a comparatively difficult problem in the theory of the bending of flat plates with variable thickness. It is known that, under the latter circumstances, in most cases which are of a practical interest for the engineer, it is not possible to find an exact solution of the differential equation for the deflected middle surface of the plate. One case where this was possible is the case of a circular plate with a varying thickness which is proportional to the distance of the point from the center of the plate.

With regard to the approximate method the authors use for the tapered plates and dished springs, the good agreement they find between the results of their calculations and the tests is due to the fact that they had to deal with freely supported annular plates in which the radial stress is zero, both on the inner and outer circle, and therefore also a negligible quantity throughout the plate. In some other cases, the agreement of their approximate theory with actual cases might perhaps not be quite so satisfactory. An example in which the radial stress could not be neglected is the bending of a circular plate which does not contain any hole and in which both the radial and the tangential stresses must become gradually equal if one approaches the center of the disk.

HAROLD C. CLAUSEN.[7] The writer is interested in the possible use of the radially tapered disk spring in railroad work where heavy helical springs are now in use. In order to estimate the value of the particular type of spring, it is useful to know the resilience of the spring—that is, the energy that can be stored up in the spring per unit volume.

For a helical spring of round wire subjected to axial loads the resilience is equal to $\frac{1}{4}\frac{S^2}{G}$, and for a beam as used in carriage springs where the maximum stress is reached in every section and the section is rectangular the resilience is equal to $\frac{1}{6}\frac{S^2}{E}$. The writer would like to know if the authors can give a formula similar to this for the radially tapered disk spring.

It would appear that the radially tapered disk spring is not as efficient as the helical spring for the following reasons:

1 In the formula we would use E (Young's modulus of elasticity) which is about $2^1/_2$ times greater than G (torsional modulus of rigidity).

2 In the case of the round helical spring the most of the material is situated in the region of maximum stress.

3 In the case of the radially tapered disk spring the spacers will weigh some appreciable proportion of the weight of the disks, and these spacers are so much dead weight not entering into the efficiency of the spring.

G. M. EATON.[8] The writer has followed the development of

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this new type of spring with a great deal of interest. Assuming that it is desirable to have a spring carry successfully the highest stresses, the writer would call attention to two ways in which the resistance of the spring to service stress can be improved. If correct in the understanding that under fatigue test the springs fail with a fatigue nucleus on the inner perimeter and on the tension corner, drop-forged blanks can never be as successful as rolled blanks.

The particular point at issue is that, in order to withstand tension most ably, it is desirable to have such structural grain distortion, as occurs during the forging operations, result in an elongation of the grains in the direction of the tension stress. This is best secured by a rolling operation such as is used in general in the production of quality rings of steel.

The second point has to do with the stresses residual in the spring after the quench and draw. It is quite obvious that, dependent on the method of quenching, the residual stress in the critical fiber noted may be either tension or compression. If the rings are quenched, with the first contact of the quenchant with the steel on the inner perimeter it is fundamental that when the ring is cold these critical fibers will be in compression. When the spring starts to pick up load, it is clear that the first increments of load will reduce the compression stress on the critical fibers. As the load increases the stress will approach zero and then pass through into tension. In this connection the severity of the quench will have to be guarded, but in quantity production on automatic quenching fixtures it is possible to control the transient and residual stresses that are set up within practical limits that will result in a superior spring.

Authors' Closure

Regarding Dr. Nadai's remarks, the authors agree that the approximate method will only hold with good accuracy where the ratio of outer to inner radius α is fairly small. For example, as mentioned in the paper, for values of α greater than 3 the approximate method will be off 10 per cent or more for deflection as compared with the more exact method, but in the case of stress, to obtain 10 per cent accuracy, the value of α must be 1.4 or less. These and Dr. Nadai's remarks make it obvious that it is necessary to exercise judgment in applying the results given by the approximate method.

Mr. Clausen remarks that the resilience of the helical spring is equal to $\frac{1}{4}\frac{S_s^2}{G}$, while that of the elliptic spring is $\frac{1}{6}\frac{S^2}{E}$. Using the shearing modulus G in the case of the helical spring, for steel it is generally agreed that $G = \frac{E}{2.6}$. This gives $G = \frac{30 \times 10^6}{2.6} = 11.5 \times 10^6$, where $E = 30 \times 10^6$ lb. per sq. in. However, it must also be remembered that recent tests have shown that in the case of pure shear the yield point is only about 0.57 times the yield point in tension. Also fatigue tests of spring steel indicate that the endurance limit in reversed torsion is only about half that in reversed flexure.[9] Therefore, for purposes of computation, we may take $S_s = 0.57\,S$. This gives for the helical spring a resilience in terms of S and E as follows:

$$U_1 = \frac{1}{4}\frac{S_s^2}{G} = \frac{1}{4}\frac{(0.57S)^2 \times 2.6}{E} = 0.21\frac{S^2}{E}$$

Comparing this with the resilience of the elliptic spring $U = 0.167\,\frac{S^2}{E}$, it is seen that the helical spring is somewhat more resilient in this case.

It should be noted that the foregoing holds only for helical springs having a *very* large index (ratio of coil diameter to wire diameter). For railway work many springs have an index of 4. For such springs the stresses are actually about 1.4 times the computed stress.[10] This fact will reduce the resilience in the ratio $\left(\frac{1}{1.4}\right)^2$, making the foregoing value for resilience of a helical spring with an index of 4

$$U_1' = \frac{0.21\,S^2}{E} \times \left(\frac{1}{1.4}\right)^2 = 0.11\frac{S^2}{E}$$

It is seen that in case of a helical spring with an index of 4, the resilience is actually less than the value $0.16\,\frac{S^2}{E}$ for an elliptic spring.

To calculate the resilience of the disk spring, one may proceed as follows: The deflection δ and load P (for small deflections) is, from Equations [6] and [7] of the paper,

$$\delta = \frac{MPa^2}{Et^3_b} \quad \ldots\ldots [1]$$

$$S = \frac{CP}{t_b^2} \text{ or } P = \frac{St^2_b}{C} \quad \ldots\ldots [2]$$

The total work done

$$W = \frac{1}{2}P\delta = \frac{1}{2}\frac{MPa^2}{Et_b^3} \times P = \frac{Ma^2}{2Et_b^3}\frac{S^2t^4_b}{C^2}$$

or

$$W = \frac{S^2Ma^2t_b}{2EC^2} \quad \ldots\ldots [3]$$

The volume of the material in a disk spring may be shown to be

$$V = \frac{2\pi}{3}a^2t_b\left(\alpha - \frac{1}{\alpha^2}\right) \quad \ldots\ldots [4]$$

The work per unit volume is therefore

$$U_2 = \frac{W}{V} = \frac{S^2}{E}\left[\frac{M}{4.18C^2\left(\alpha - \frac{1}{\alpha^2}\right)}\right] \quad \ldots\ldots [5]$$

where the factors M and C in this formula are to be taken from Figs. 5 and 6. Taking, for instance, $\alpha = 1.4$, we obtain from [5]

$$U_2 = 0.134\,\frac{S^2}{E}$$

Comparing this with the foregoing expression for resilience of an elliptic spring, it is seen that in this case the disk spring is somewhat less resilient than an elliptic spring $\left(\frac{1}{6}\frac{S^2}{E}\right)$ and somewhat more resilient than a helical spring with an index of 4. In the case of a disk spring having a ratio $\frac{a}{b} = \alpha = 2$, the resilience from [5] is $0.10\,\frac{S^2}{E}$, which makes it comparable to the helical spring having an index of 4 $\left(\text{resilience} = \frac{0.11\,S^2}{E}\right)$. From this it may be seen that the disk spring with an index of 2 has a resilience about the same as that of heavy helical springs with fairly small indices such as are used in railway work. Of

[9] Moore and Kommers, "The Fatigue of Metals," p. 147.

[10] "Stresses in Heavy Closely Coiled Helical Springs." *Trans. A.S.M.E.*, APM-51-17.

course, the resilience of the disk spring is much less than that of helical springs of very large index, but these latter are seldom used in railway work.

It is true, as Mr. Clausen points out, that the efficiency of the disk spring is somewhat reduced by the weight of the spacers.

Macro-etchings have been made of several disks that were manufactured by different methods and which failed in a fatigue test. These photographs show conclusively that the rolled disks have the best grain alignment, as Mr. Eaton predicts.

Experiments are being made with various methods of quenching to establish a method which will give the best results as regards residual stress conditions and uniformity of hardness.

APM-52-5

Performance of Oil-Ring Bearings

By G. B. KARELITZ,[1] EAST PITTSBURGH, PA.

This paper comprises notes on the performance of oil-ring-lubricated bearings. The safety of the bearing determined by the minimum oil-film thickness is discussed; a diagram for computation of this value is given. The end leakage and the amounts of oil supplied by oil rings are shown, and the mechanism of operation of oil rings is described. The friction losses and the heat dissipation in oil-ring bearings are estimated and a way to predetermine the temperature rise of a bearing is indicated.

OIL RINGS have been applied to bearings for at least 80 years. Bearings of this type exhibited in London in 1848[2] had a split ring on the shaft; the lower bearing half, however, was cut in two to make space for the ring. Designers recognized the importance of a continuous stream of oil through the bearing, apparently desiring to improve the cooling of the bearing. The present-day oil-ring bearing, with a continuous lower bearing half or with a slotted sleeve, was not introduced till much later.[3] The existence of a load-carrying oil film which requires a certain minimum supply of lubricant to compensate for end leakage from the bearing has only recently been discovered. These new ideas were the result of the work of several prominent physicists at the end of the last century. These men have also been responsible for the explanation of the mechanism of the oil film. Petroff[4] published his work in 1883; Reynolds[5] published in 1886 his explanation of the famous Tower's experiments made in England at the request of the Institution of Mechanical Engineers. However, these two papers were so difficult to read that they apparently did not make any impression in wide engineering circles. The idea of the load-carrying film of lubricant was further developed in 1904 in a paper by Sommerfeld,[6] who discussed in simpler terms the mechanism of an oil film in a very long bearing ("of infinite length") and by Michell,[7] who gave an analysis of the oil film under a plane shoe of finite dimensions. A practical application of the hydrodynamic principles of lubrication has since been made independently by Kingsbury and Michell in designing the well-known tilting-shoe type of thrust bearing.

The analytical expressions resulting from a hydrodynamical investigation of the oil film in a cylindrical bearing are rather involved. Numerical tables which allowed the presentation in graphical form of the interrelation between various elements in a bearing shell and the physical properties of the lubricant used were prepared by Harrison, Guembel, Howarth, Boswall,[8] and others. Most of this work has been prepared rather recently and is still being developed. However, sufficient information is now available to attempt a numerical analysis of the performance of oil-ring-lubricated bearings.

The oil-ring bearing is still holding an important place in the design of rotating machinery. It is true that in machines running at high speed, where the amount of heat created in the bearings requires artificial cooling, forced-feed lubrication is gradually replacing the oil rings, the practice of cooling the bearing shell proper thereby being abandoned. But the oil-ring bearing is a self-contained unit requiring very little attention or maintenance and is very desirable wherever it can be used. Efforts are therefore made, on the one hand, to raise the capacity of these bearings and, on the other hand, to be able, when designing a new machine, to predetermine closely the performance of the bearings in order to decide whether an oil-ring type will be adequate. As a result, an investigation partly reported in the present paper has been undertaken.

The paper deals with the questions of safety of a bearing and the heat balance in oil-ring types, these questions being basic to the performance of any bearing. The author realizes that many other considerations must be taken into account when designing a bearing, such as strength, oiltightness, dustproofness, convenience of inspection and maintenance, ease of manufacture, etc. Many of these properties have been considerably improved recently[9] in line with the generally stricter demands of modern industry and the serious competition offered by ball and roller bearings.

The Oil Film in Oil-Ring Bearings

The general idea of a load-carrying oil film is now well known and does not need much space here. It will be sufficient to mention that the existence of such a film is caused by the viscosity of the lubricant and by its ability to adhere strongly to the rubbing surfaces of the shell and the journal. The radial clearance provided in the bearing automatically causes the existence of a wedge-shaped film between the journal and the shell. The oil, being entrained by the journal into the clearance, tends to be compressed; due to its practical incompressibility, strong currents and countercurrents of oil are created in the film. The rate of shear in the oil is naturally very high, the gradient of velocity from the journal surface to the shell surface being of the order, say, of 500,000 ft. per min. per inch of oil film, and the viscous resistance of the oil to this motion creates high hydrostatic pressures, quite ample to carry the load on the journal.

Under these conditions, the only frictional losses in the bearing are caused by internal friction of the lubricant. This friction is only a fraction of that which would be present should the load be carried by direct contact between the rubbing surfaces, even in the case when such surfaces are covered by a good "greasy" (or "oily") film of lubricant.

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Presented at the Annual Meeting, New York, N. Y., Dec. 2 to 6, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[8] R. O. Boswall, "The Theory of Film Lubrication." Longmans, 1928. See bibliography on p. 23. See also bibliography in "Progress in Lubrication Research," paper APM-50-4, Trans. A.S.M.E., 1928.

[9] R. Pruger, "Some Ways to Cure Sleeve Bearing Troubles." *Ind. Eng. Chem.*, March, 1924. *Coal Age*, June 26, 1924.

R. W. Schuck, "Dustproof Qualities of Sealed Sleeve Bearings." *Concrete*, December, 1927.

It is therefore important to be in a position to evaluate the minimum thickness of the oil film when designing a bearing in order to be certain that this thickness is above the safe limit for the type of bearing under consideration. Now, this limit may vary considerably, depending on the kind of service, the finish

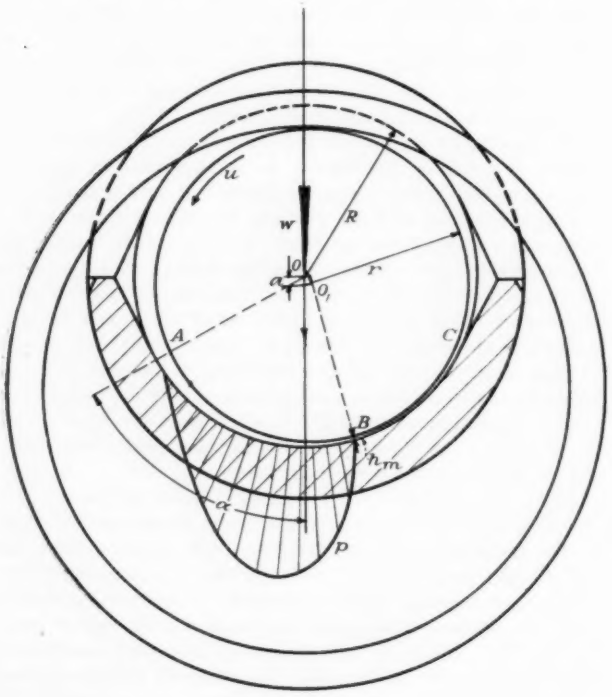


FIG. 1

of the surfaces, and probable misalignment or thermal distortion of the bearing. In finely bored small bronze bushings a film of the order of 0.0001 in. will be sufficient,[10] while for ordinary commercial bearings, with a steel shaft and a babbited shell, a thickness of oil film above 0.00075 in. should be maintained if the speeds are high, and a thickness not below 0.0004 in. in all other cases.[11] This limit was checked on several occasions and appears to be well founded. It will be shown later that the determination of the minimum thickness is a comparatively simple matter, and the safety of a bearing should be checked by a designer on this basis rather than by empirical formulas which are usually obsolete a few years after being established. For example, there is one formula giving for the safe limits of bearing design[12] $pv = 50{,}000$, where p is the pressure on the projected area in pounds per square inch and v is the peripheral speed in feet per minute. Bearings are now operating successfully where $p = 100$ and $v = 2100$, pv being above 200,000. Another formula derived by Moore[13] has been used indiscriminately; $p = 7.47\sqrt{v}$. This would give as a limiting safe load $p = 335$ lb. per sq. in., which is obviously too high. An oil-ring bearing of standard construction shown in Fig. 1 will now be considered. Let

$d = 2r$ = diameter of journal
$D = 2R$ = diameter of shell

[10] See discussion of the paper by S. A. McKee and T. R. McKee, "Friction of Journal Bearings as Influenced by Clearance and Length." Trans. A.S.M.E., 1929, paper APM-51-15.

[11] V. Vieweg, "Optische Massgeräte zur Bestimmung der Dicke der Oelschicht in Lagern . . ." *Petroleum Zeit.*, 1922, p. 1405.

[12] "Mechanical Engineers' Handbook," L. S. Marks. McGraw-Hill, New York, 1924.

[13] H. F. Moore, "Bearings." *American Machinist*, Sept. 10, 1903.

$D - d$ = clearance (play) in shell; $R - r$ = radial clearance
$m = (D - d)/d = (R - r)/r$ = clearance ratio; $R - r = mr$
h_m = minimum oil-film thickness
a = linear eccentricity between center O of shell and center O_1 of journal in running position
$c = a/(R - r)$ = eccentricity ratio; $a = (R - r)c$
p = pressure in the oil film at any point
h = the thickness of the oil film at this point
W = total load on bearing
L = active load-carrying length of bearing between end grooves
$P = W/dL$ = pressure on projected active area
α = angle between entering edge A of oil film and direction of load W. (In case of symmetrical bearings with vertical load on the journal, the bearing angle $AOC = 2\alpha$.)
u = peripheral velocity
N = revolutions per minute of the journal
$\omega = \pi N/30$ = angular velocity in radians per second
μ = absolute viscosity of the oil in the oil film, assumed to be constant throughout the film
Z = the viscosity measured in centipoises.

Assume for the time being that the oil film is sealed at the bearing ends, i.e., that there is no end leakage, making the conditions therefore identical in each cross-section of the bearing.

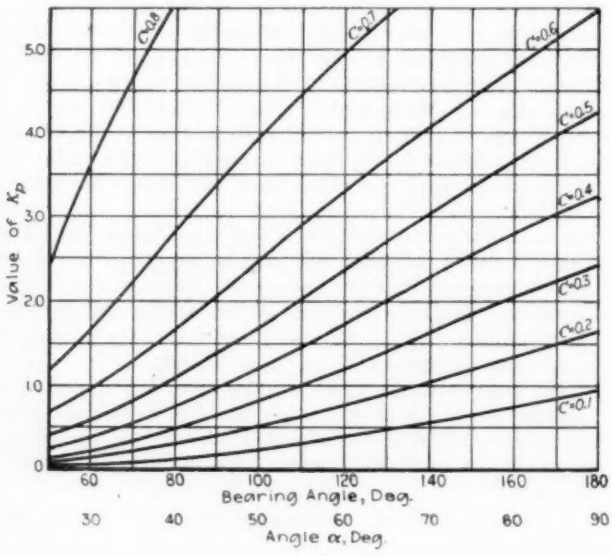


FIG. 2

The middle strip of the bearing, a unit of length (inch, centimeter) wide, carries then a load $w = W/L$. If equilibrium exists, the resultant of all oil pressures p on the journal strip is equal and opposite to w.

The center O_1 of the journal can take any position inside a small circle described around O by the radius $(R - r)$ (see Fig. 4), and the journal will automatically adjust itself to a position where this equilibrium exists.

The pressures p are, however, determined completely by the viscosity of the oil μ, the velocity u, and the shape of the clearance. These values are interconnected by the equation:[14]

[14] The derivation of this equation is given very concisely in A. Stodola's "Steam and Gas Turbines," translated by Louis C. Lowenstein. McGraw-Hill, New York.

$$\frac{dp}{dx} = -6\mu u \frac{h - h_1}{h^3}$$

where h_1 is a constant the value of which happens to be larger than h_m but smaller than h_a.

It was observed on numerous occasions that the oil pressure at B, the point of minimum oil-film thickness, is very near zero, the pressure in BC where the film is divergent being slightly negative. A vacuum up to several inches of mercury may be observed even in forced-feed bearings when the oil is supplied under a low head of, say, 10 lb. per sq. in. The assumption that the pressure p_B at B is equal to zero in oil-ring-lubricated bearings is therefore justified. On the other hand, under usual conditions of bearing construction, the pressure at A is quite low and the pressure p_A may also be taken equal to zero. It is true that pressures may be observed sometimes under the upper cap of the shell or in the relief at A, but these are usually small when compared with the pressures in the oil film proper and they do not affect appreciably the values of p.

With $p_A = p_B = 0$, the pressures p can be evaluated from the differential equation referred to for each position of O_1. Charts[15] presenting the values of p were prepared by the author in an earlier paper. The pressures p, when combined vectorially, give for each position of O_1 the value of w and its direction, i.e., the angle α between w and the entrance edge A. As a rule, the shape of the pressure curve p is such that the angle between

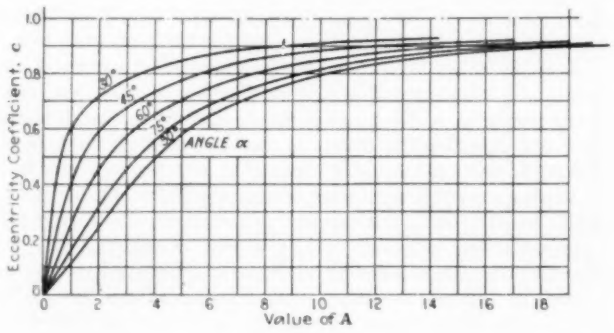


FIG. 3

w and B is smaller than α, the difference growing with the increase of the linear eccentricity a.

It will be noted from the charts that the pressures p are proportional to the value $\mu\omega/m^2$, the coefficient of proportionality being determined only by the relative position of O_1 with respect to O, while the load-carrying capacity of the strip of unit width is

$$w = K_p r \frac{\mu\omega}{m^2}$$

where K_p is also determined only by the position of the journal. By tabulating the values of w and α for a large number of positions O_1 and by graphical interpolation, the relation between K_p, α, and $c = a/(R - r)$ can be represented in a chart shown in Fig. 2.

In actual bearings, the end leakage causes a gradual decrease in oil pressure from the middle section to the ends where it is zero. Experimental data by Lasche,[16] Kingsbury, and others show that with normal ratios of length to diameter of from 1 to 3, the pressure distribution lengthwise may be taken as parabolic. The load-carrying capacity of the bearing is then:

$$W = \frac{2}{3} wL = \frac{2}{3} K_p Lr \frac{\mu\omega}{m^2}$$

Since $W = PLd = 2PLr$, $P = {}^1/_3 K_p \mu\omega/m^2$. With a given load, load direction, speed, viscosity, clearance, the coefficient $K_p = 3Pm^2/\mu\omega$ can be found, and from Fig. 2 the value of c will be determined. The linear eccentricity OO_1 is $a = c(R - r)$, and the minimum thickness of the oil film is $h_m = (R - r) - a = (1 - c)(R - r) = (1 - c)mr$.

The chart in Fig. 2 is not convenient. It has been rearranged

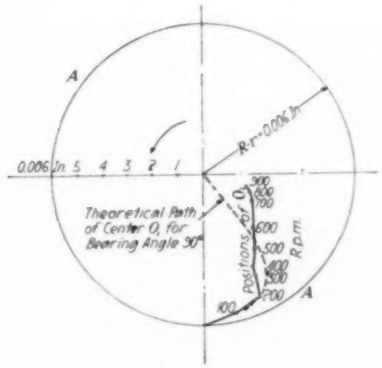


FIG. 4

and extended partly on the basis of the calculations of Guembel,[17] partly by extrapolation, and the chart in Fig. 3 was drawn. The coefficient K_p is replaced by A, allowing for the diverse units used in engineering; P is measured in pounds per square inch, and $\omega = \pi N/30$, where N is the revolutions per minute. The absolute viscosity μ should be measured in (pounds per sq. in. of rubbing area) per (inch per second per inch of film thickness) or in $\frac{\text{lb. sec.}}{\text{in.}^2}$, while it is given usually in centipoises, where 1 poise = 1 (dyne[18] per square centimeter of area) per (centimeter per second per centimeter of film thickness) $= 1 \frac{\text{dyne sec.}}{\text{cm.}^2}$; the viscosity in centipoises being Z, $\mu = 1.45 \times 10^{-7} Z$. Now

$$P = \frac{1}{3} K_p \frac{\mu\omega}{m^2}$$

$$= \frac{1}{3} A \frac{ZN}{m^2} \frac{1.45 \times 10^{-7}\pi}{30}$$

therefore

$$A = 198(1000m)^2 \frac{P}{ZN}$$

The value ZN/P, which is a non-dimensional number, may be called the characteristic number of the bearing. It not only defines the running position of the journal in the shell but determines the running friction in the bearing as well, as has been pointed out repeatedly in the literature.

As an example, a slow-speed oil-ring bearing of the following specification will be considered: diameter d = 11 in.; full length, 25 in.; end grooves occupy $1^1/_4$ in. on each end, giving an active length L = 22.5 in.; clearance on diameter s = 0.0125 in., or m = 0.0012; the relief at the split is 15 deg. on each side; load W = 35,400 lb. is directed downward; speed of

[15] G. B. Karelitz, "Charts for Studying the Oil Film in Bearings." Trans. A.S.M.E., vol. 47 (1925), p. 1101.

[16] O. Lasche-W. Kieser, "Materials and Design in Turbo-Generation Plant." English edition, Oliver & Boyd, Edinburgh, 1927.

[17] Guembel, "Einfluss der Schmierung auf die Konstruktion." Jahrbuch der Schiffbautech. Gesellschaft, J. Springer, Berlin, 1917.

[18] 1 dyne = $^1/_{981}$ gram.

rotation $N = 187.5$ r.p.m. The oil used was machine oil having a viscosity of 320 sec. Saybolt at 30 deg. cent. and 50.5 sec. Saybolt at 100 deg. cent.; specific gravity 0.86.

The pressure on the projected area of the bearing is $P = W/dL = 141.5$ lb. per sq. in. The temperature rise in the oil film may be assumed to be 20 deg. cent. above the ambient temperature; with a possible air temperature of 40 deg. cent., the temperature in the oil film will be 60 deg. cent., or $Z = 17.0$

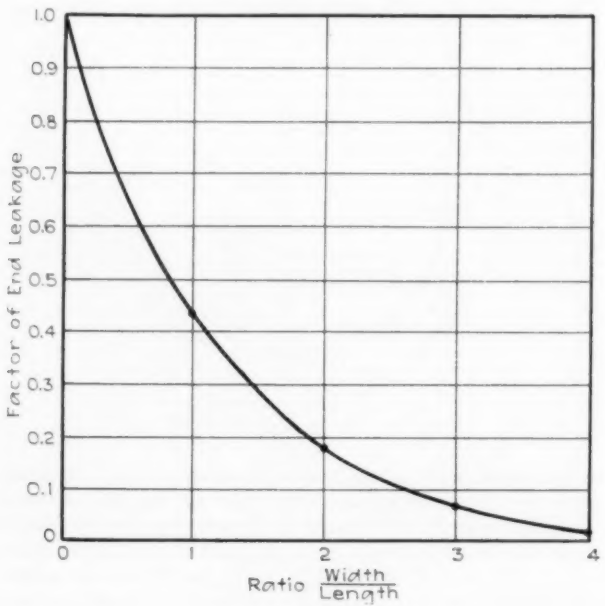


FIG. 5

centipoises (see Appendix No. 1). The coefficient $A = 198 \times 1.2^2 \times \frac{141.5}{17 \times 187.5} = 12.7$. The angle α in this case is 90 deg. — 15 deg. = 75 deg. Entering $A = 12.7$ in the chart (Fig. 3), the curve $\alpha = 75$ deg. gives $c = 0.86$. The minimum oil-film thickness is then $h_m = (1 - c)(R - r) = 0.14 \times \frac{0.0125}{2}$ = 0.0009 in. This bearing works therefore within safe limits.

Observations on actual bearings seem to confirm the accuracy of this computation of the minimum thickness of the oil film. It is true that a report to the contrary was published recently by Wolff,[19] but very accurate electromagnetic measurements of journal positions carried out on an oil-ring bearing of 9 in. diameter and 18 in. long with a 45-deg. relief at the split on each side gave an oil-film thickness very close to the precomputed values.[20] The bearing was loaded by the constant weight of the armature of a synchronous motor, while the speed was varied from 100 to 900 r.p.m. The oil-film thickness was found to be as in Table 1.

The actual path of the journal center O_1 did not coincide, however, with the theoretical path as determined on the basis of the charts for the oil pressure, mentioned above. Fig. 4 shows the discrepancy. The circle AA of radius $(R - r)$ encloses the area of all possible positions of the journal center O_1. The journal appears to be pressed to the right from its theoretical path, no doubt due to the initial pressure in the relief pocket on the "down" side of the journal. It will be appreciated that a small pressure of several pounds per square inch in the pocket produces an appreciable horizontal force on the journal.

TABLE 1

R.p.m.	Min. oil-film thickness, in. Computed	Observed
100	0.0007	0.0005
200	0.0012	0.0010
300	0.0016	0.0015
400	0.0019	0.0020
500	0.0023	0.0025
600	0.0025	0.0029
700	0.0028	0.0030
800	0.0030	0.0031
900	0.0031	0.0031

It is obvious that oil-ring-lubricated bearings work in the best conditions when the load applied to the journal is directed downward or at a small angle to the vertical. Should the load be directed upward, as may happen with an industrial motor with a belt or gear drive, the slot or slots through which the oil rings pass divide the bearing into two or more parts. In fact, the shaft may be considered as supported by several separate bearings. In a bearing of a length equal to two diameters, or longer, with one oil ring at midlength of the bearing, the top half presents two bearings each one diameter long. Such bearings work quite efficiently, and the performance of the bearing as a whole is practically identical in any direction, provided oil-distributing grooves are cut properly. In shorter bearings, the oil film will be inherently thinner with loads acting upward than when the same loads act downward upon the journal. This fact must be remembered when rating oil-ring bearings for general service.

A quantitative idea about the influence of the ratio of the length of a load-carrying oil film (in direction parallel to motion) to its width (in a direction normal to motion) can be obtained from Fig. 5, which was drawn on the basis of Michell's paper.[21] Taking as unity the carrying capacity in pounds per square inch of bearing area of a very wide bearing with a ratio of length to width equal to zero, the capacity of a square plate is only 0.4, approximately. Although the pressure distribution in the bearing clearance is different from that in the plane bearing considered by Michell, the order of the influence of end leakage on the bearing performance will be the same in both instances, and the curve in Fig. 5 is useful when necessarily short bearings are designed.

MECHANISM OF OIL-RING OPERATION

All previous considerations were made with the assumption that the oil film in the bearing is not starved, i.e., the oil supply to the bearing is quite sufficient to compensate for the loss of lubricant due to end leakage and that the design of the bearing permits free axial distribution of the oil to the load-carrying oil film. Should this not be the case, the oil film will naturally become thinner and the running position of the journal, i.e., the minimum oil-film thickness, will depend on the amount of oil supplied to the bearing. Obviously such a situation is undesirable.

The end leakage can be estimated by the formula (see Appendix No. 2)

$$Q = 0.00022\, k\, \frac{P}{Z}\, \frac{d}{L} \left(1000 s\right)^3$$

where Q = oil loss through end leakage in gal. per min.
P = pressure on the projected area in lb. per sq. in.
Z = viscosity of oil in the oil film in centipoises
d = diameter of the journal, in.
L = active length of bearing, in.
s = clearance on diameter, in.

[19] R. Wolff, "Ueber die Schmierschicht Messung durch Interferenz." Forschungsarbeiten, No. 308, V.D.I., Berlin, 1928.

[20] These tests were made by M. Stone, Power Engineering Department, Westinghouse Elec. & Mfg. Co., as a part of a bearing vibration study and will be published in detail at a later date.

[21] A. G. M. Michell, loc. cit.

The coefficient k depends on c and α, as shown in Fig. 6. In order to form a numerical idea of the oil demand, the end leakage has been computed for a series of bearings of similar dimensions, taking for all of them P = 100 lb. per sq. in.; Z = 15 centipoises; symmetrical bearings with a downward load, the relief at the split being 30 deg. on each side, i.e., the angle α = 60 deg. Under these conditions, the end leakage is estimated for various speeds of the journal. The results are given in Table 2.

The amounts of oil required for the maintenance of a perfect oil film are not large; however, in order to have a margin of safety, the oil supply should be, say, twice the oil loss through end leakage. The capacity of standard oil rings has been investigated by the author. Incidentally a number of observations have been made on the mechanism of performance of oil rings.

Preliminary tests were carried out on a disk 4 in. in diameter

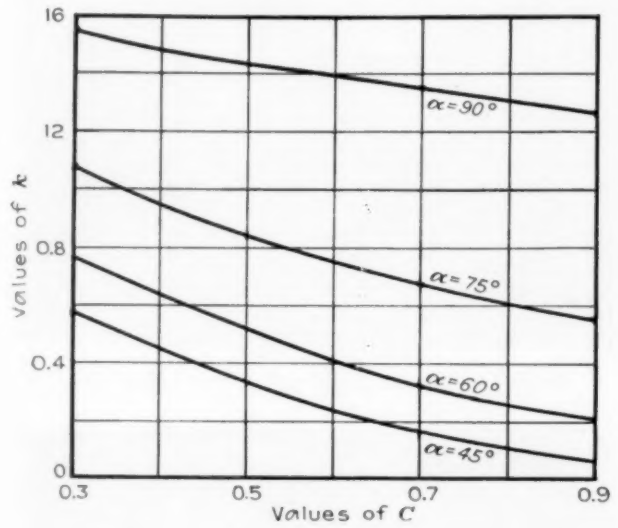


FIG. 6

carrying an oil ring used with this size of bearing. The oil ring was of 7 in. inside diameter, $^3/_8$ in. wide, and was made of brass. The arrangement is seen in Figs. 7 and 8. The ring was kept in place by a guide placed at its lower point so that nothing interfered with the transfer of oil by the ring. The speed of the motor was 1760 r.p.m. Medium machine oil was used for the test. The amount of submersion of the oil ring in the oil was varied and the corresponding speed of the oil ring measured. When the oil ring just dipped into the oil, and practically no oil was carried along, the ring raced on the journal with a metallic sound. At submersion of $^1/_8$ in. and upward an oil film was built up between the ring and the journal and no metallic sound could be heard. The breaking of the metal-to-metal contact was checked by a test light and voltmeter.

The speeds of the oil ring at different amounts of submersion, with oil at room temperature and at 53 deg. cent., are given in Fig. 9. As the temperature of the oil was raised, the oil ring slowed down, due to the fact that the viscosity of the hot oil decreased, the friction force exerted by the oil film on the oil ring decreasing correspondingly.

Figs. 7 and 8 show clearly the mechanism of operation of oil rings on high-speed journals. A heavy oil film was lifted on the "up" side of the oil ring. This film was very stable; when some solid object was put in its way, the film went around the object and did not seem to diminish in body. On the upper part of the ring, most of the oil was carried on the outer surface,

TABLE 2 END LEAKAGE IN GALLONS PER MINUTE

Value of d, in.		4	8	12	16
Nominal length,[a] in.		8	16	24	32
Value of s, in.		0.007	0.0095	0.012	0.016
Value of m		0.00175	0.0012	0.0010	0.0009
For N = 200 r.p.m.	c	.....	0.84	0.76	0.72
	Q	.....	0.17	0.40	0.75
For N = 400 r.p.m.	c	0.85	0.69	0.60	0.54
	Q	0.07	0.24	0.57	1.12
For N = 600 r.p.m.	c	0.77	0.58	0.47	0.42
	Q	0.08	0.30	0.77	1.46
For N = 1000 r.p.m.	c	0.65	0.44	0.33	....
	Q	0.10	0.41	0.98	....
For N = 1500 r.p.m.	c	0.46	....	....	....
	Q	0.13	....	....	....

[a] The ratio d/L is taken in all these bearings to be 1/1.8.

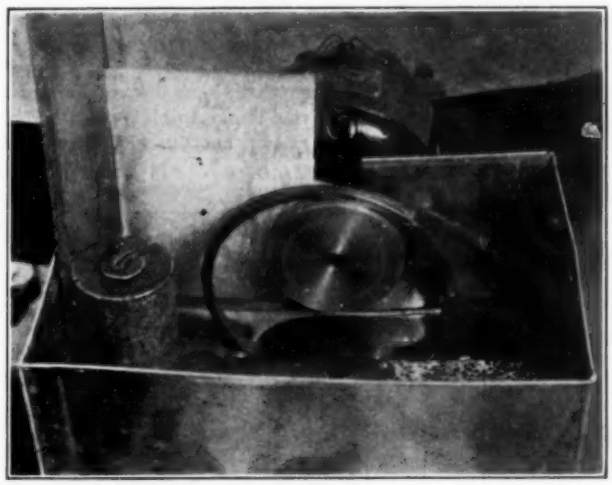


FIG. 7 OPERATION OF 7-IN. OIL RING, 1-IN. IMMERSION

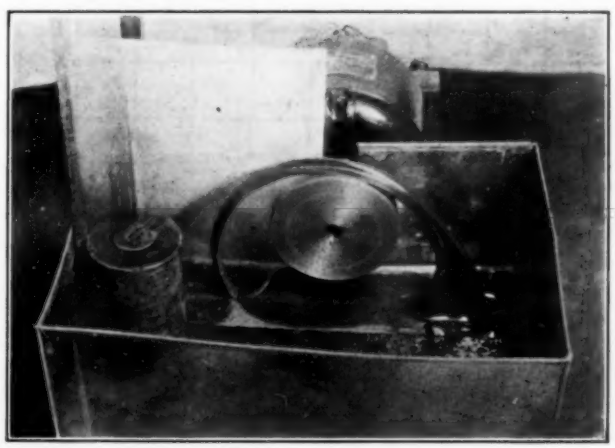


FIG. 8 OPERATION OF 7-IN. OIL RING, 2-IN. IMMERSION

forced there by the centrifugal force. The oil carried on the inner surface formed the oil film on which the oil ring was riding. The oil was squeezed out from underneath the ring by the pressure in the oil film, and coming in contact with the surface of the journal, was thrown off, so that two sprays of oil were built up on both sides of the ring. A very thin oil film was formed between the down side of the journal and the inner surface of the ring on this side. The oil from the outer surface and sides of the oil ring fell in a rainy spray from the down side of the ring into the oil bath. Fig. 7 shows the ring submerged 1 in., while the submersion in Fig. 8 is 2 in.; at submersions below 1 in. the oil film was thrown over the top of the ring. At higher

temperatures all oil films were considerably thinner, the viscosity of the oil being lower. A scoop was used to collect the oil delivered by the oil film on the down side, as shown in Fig. 10. The amount of oil collected varied with the distance from the knife edge of the scoop to the journal. It was measured at submersions of 1 in. and 1⅝ in. The amounts collected were in cubic centimeters per minute (1 gal. = 3785 cc.), as shown in Table 3.

The amount of oil delivered by the spray was obtained for the submersion of 1⅝ in., at different temperatures, by collecting the spray into a separate container. The rate of oil flow was:

Temperature of oil, deg. cent.....	23	34	43	51
Oil delivered by spray, cc. per min..	618	616	421	288

The amount of oil adhering directly to the journal was determined by applying a wiper to it and was found to be approximately 30 cc. per min. for oil at 22 deg. cent. and 60 cc. per min. for oil at 50 deg. cent.

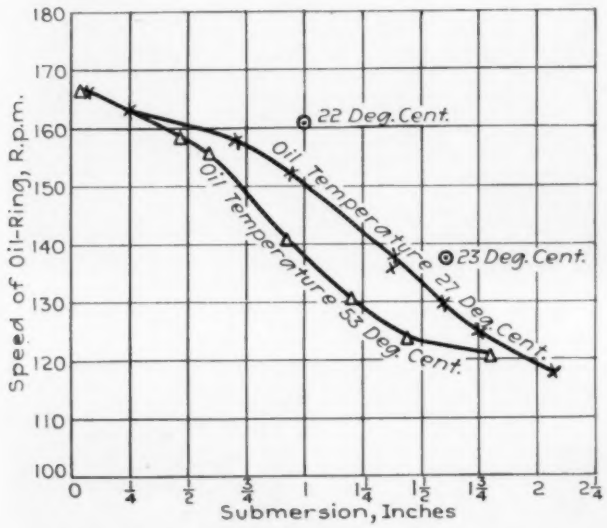


FIG. 9 SPEEDS OF OIL RING AT DIFFERENT IMMERSIONS
(Speed of disk, 1760 r.p.m.; diameter of disk, 4 in.; diameter of ring, 7 in.)

TABLE 3

Distance from journal to edge of scoop, in.	Amount of oil collected, cc. per min. — Submersion 1 in. Temp. 22 deg. cent. Speed, 161 r.p.m.	Submersion 1⅝ in. Temp. 23 deg. cent. Speed, 138 r.p.m.	Submersion 1⅝ in. Temp 50 deg. cent. Speed 119 r.p.m.
⅛	17.5	18	19
¼	30.4	36	44
⅜	63	56	65
½	87	66	72
⅝	94	80	79
¾	118	94	83
⅞	125	120	82
1	127	113	..
1⅛	134	117	..

Complete tests were made with a large oil ring and a 16-in. journal. The arrangement of the test apparatus is shown in Fig. 11. The disk 16 in. in diameter, 2 in. thick, was mounted on the shaft of a variable-speed motor. A brass oil ring of 27 in. inside diameter, 0.762 in. wide, and with a cross-section as shown in Fig. 12 was employed, the weight of the ring being 16.15 lb. The clearance between the lowest point of the oil ring and the bottom of the oil bath was 1¾ in. Three guides held the ring in position; two of these are seen in Fig. 11, while the third was at the bottom of the ring in the oil. Heaters placed in the oil allowed the temperature to be varied. Light machine oil of a viscosity approximately 300 sec. Saybolt at 30 deg. cent. was used for these tests.

In Fig. 11 the ring is shown running counterclockwise. The oil ring carried on its up side a heavy oil film in which the oil was in an eddy. The oil from the sides of the ring was shifted to the outer surface by the centrifugal force and formed a crown on the outer surface. This oil was carried over the top of the ring and back into the oil well. The oil on the inner surface formed an oil film on which the ring was riding. As in the case of the small ring the break of contact was verified by a test lamp. The oil from the inner surface was partly squeezed out by the weight of the ring forming two sprays on either side of the ring, the other part passing by the journal and forming a



FIG. 10 METHOD OF COLLECTING OIL IN OIL-RING TEST

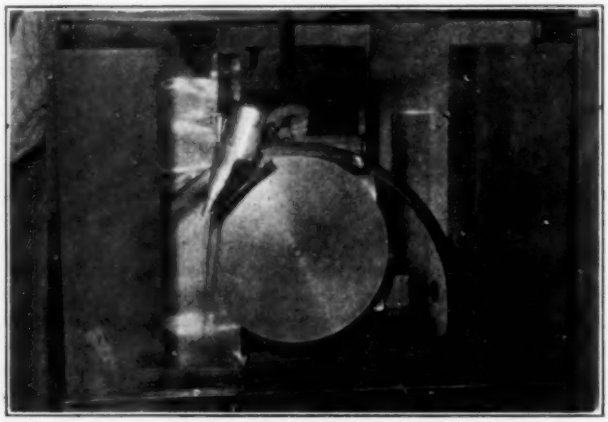


FIG. 11 TESTING OF 27-IN. OIL RING ON 16-IN. DISK

film between the journal and the ring on the down side. It must be noted that the spray appeared only after a certain speed of the journal was attained. On the other hand, the down-side film was heavy at low speed, nearly disappearing as the speed of the journal reached 300 r.p.m., when the oil was thrown off effectively from the journal surface by centrifugal force.

At low speeds, the oil ring followed the journal, their peripheral velocity being equal. The revolutions per minute of the journal and the ring were at this stage inversely proportional to their diameters, with but a slight slippage. As the revolutions of the journal increased, a transition speed was reached when the oil film between the journal and oil ring became sufficiently strong to carry the weight of the oil ring. A considerable slipping

between the journal and the ring took place and the oil ring slowed down considerably. Below the transition speed the test light connected across the ring and journal burned brightly with occasional flickering. This flickering increased when the transition speed was approached, and the light was dimmed at higher speeds. Fig. 12 shows the change of speed of the oil ring with increase in revolutions of the journal for various amounts of submersion, the oil being at room temperature. Fig. 13 gives the same data with oil heated to 50 deg. cent. The transition speed was considerably higher at the elevated temperature, corresponding to a lower viscosity of the oil.

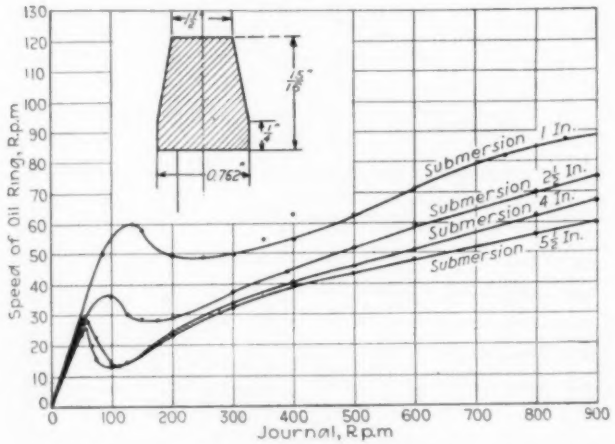


FIG. 12 CHANGE OF SPEED OF OIL RING WITH INCREASE IN REVOLUTIONS OF JOURNAL

(Diameter of journal, 16 in.; diameter of ring, 27 in.; weight of ring, 16.15 lb.; temperature of oil, 23 to 27 deg. cent.)

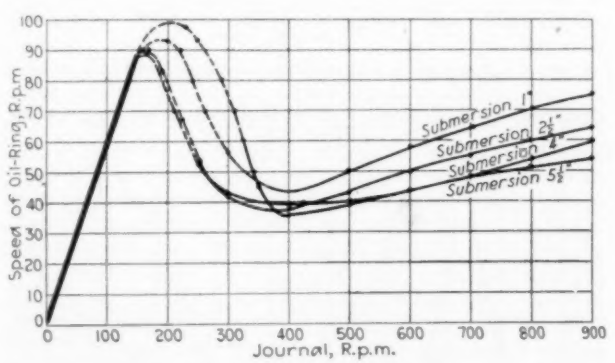


FIG. 13 CHANGE OF SPEED OF OIL RING WITH INCREASE IN REVOLUTIONS OF JOURNAL

(Diameter of journal, 16 in.; diameter of oil ring, 27 in.; width of oil ring, 0.762 in.; weight of oil ring, 16.15 lb.; temperature of oil, 48 to 55 deg. cent.)

The amounts of oil delivered by the oil ring were measured by collecting the oil from the sprays through the funnels shown in Fig. 11. A scoop, similar to the one used with the small oil ring described above, was employed to measure the amount of oil available in the oil film on the down side. It was found that in the case of the large ring this oil film thinned out very fast toward the ring; the scoop edge was therefore kept at a constant distance $^3/_4$ in. from the journal surface. Figs. 14 and 15 show the amounts of oil delivered by the sprays and oil film at various amounts of submersion for cold and hot oil, respectively.

In order to investigate the effect of weight upon the performance of an oil ring, the tests described above were repeated with oil rings of the same geometrical dimensions made of aluminum and of reinforced lead. The weights of the three rings were:

Material.....	Aluminum	Brass	Lead
Weight, lb.....	4.56	16.15	23.2

Figs. 16 and 17 show, respectively, the oil-ring speeds and amounts of oil delivered for the three rings in the case of a submersion of 4 in. and an oil temperature 55 deg. cent., this case being the closest to actual service conditions. As may be expected, the transition speeds were higher with heavier rings, and more oil was delivered in the sprays produced by these rings. It is therefore clear that light rings, sometimes advocated by designers, have no justification. Naturally, although heavier rings were shown to be better, the use of lead rings would be impractical.

The influence of the width of oil rings was studied in the same way; three brass rings of the same shape but of various widths were prepared, as shown in Fig. 18.

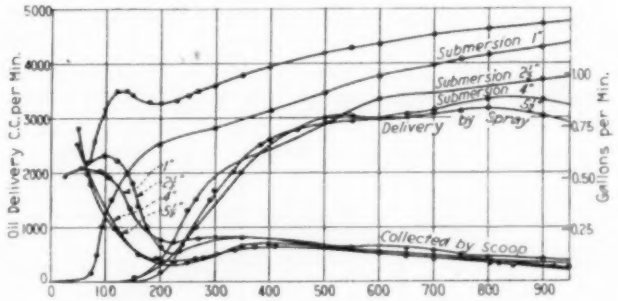


FIG. 14 AMOUNT OF OIL DELIVERED BY SPRAYS AND OIL FILM

(Diameter of journal, 16 in.; diameter of oil ring, 27 in.; weight of oil ring, 16.15 lb.; temperature of oil, 23 to 27 deg. cent.)

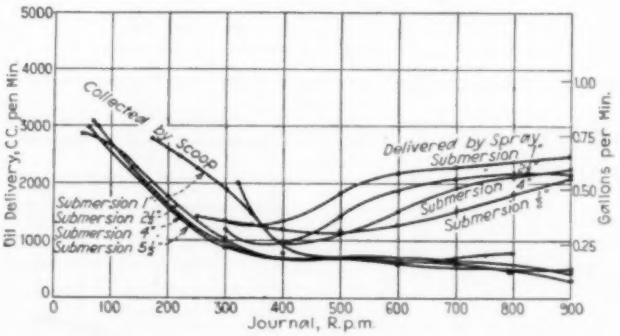


FIG. 15 AMOUNT OF OIL DELIVERED BY SPRAYS AND OIL FILM

(Diameter of journal, 16 in.; diameter of oil ring, 27 in.; weight of oil ring, 16.15 lb.; temperature of oil, 45 to 52 deg. cent.)

The ring speeds and amounts of oil delivered by the oil rings at a submersion of 4 in. and with oil heated to 55 deg. cent. are shown in Figs. 18 and 19. The speeds, at higher revolutions of the journal, are approximately the same, but the amounts of oil appear to be roughly proportional to the width of the rings. This result is important, giving a simple method to increase the oil delivery from an oil ring, if so desired.

In high-speed machines very little oil can be obtained from the oil film on the down side of the journal, most of the oil being delivered by the oil ring to the sprays. It is therefore necessary to provide means for conducting this oil into the shell and preventing it from flowing directly back into the oil well. The recent designs of larger oil-ring bearings produced by the Westinghouse Company are provided with special grooves which perform this duty.

Figs. 20 and 21 give the oil-ring speeds for an oil ring of 17 in. inside diameter mounted on a 10-in. journal. The surface of the ring was slightly rough, which was responsible for the large variation in the transition speed. Fig. 22 shows the amounts of oil carried in the oil sprays of this ring at room temperature.

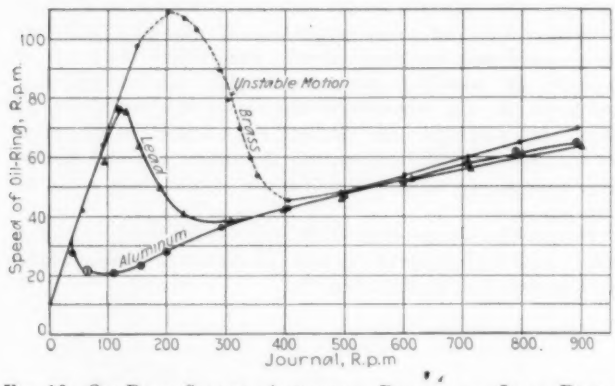


FIG. 16 OIL-RING SPEEDS, ALUMINUM, BRASS, AND LEAD RINGS
(Diameter of journal, 16 in.; diameter of oil ring, 27 in.; depth of submersion, 4 in.; temperature of oil, 55 deg. cent.)

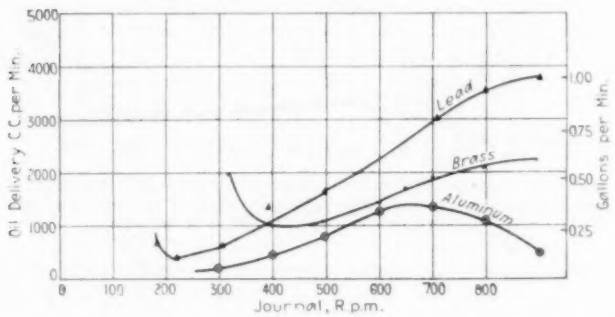


FIG. 17 AMOUNT OF OIL DELIVERED, ALUMINUM, BRASS, AND LEAD RINGS
(Diameter of journal, 16 in.; diameter of oil ring, 27 in.; width of oil ring, 3/4 in.; depth of submersion, 4 in.; temperature of oil, 55 deg. cent.)

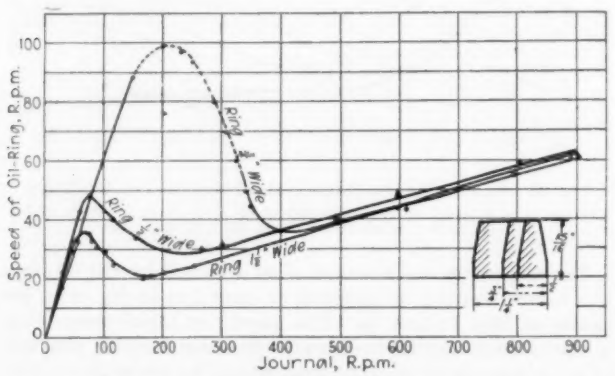


FIG. 18 OIL-RING SPEEDS FOR RINGS OF DIFFERENT WIDTHS
(Diameter of journal, 16 in.; diameter of all oil rings, 27 in.; weight of oil ring 1/2 in. wide, 12.3 lb.; weight of oil ring 3/4 in. wide, 16.3 lb.; weight of oil ring 1 1/4 in. wide, 27.9 lb.; depth of submersion, 4 in.; temperature of oil, 55 deg. cent.)

FRICTIONAL LOSSES AND TEMPERATURES IN OIL-RING BEARINGS

While the running positions of the journal, i.e., the minimum oil-film thickness, can be computed with a fair amount of accuracy, the friction losses and the temperature of a bearing can be estimated only as a first approximation. The determination of the running temperature of a bearing involves the consideration of the thermal equilibrium of the bearing pedestal—the amount of heat created by the frictional losses must be equal to the heat carried away by direct dissipation through the pedestal surface and by the cooling liquid, when artificial cooling is used.

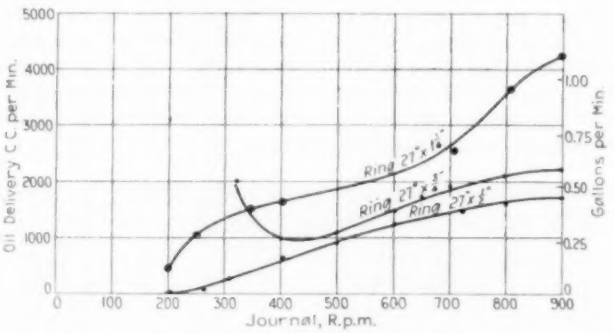


FIG. 19 AMOUNT OF OIL DELIVERED BY RINGS OF DIFFERENT WIDTHS
(See note under Fig. 18.)

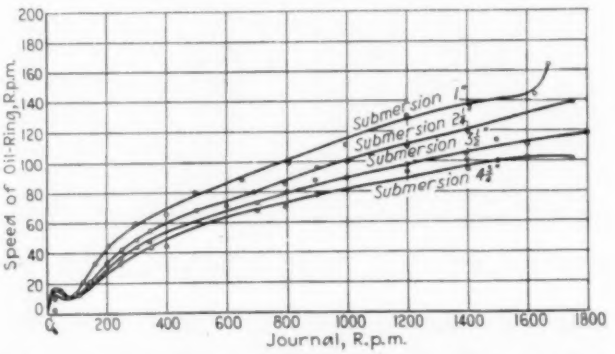


FIG. 20 OIL-RING SPEEDS WITH TEMPERATURE OF OIL AT 25 DEG. CENT.
(Diameter of journal, 10 in.; diameter of oil ring, 17 in.; weight of oil ring, 4.15 lb.)

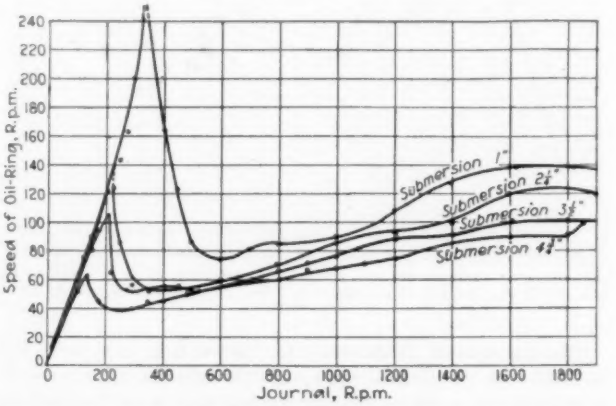


FIG. 21 OIL-RING SPEEDS WITH TEMPERATURE OF OIL AT 50 TO 55 DEG. CENT.
(See note under Fig. 20.)

It is customary to measure the frictional losses in a bearing in terms of its coefficient of friction. This is a purely fictitious conception in so far as usual bearings are concerned, since only the friction torque T of the journal can be measured. The coefficient of friction f must be defined as the ratio of the fictitious friction force $F = T/r$ acting on the journal surface to the total load W on the journal:

$$f = \frac{F}{W} = \frac{T}{rW}$$

The value of f depends on the design of the bearing, the size of side reliefs, the shape and type of the upper bearing half, the type of end seals used to prevent oil leakage from the pedestal, and on the misalignment of the shell.

Limiting the discussion to oil-ring bearings of good mechanical design with top and bottom half bored to the same diameter D, without any tight felt or other packing to prevent leakage (these at times considerably increase friction losses), it may be assumed that the coefficient of friction in such bearings is equal to the

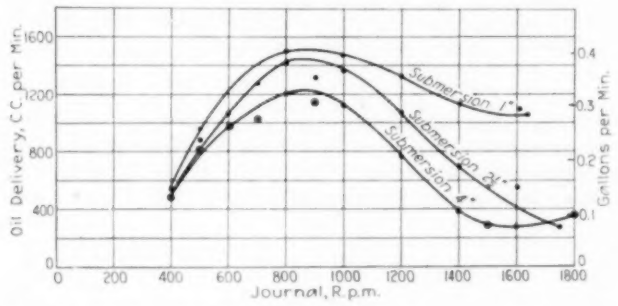


FIG. 22 AMOUNT OF OIL CARRIED IN OIL SPRAYS AT DIFFERENT DEPTHS OF SUBMERSION

(Diameter of journal, 10 in.; diameter of oil ring, 17 in.; temperature of oil, 25 deg. cent.)

coefficient of friction of a tubular bearing of the same size and clearance.

The power necessary to drive the oil rings proper is quite small. This was determined for the 27-in. × $^3/_4$-in. oil ring previously mentioned. It was found to be nearly proportional to the speed of the journal with a slight drop at the transition speed, reaching 115 watts at 900 r.p.m. The relief at the split tends to decrease the losses, but this effect is compensated by the additional friction at the ends of the bearing, in the end grooves, and on the oil throwers. It is therefore justifiable to assume that the losses in an oil-ring bearing may be computed considering it as a tubular shell of the length L, i.e., the active length of the bearing. Very little work has been done on the friction in actual oil-ring bearings, while experimental and theoretical determinations of friction in tubular bearings are available.

The theoretical values of the coefficient of friction in tubular bearings can be found in the literature. Charts for its numerical value are supplied, for instance, in the work of Howarth.[22] A recent experimental determination of the coefficient of friction in tubular bearings was made by McKee.[23] These experiments were made on four small bearings mounted simultaneously on one shaft and may therefore have the virtue of an automatic averaging of results. On the basis of these experiments, the chart in Fig. 23 gives the coefficient of friction as a function of the characteristic number of the bearing ZN/P for different clearance ratios m. The coefficient of friction is seen to increase with the speed of rotation N and with the viscosity Z and to decrease with an increase in the pressure P on the journal. The coefficient depends also considerably on the bearing clearance m.

The coefficient of friction f having been determined, the power losses in the bearings are

$$H = \frac{fWV}{33{,}000} \text{ hp.}$$

$$= \frac{fWV}{44{,}300} \text{ kw.}$$

$$= \frac{fWV}{46{,}700} \text{ B.t.u. per sec.}$$

where V is the peripheral speed of the journal in ft. per min.

The temperatures throughout the pedestal and bearing are such that when a steady state is reached in three to six hours after starting a machine, equilibrium exists between the heat generated in the bearing shell and that lost through radiation, convection, and conduction.

It can be easily shown that conduction generally plays a small part in the heat transfer. The conduction of heat through steel is 0.9 to 1.1 watts per sq. in., and for cast iron, 1.0 watt

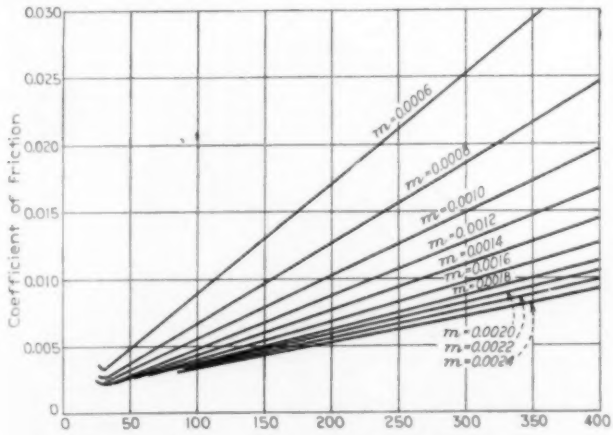


FIG. 23 JOURNAL FRICTION OF SLEEVE BEARINGS

$$\left(m = \frac{\text{clearance}}{\text{diameter}}\right)$$

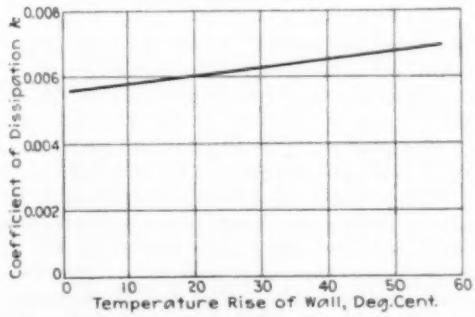


FIG. 24 VARIATION OF COEFFICIENT k WITH TEMPERATURE RISE

per sq. in., for a temperature gradient of 1 deg. cent. per in. The temperature gradient is ordinarily not large, of an order, say, of 2 deg. cent. per in. along the shaft. (It may be larger along a shaft between a bearing and a coupling rotating in open air.) The conduction of heat to the foundation depends on the height of the pedestal. It is negligible in case of a pedestal electrically insulated from the foundation. It is therefore advisable for general design purposes to neglect the heat conduction entirely, assuming that the heat is lost only through dissipation from the pedestal surface and through artificial cooling. The amount of heat dissipated from a hot surface in air depends on many factors, such as the color and finish of the surface, the temperature of the pedestal wall, the angle of the surface to the horizon, and the air currents around the surface. The velocity of the air adjacent to the surface has a remarkable influence on the heat transfer.

[22] H. A. S. Howarth, "A Graphical Analysis of Journal-Bearing Lubrication." Trans. A.S.M.E., 1923, 1924, 1925.

[23] S. A. McKee and T. R. McKee, loc. cit.

For a smooth metallic surface, such as sheet metal, the dissipation of heat in quiet air, as determined in the Westinghouse Laboratories by the late G. E. Luke, with plates and large horizontal pipes, is approximately $k = 0.006$ watt per sq. in. per 1 deg. cent. of temperature rise of the surface over the ambient temperature. This coefficient varies slightly with the temperature rise as shown in Fig. 24.

In order to determine the heat dissipation from actual bearings, a standard pedestal with a 4-in. × 8-in. bearing shown in Fig. 25 was employed. The pedestal had a dull black finish. Its outer surface was 1980 sq. in.

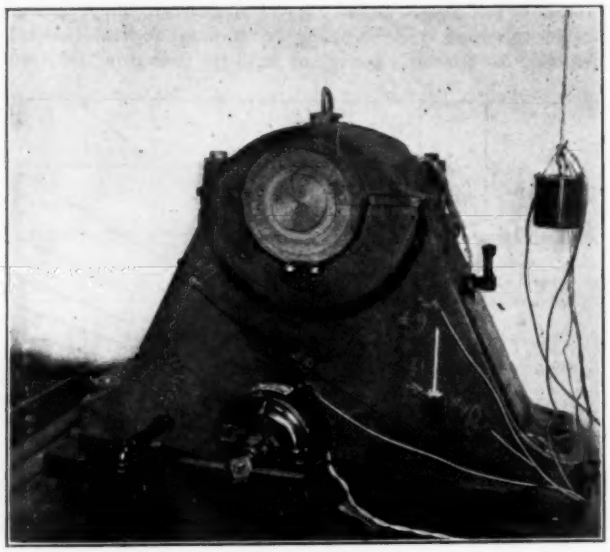


FIG. 25 APPARATUS FOR DETERMINING HEAT DISSIPATED FROM ACTUAL BEARING

A dummy shaft was placed in the bearing shell. The oil well was filled with oil. Two electric heaters were placed in the oil and a third cartridge heater was mounted in the shaft in order to produce temperature gradients in the pedestal comparable to those met in practice. The temperatures were measured by means of thermocouples welded to the surface of the pedestal. As may be seen from Fig. 25, the thermocouple leads were held flat against this surface for a length of one inch, approximately, in order to be sure that the temperatures of the hot junctions were equal to the temperatures of the surface. The temperature of the dummy shaft was also recorded by a thermocouple placed in a hole drilled through the bottom half of the shell. The observations of the temperature rise in the pedestal for various inputs into the heaters were carried out in quiet air and under the blast of two 16-in. electric fans. The average velocity of the air near the pedestal walls was approximately 500 ft. per min. Figs. 26 and 27 show the temperature rise and the coefficients of radiation observed with various inputs of power into the heaters. The coefficient of heat dissipation $k = W/St$, where W = the input in watts, S = the area of the pedestal in square inches, and t = the average temperature rise of the pedestal wall in degrees centigrade, was found to be

$k = 0.0075$ in quiet air
$k = 0.0210$ in a draft of velocity of 500 ft. per min.

The computation of power losses using the coefficients of friction from Fig. 23 was applied to a number of actual machines tested in the Westinghouse Works. The bearings of these were of the oil-ring type, without artificial cooling, i.e., the heat created by bearing losses was carried off by direct dissipation through the pedestals. The temperature rise of the walls was measured. The coefficients of dissipation computed are shown in Fig. 28. They fall well in line with the values determined in the laboratory.

It must be noted that the power losses in the oil film depend on the viscosity, hence on the temperature of the oil film, while the heat dissipation is determined by the temperature of the wall. The temperature of the oil film is naturally higher than that of the wall, since heat flows from the oil film to the wall. Observations on numerous bearings allow the compilation of the following approximate values:

Temperature rise of the pedestal wall, deg. cent	12	18	24	30
Average temperature of oil film above temperature of wall, deg. cent	18	25	32	40

With these data, an estimate of the temperature of a self-cooled bearing can be made when a new machine is designed. The procedure is as follows:

Assume a reasonable temperature rise of the pedestal wall.

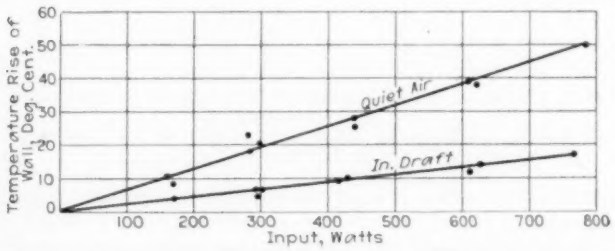


FIG. 26 TEMPERATURE RISE IN 4-IN. × 8-IN. BEARING PEDESTAL WITH 1980 SQ. IN. OF SURFACE

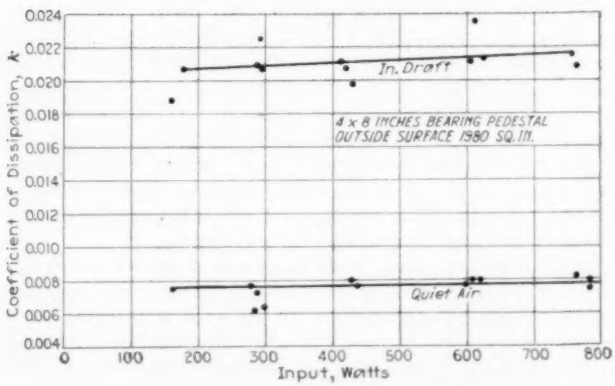


FIG. 27 VARIATION OF COEFFICIENT OF DISSIPATION k
(4-in. × 8-in. bearing pedestal, 1980 sq. ft. of surface.)

Take the lowest possible ambient temperature, since the losses and consequently the temperature rise will be the highest when the temperatures are lower and the viscosity of the oil higher. Find the temperature of the wall, then the probable average oil-film temperature, and the corresponding value of viscosity Z. Find ZN/P and the value of f. Find the losses H in watts. Knowing the radiating area of the pedestal S, take a suitable value of k, and find the temperature rise of the pedestal wall $t = H/kS$. Compare with the assumed value. If the values do not check within sufficient accuracy, make another trial. Two or three trials generally will be sufficient.

It should be noted that the radiating area S of the pedestal is rather indeterminate. In tall pedestals, it is convenient to assume the radiating area extending, say, 4 in. below the bottom

of the oil well. In short pedestals with electric insulation, the total area above the bedplate must be taken. A designer must use his judgment in all instances.

Conclusion

The author realizes that the discussion brought forth in the above notes may be criticized because of a number of assumptions and rough approximations. Unfortunately, all mechanical engineering consists, when applied to actual designing, in an approximate application of "exact science." Exact analysis is a powerful tool, but its application must always be controlled by sound judgment of the designer. This personal element may be eliminated only by an accumulation of facts and observations which will gradually narrow the range of the correction coefficients employed in the reported computations.

Appendix No. 1

COMMERCIALLY, the viscosity of an oil is given in this country in "seconds Saybolt," this being the time necessary for 60 cc. of the oil at a given temperature to flow out of a standard Saybolt viscosimeter.[24] An empirical chart with logarithmic coordinates, prepared by the Texas Company, makes it possible to plot the temperature-viscosity curve as a straight line for usual petroleum oils. Given viscosities at two temperatures, the viscosities at any other temperature are easily found by the use of this chart. To the right, a scale for kinematic viscosity is plotted. The absolute viscosity is the kinematic viscosity times the specific gravity of the oil.

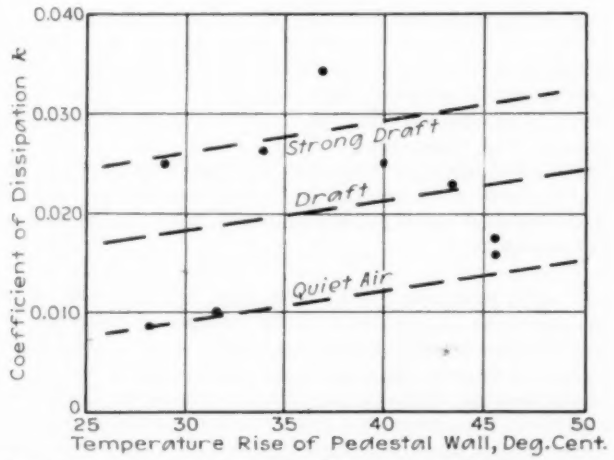


FIG. 28 VARIATION OF COMPUTED COEFFICIENT OF DISSIPATION FOR ACTUAL MACHINES

For the machine oil mentioned in this paper, at 60 deg. cent. the viscosity is found to be 102 sec. Saybolt, or a kinematic viscosity of 19.8; the absolute viscosity is 19.8 × 0.86 = 17.0 centipoises.

Appendix No. 2—End Leakage in Oil-Ring Bearings

AN EXACT determination of the oil flow on basis of the hydrodynamical theory would be quite difficult. A rough estimate of this quantity can be made, however, as follows. In so far as the axial flow is concerned, the surfaces of the journal and shell are at rest. In this case, the volume of viscous liquid passing through both ends of the bearing in a unit of time is:

$$Q = 2\,\frac{b}{12}\,\frac{dp}{dZ}\,\frac{h_0^3}{\mu}$$

where b = the width of the oil stream
$\frac{dp}{dZ}$ = the average pressure gradient at the bearing end
h_0 = the average oil-film thickness
μ = the absolute viscosity.

This equation is obtained easily from the corresponding equation given by Stodola[25] when $\mu = 0$. The value of b depends on the values of the angle of α and coefficient c. The angle of the pressure zone varies between 1.5α and 2α;[26] assume it to be 1.8α, then $b = 1.8r\alpha$. Considering the geometry of the oil film,

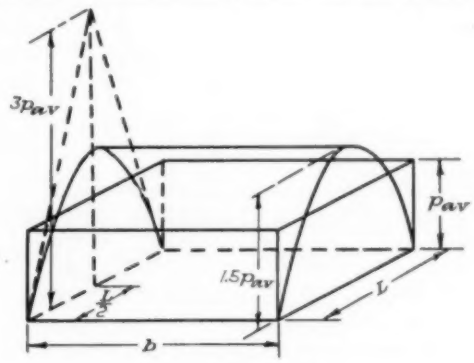


FIG. 29

$$h_0 = \frac{1}{1.8\alpha}\int h\,d\theta = \frac{1}{1.8\alpha}\int_0^{1.8\alpha} mr(1-\cos\theta)\,d\theta$$

$$= mr\left(1 - c\,\frac{\sin 1.8\alpha}{1.8\alpha}\right)$$

The average gradient of pressure at the bearing end may be estimated as in Fig. 29

$$\frac{dp}{dZ} = 6\,\frac{p_{av}}{L}$$

where p_{av} is the mean pressure in the oil film and L is the active length of the bearing. Collecting,

$$Q = \frac{1.8r\alpha}{6\mu}\,6\,\frac{p_{av}}{L}\,m^3r^3\left(1 - c\,\frac{\sin 1.8\alpha}{1.8\alpha}\right)^3$$

$$= \frac{0.9\,d}{\mu\,L}\,p_{av}\,(R-r)^3\,\alpha\left(1 - c\,\frac{\sin 1.8\alpha}{1.8\alpha}\right)^3$$

$$= k_1\,\frac{p_{av}}{\mu}\,\frac{d}{L}\,(R-r)^3,$$

where $k_1 = 0.9\,\alpha\left(1 - c\,\frac{\sin 1.8\alpha}{1.8\alpha}\right)^3$

The values of k_1 are as follows:

	$c = 0.3$	$c = 0.5$	$c = 0.7$	$c = 0.9$
45 deg.....	0.34	0.20	0.09	0.035
60 deg.....	0.58	0.40	0.25	0.16
75 deg.....	0.90	0.70	0.59	0.46
90 deg.....	1.28	1.21	1.13	1.05

[24] W. H. Herschel, "Viscosity and Friction." Jour. S.A.E., 1922.

[25] Loc. cit., p. 470, et seq.

[26] G. B. Karelitz, loc. cit.

Assuming the values of p_{av} to be:

α, deg.....	45	60	75	90
p_{av}.....	1.7 P	1.3 P	1.2 P	1.2 P

in order to take into account the difference between the area of the pressure zone and the projected area

$$Q = k \frac{P}{\mu} \frac{d}{L} (R - r)^3$$

where P is the pressure on the projected area and k is a coefficient equal to

	$c = 0.3$	$c = 0.5$	$c = 0.7$	$c = 0.9$
45 deg.....	0.58	0.34	0.15	0.06
60 deg.....	0.75	0.52	0.32	0.21
75 deg.....	1.08	0.84	0.71	0.55
90 deg.....	1.54	1.45	1.35	1.26

If Q_{gal} is expressed in gallons per minute, pressure P in pounds per square inch, viscosity Z in centipoises, the clearance $s = D - d = m(R - r)$ in inches, $Q_{gal} = 0.00022\, k \frac{P}{Z} \frac{d}{L} (1000\, s)^3$.

It should be noted that the influence of each element of the bearing on the oil leakage is not apparent directly from the expression derived, since the change of each element affects the value of c, hence the coefficient k.

Discussion

L. J. Bradford.[27] During the last year considerable work has been done at the Pennsylvania State College on a machine designed to permit accurate determination of the pressure distribution within the oil film in a complete bearing, lubricated from one end. In no case has the pressure been found to be zero at the point of minimum film thickness. For the cases investigated the pressure has varied from 55 per cent to 93 per cent of the maximum pressure. Positive pressure was found to extend for 20 to 25 deg. beyond the point of minimum film thickness. The point at which the pressure becomes zero seems to be affected by all the variables that enter into a bearing, but there has not as yet been done enough work to enable one to say just how it is affected by each. This discrepancy between the assumed and actual pressure distribution probably accounts for the deviation of the observed path of the shaft center from that predicted by the author.

The author's equation for shaft displacement, however, gives results in remarkable agreement with the values actually observed. Comparison between calculated and observed shaft displacements for load ranging from 100 to 400 lb. per sq. in. showed discrepancies that at no time exceeded 0.0003 in. This was about 10 per cent of the displacement and about the limit of accuracy of observation.

S. A. McKee.[28] The author has presented an interesting and valuable paper relating to bearing design. Its value as a practical working guide, however, depends chiefly upon the correctness of certain assumptions that have been made. Possibly the place where the uncertainty of the correctness of the assumptions has its greatest significance is in the evaluation of the minimum film thickness for a given bearing under given conditions, and in deciding from this value whether or not the bearing will operate in a satisfactory manner. Unfortunately this is a point where the lack of sufficient reliable and systematic experimental evidence causes a considerable degree of uncertainty. The foregoing is intended not to especially criticize the particular methods or values chosen by the author, but to emphasize the need for research on this phase of the problem.

There is one factor in the determination of the minimum film thickness that possibly might have been given greater emphasis by the author. This is the question of the effect of the deflection of the journal and bearing under the load imposed. This question in many cases may be neglected when the loads are relatively light, as compared to the rigidity of the parts involved. However, the writer would like to ask the author to what extent the deflection of the journal will affect the value of the minimum film thickness of 0.0009 in. as given in the example of the slow-speed oil-ring bearing. In this case a journal 11 in. in diameter with a working length of 22.5 in. has a total load of 35,400 lb., and even if the bearing is so constructed that it may align itself properly, the deviation of the journal axis from a straight line may be such that in some parts of the bearing the minimum film thickness is much smaller than the computed value. This is suggested not to imply that this particular bearing will not operate successfully under the given conditions, but to point out that there are cases where serious difficulty may be encountered unless journal deflection is considered.

A. E. Flowers.[29] The Society and the author are to be congratulated upon this paper, which is the first contribution to the subject of oil-ring bearings since the classical work of Lasche in Germany many years ago.

It is to be hoped that if the author has not already carried out similar studies on the lubrication of vertical bearings, he will find an opportunity to do so and will present his results to our Society for publication.

H. A. S. Howarth.[30] The writer was glad to see Fig. 5 in this paper which gives a factor for end leakage, showing how it depends upon the width-length ratio. Although this was derived for slightly inclined flat surfaces and does not apply exactly to journal bearings of any ordinary kind, it gives a fair idea of the magnitude of the influence of leakage. Charts of journal-bearing characteristics, based on the hydrodynamic theory, but neglecting leakage, may be corrected by means of this curve. The "factor for end leakage" should be represented by some letter, such as S. Then, when studying the bearing by means of a chart neglecting side leakage, if we find that an oil film of a given form and dimensions would appear to carry a unit pressure w, we must multiply this by S in order to determine the actual unit pressure w_1 that would be supported by this film when leakage is taken into account. Thus, we have $w = w_1/S$. w and the width-length ratio would be found from the problem data; but w would be a fictitious value of the unit pressure that should be used for entering the scale of a chart, such as contained in the pamphlet entitled "A Graphical Analysis of Journal-Bearing Lubrication."

A chart somewhat like that offered by the author, but not so generally useful, was published in London *Engineering* in a discussion of the Michell bearing, February 20, 1920. The author has, however, used the opposite names for the same bearing dimensions. He uses "width" for what *Engineering* calls "length," and vice versa. It seems more logical to use the word "length" for the dimension in the direction of motion and "width" for that at right angles thereto or along the axis of the shaft. The term "end leakage" means the same as what others, including the writer, have called "side leakage." In view of this confusion in bearing nomenclature it would seem to be high time that some

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[28] Associate Mechanical Engineer, Bureau of Standards, Washington, D. C.

[29] De Laval Separator Company, Poughkeepsie, N. Y.

[30] Vice-President and General Manager, Kingsbury Machine Works, Philadelphia, Pa. Mem. A.S.M.E.

committee should endeavor to coordinate the work of various investigators and to have them use similar names and symbols, thereby avoiding confusion in the minds of engineering designers.

The leakage factor may be expressed in different ways. The writer made a study of Mr. Michell's theory on leakage for flat bearing surfaces many years ago and drew a chart. This was published as Fig. 2 in a letter entitled "A Graphical Study of Journal Lubrication," April, 1927, *Mechanical Engineering*. This shows a correction factor k to be applied for determining the relation of the friction to be expected in a bearing when leakage is taken into account to the friction that would exist in the bearing if it could carry the same load without leakage. It indicates that when the bearing surface extends three times as far in the direction of motion as it does at right angles thereto, the friction would be about three times as great as could be expected for the same load if the influence of leakage could be neglected. The approximate relation between this curve and that of the author is that the ordinates are the square root of the reciprocals of his. For a further study of the influence of film leakage upon friction, the writer would refer those interested to the letter above mentioned.

Film leakage as influencing the carrying capacity and the friction in journal bearings is now being intensively investigated with a view to determining more reliable factors for application to the study of journal bearings. It is hoped that the work along this line that is being done by Mr. Kingsbury will be completed within the following year. When this is available the whole field of journal-bearing film investigation can profitably be reviewed, theoretical considerations coordinated, and the various charts made more available for general use.

The author suggests as an alternative to the use of oil rings the employment of forced-feed lubrication. It seems, however, that there is another alternative that is very important indeed—the field of disk oiling or oiling by means of viscosity pumps. Disk oiling has been used in one form or another for many years, but it is being further developed by Kingsbury Machine Works, Inc., as a direct result of its employment for oiling thrust bearings. In this use, disk oiling requires a minimum of the upper shell to be cut away. Suitable oil channels may therefore be employed and the bearing made to carry a load in any radial direction. This is important for journal bearings of the film type when used in such machines as electric motors. Disks need not slip as rings do.

The author refers back to pressure curves that he presented several years ago. Although based upon certain assumptions that he was satisfied to use, they are not rigorously correct, so far as the Reynolds hydrodynamic theory is concerned. Hence, as previously pointed out, they do not quite agree with the charts in "A Graphical Analysis of Journal-Bearing Lubrication." An extension of the Reynolds theory indicates clearly that it is not necessary for the pressure to be zero at B, the point of closest approach, in Fig. 1. It seems, therefore, that these differences are worthy of further study and that the results of experiments by various investigators should be published so far as relevant thereto.

HAROLD C. CLAUSEN.[21] The author is to be congratulated on presenting a valuable paper which will be of use to all engineers interested in the bearing problem. Two and one-half years ago the writer had occasion to discuss with Mr. Karelitz some questions in connection with the bearing problem, and on the question of heat dissipation sent him the following note:

The writer has used the figures of 2–5 and even 10 B.t.u. per hour per sq. ft. per deg. fahr. for the calculation of the rate of heat transmission from a solid surface where a liquid is being cooled on one side and a gas being warmed on the other side. These figures are taken from "Engineering Thermodynamics" by C. Lucke and are based on experiments on hot-water radiators. The figures are equivalent to 0.0073 to 0.01825 and 0.0365 watts per sq. in. per deg. cent. and depend on the character of the liquid surface and the air velocity, the lower figure corresponding to still-air conditions and the top figure to where there is considerable turbulence of the air.

It is pleasing to note that the foregoing figures are in general agreement with the results shown in Fig. 28 of the paper entitled "Variation of Computed Coefficient of Dissipation for Actual Machines."

The question of friction and temperature was also discussed, and the writer noted to Mr. Karelitz that—

The formula used by the writer for estimating the friction loss in high-speed bearing is: B.t.u. lost in journal friction per hour $= \frac{190\, l\, d\, v}{t\text{-}32}$ where l = journal length and d the diameter, both in inches, v the velocity in feet per second, and t the temperature in degrees fahrenheit. This formula is based on the shearing resistance of the oil of the order of 0.45 lb. per sq. in. These figures are taken from a paper on "High Speed Bearings," by G. Stoney, *Engineering*, London, Aug. 7, 1914. Also see *Engineering*, London, March, 1922.

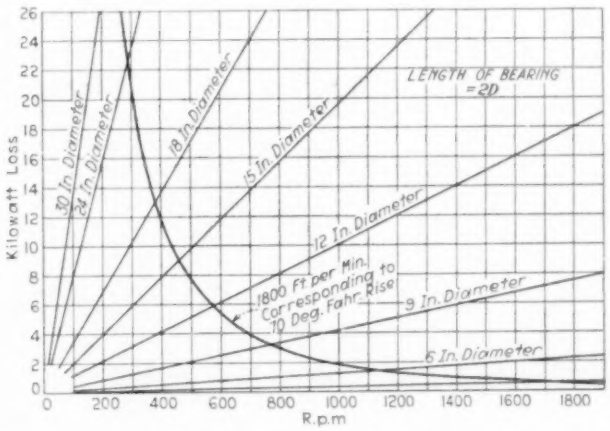


FIG. 30 FRICTION LOSS IN DIFFERENT BEARINGS FOR DIFFERENT R.P.M.

Using the information contained in the foregoing, a curve was made (see Fig. 30) showing the friction loss in different bearings for different r.p.m., and also indicating the limiting speed at which the bearing can be air-cooled. This corresponds to a circumferential velocity of the shaft of about 1800 ft. per min. for the 70 deg. fahr. rise of bearing temperature. Similar curves can be made up for Kingsbury thrust bearings, and the velocity of 1800 ft. per min. for the mean diameter of the shoe will give 70 deg. fahr. rise in bearing temperature. In both cases it is assumed that the air is kept in motion by some nearby revolving element—that is, we have some draft of air on the bearings. These results appear to agree fairly well with practice.

The writer would like to draw attention to three valuable references on the general bearing problem, and they are as follows:

1 "Reibung and Schmierung in Machinenbau" ("Friction and Lubrication in Machine Construction"), by Gumbel and Everling, 1925.

2 "Grungzüge der Schmiertechnik" ("Fundamentals of Lubrication Technic"), by E. Falz, 1926.

3 "Bedeutung der Hydrodynamischen Lagerreibungstheorie für die Praxis" ("The Practical Application of the Hydrodynamic Theory of Bearing Friction"), by Dr. Ing S. Kaisskalt, Berlin, *V.D.I.*, band 71, no. 7, Feb. 12, 1927.

In these references the value of the coefficient of friction

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$\mu = \text{constant}\sqrt{\frac{ZN}{P}}$, and this does not agree with $\mu = \text{const.}\frac{ZN}{P}$ as shown in Fig. 23, "Journal Friction of Sleeve Bearings." The writer would like to know if the author can give any explanation of the different formulas for the coefficient of friction.

Author's Closure

The author agrees with Professor Bradford and Mr. Howarth that the location of the zero pressure point in a bearing must be studied further. It is significant, though, that on many occasions a vacuum has been observed in the "up part" of partial bearings.

Vertical bearings are being considered at present, together with the question of bearings under rotating loads.

Disk lubrication, as suggested by Mr. Howarth, is used, indeed, in some application such as marine bearings. But the arrangement is rather expensive and could not be conveniently embodied in, say, electric motors in face of the strong competition in this field.

The feasibility of such a device should certainly not be overlooked.

In addition to the remarks of Mr. Clausen, it is probably worth while to point to the bibliography on lubrication given in the Progress Report of the A.S.M.E. Special Committee on Lubrication, Trans., 1928, paper APM-50-4, where the authors indicated by Mr. Clausen in his discussion are mentioned among the references.

APM-52-6

Power Application to Oscillating Axles

By ALAIN MADLÉ,[1] LIMA, OHIO

In this paper a type of axle is described in which the load is not carried by the wheel center but is suspended at the end of an arm which is permitted to swing like a pendulum about the wheel center. This mechanism is called a differential drive, and its application to motor-driven vehicles is described. The forces acting in the mechanism and the conditions necessary for an equilibrium of forces are worked out mathematically, and a comparison is given of starting motor trucks by two systems: first, when equipped with an oscillating axle with differential drive, and second, with the usual method of drive.

By means of the differential drive it is claimed that torque can be applied momentarily to the wheel without turning it, and that an elastic connection is provided between the power line and the wheel which cushions torsional impacts. Whenever there are changes in the running conditions of the vehicle, fluctuations of energy between the wheel assembly and the vehicle body take place. The reaction forces of these fluctuations provide an increased pressure against the ground, an additional accelerating force, and a greater retarding force.

WITH a view toward cushioning the horizontal road shocks and improving the general riding qualities of vehicles, a type of axle has been developed in which the load is not carried by the wheel center but is suspended at the end of an arm which is permitted to swing like a pendulum about the wheel center. The influence of this suspension on the ability of the axle to absorb road shocks is outside the scope of this paper.

Here will be discussed only the peculiar qualities of such an axle which appear when a power drive is applied. The principle of such an axle is illustrated in Fig. 1.

Since a certain movement is possible on the part of the wheel center in relation to the frame, the logical point to enter the wheel with the power line is the point *C*, Fig. 1, which is common to both the frame and the wheel. The drive which lends itself best to this arrangemnt is a planetary-gear drive in which the sun wheel is connected to the road wheel and the planet is carried to the end of the swinging arm as indicated in Fig. 2.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The Planetary Drive

This planetary or epicylic mechanism consists of three members—the sun wheel or track gear, the planet or track pinion, and the arm or pinion carrier. For the movement of these three members, the fact is typical that the relative velocity between the circumference of the track gear and pinion is constant, or, in other words, that the velocity of the carrier at the radius of the pinion center equals the difference between the circumferential velocity of the track gear and the circumferential velocity of the pinion.

Any one of the three members can be driving the other two members. From the relation of forces as shown in Fig. 3, it readily can be seen that the mechanism is able to transmit torque from the driving member to one of the driven members only when the second driven member is retained or supported by a circumferential force equal to the circumferential force of the output torque.

It is of course possible for the supported member to be at rest.



Fig. 1 Axle in Which Load Is Suspended at End of an Arm Which Swings Pendulum-Like About Wheel Center

Then the total transmitted energy flows from the driving member to the driven member and the three-member drive becomes a normal gear drive. It is also possible for both driven members to be rotating—then the input energy is distributed between them. Because of its peculiar conditions, the mechanism characterizes itself as a so-called differential drive.

A mechanism of this kind is sometimes used as a clutch, in which case a brake is provided to retard the movement of the part which is to be held stationary when the drive is fully engaged. The sliding engagement of the brake allows the gradual application of torque to the driven shaft, even before it turns.

Application of the Differential Drive to Vehicle Propulsion

As previously pointed out, an epicylic-gear mechanism can be used to advantage as driving mechanism for oscillating

vehicle axles. When so used the track gear of the epicylic drive is fastened to the road wheel of the vehicle, which itself is mounted rotably about a dead axle or stub shaft. The power line is enclosed in an axle housing which, by means of the spring, carries the weight of the vehicle. The swinging arm carries on its one end the dead axle, and on the other end a trunnion bearing for the axle housing and a bearing for the track pinion which is in mesh with the track gear. (Fig. 5.)

Forces Acting on the Vehicle

In order to analyze the dynamics of this application, the diagram shown in Fig. 6 will be used. The point O signifies the wheel center, and the point C the center of the pinion or the trunnion bearing. The radius of the wheel will be called R', the pitch radius of the track gear R, and of the pinion r. The length of the pinion carrier will be indicated as ρ.

According to the assumptions previously made concerning the design of the axle, the weight of the vehicle is always carried at the point C. Therefore when no other forces are applied the carrier will stand vertically.

It will be assumed for the moment that the carrier is held in this vertical position by some mechanical means. If now a torque T through the drive shaft, pinion, and track gear is applied to the wheel, a propelling force P_w will act at the wheel center. As reaction, this force is of equal amount and opposite in direction to the tractive effort acting at the contact point of tire and ground. At the same time the arm exerts a pressure P_c against its support, which is equal in amount and opposite in direction to the circumferential component of the tooth pressure in the track-gear drive.

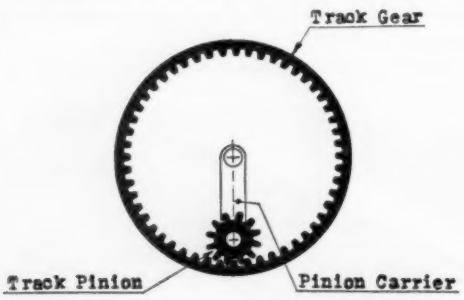


Fig. 2 Planetary-Gear Drive

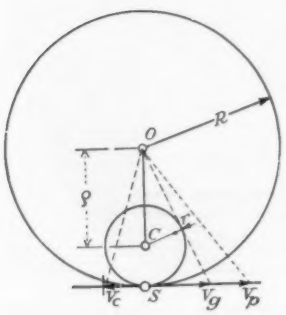


Fig. 3

ω_c = angular velocity of carrier
ω_p = angular velocity of pinion
ω_g = angular velocity of track gear
V_c = circumferential velocity of carrier (at C)
V_p = circumferential velocity of pinion at pitch line
V_g = circumferential velocity of track gear at pitch line
$V_p = V_g + V_c$; $\omega_p r = \omega_g R = \omega_c \rho$

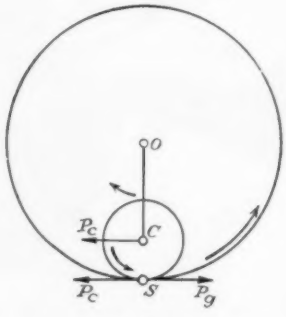


Fig. 4

P_g = circumferential force at pitch line of track gear
P_p = circumferential force at pitch line of pinion
P_c = circumferential force at pinion center
$P_p = P_g = P_c$

The propelling force can be represented as:

$$P_w = T/R'$$

and the pressure against the support of the swinging arm as:

$$P_c = T/R$$

The force P_c—the pressure of the swinging arm on its support—is therefore greater in the proportion of the radii R'/R than the propelling force P_w.

Conditions for an Equilibrium of Forces

If now the support of the point C is removed, the force P_c must be sustained by a force of equal amount in order that the torque T can be applied to the wheel.

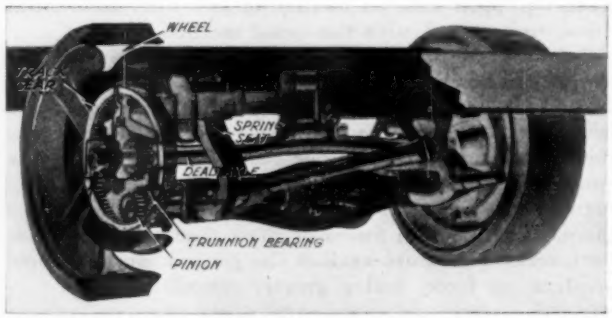


Fig. 5 Epicyclic-Gear Driving Mechanism for Oscillating Vehicle Axle

It can be seen readily from Fig. 7 that the weight supported at point C will, with increasing angle φ—the angle between the carrier and the vertical direction—develop an increasing component in the direction of the force P_c. This component can be expressed as $W \sin \varphi$.

Because of the fact that by the incorporation of the oscillating axle the vehicle practically consists of two parts, each of which may move with relation to one another, it appears to be appropriate to consider the resistances of these possible movements for both portions separately. It therefore becomes necessary to distinguish between the front-end resistance R_c, consisting of the rolling resistance of the front wheel and the air resistance of the vehicle, and the rear-wheel resistance R_w, the rolling resistance retarding a movement of the rear wheels.

The front-end resistance R_c can be assumed to be acting at the point C in a horizontal direction, and the rear-wheel resistance at the contact point of the rear wheels and the ground. The sum of these two resistances is the total resistance of the vehicle which must be overcome by the propelling force.

Assuming both resistances and the applied torque to be constant and the velocity unchanging, then the resultant of all forces and of all the turning moments about the wheel center, respectively, must be zero. To fulfil these conditions the carrier must assume a certain angular position. These conditions can be expressed as:

$$P_w = R_w + R_c \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

$$P_c = Mg \sin \varphi + R_c \cos \varphi \quad \ldots\ldots\ldots\ldots\ldots [2]$$

Here the force P_c is still sustained by concrete forces—the dynamics of the mechanism are not different than they would be if the carrier would be held in a fixed position by mechanical means.

The angle between the carrier and the vertical position is a function of the engine torque, the weight of the load, and the

resistance against the front portion of the vehicle, and is determined by the second of the equations just presented.

Transition Between Two Stages of Equilibrium

It is evident that each change in the acting forces will demand another angle between the carrier and the vertical position to bring the system again into a state of equilibrium. Any change in the condition will accordingly necessitate a movement of the carrier.

Whenever the carrier moves, the mass of the weight which is supported on the rear axle or at the end of the carrier must be accelerated in the path of a circle about the wheel center. The accelerating force of this movement will sustain the force P_c, and the constant flow of torque to the wheel will not be interrupted.

In order to study the process of transitions between two stages of equilibrium, the vehicle with the oscillating driving axle can be reduced to a system as shown in Fig. 8. The mass of the weight which is supported by the rear axle M_1 is assumed to be concentrated at the point C, and the mass of the weight of the wheel assembly m is concentrated at the point O, the rear-wheel center; furthermore the mass of the weight supported on the front axle M_2 is concentrated at the front wheel center. The movement of the masses m and M_2 is held to a horizontal straight-line path.

Whenever the equilibrium of such a system is disturbed by the variation of the outer forces, oscillations of the masses with relation to each other will result. This case has been treated

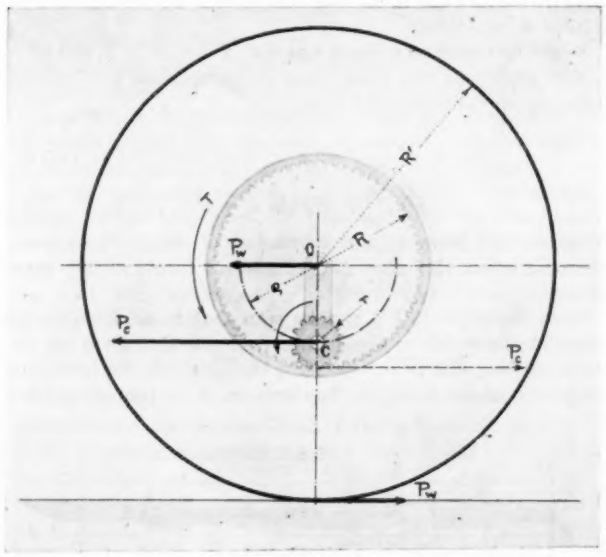


Fig. 6

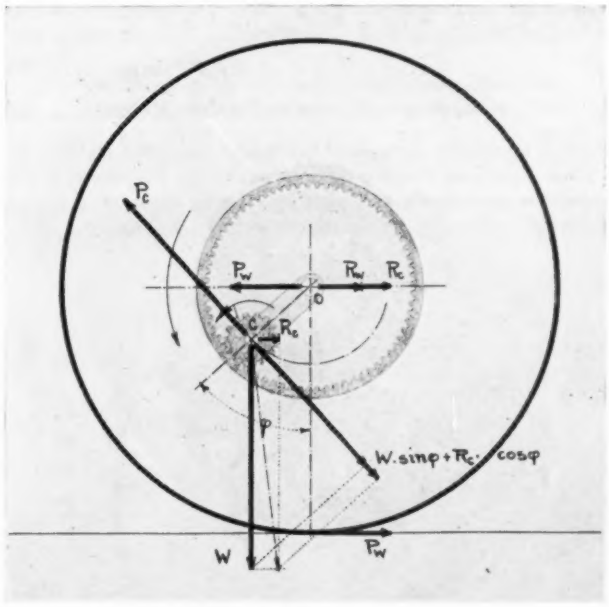


Fig. 7

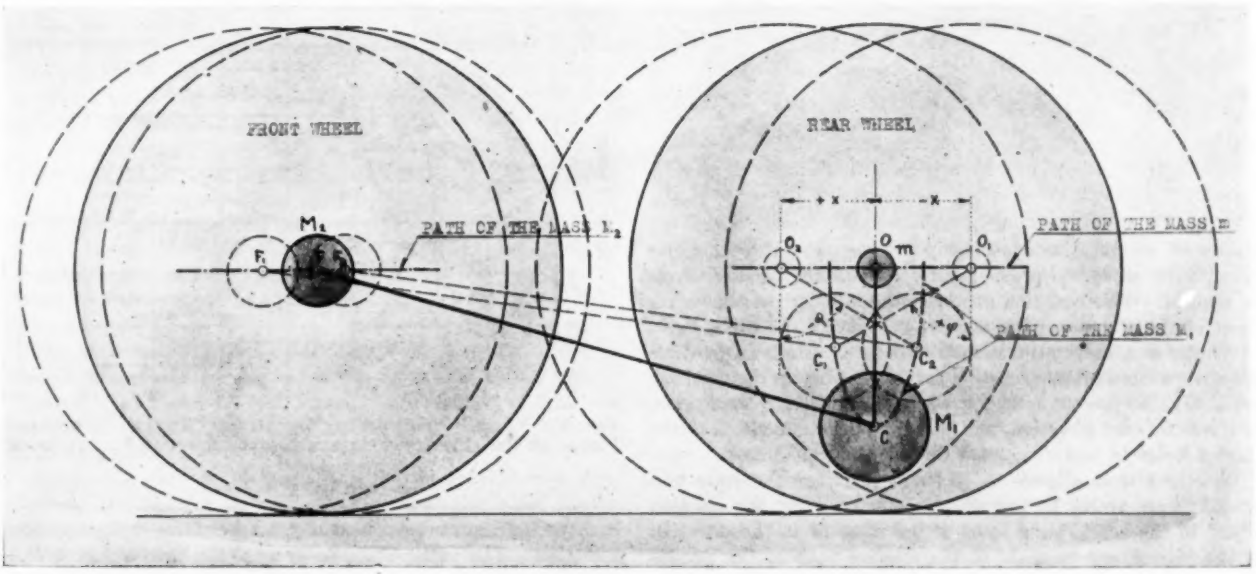


Fig. 8 Diagrammatic Representation of Vehicle With Oscillating Driving Axle

mathematically in the Appendix, and through the introduction of two independent coordinates, x for the translatory, and the angle φ for the rotary motion and neglecting M_2, the equations of the movement of this system have been established as

$$\frac{R}{R'}P_c - R_w - R_c = (m + M)x'' + M\rho\varphi''\cos\varphi - M\rho(\varphi')^2\sin\varphi \quad [3']$$

$$P_c - Mg\sin\varphi - R_c\cos\varphi = M\rho\varphi'' + Mx''\cos\varphi \quad [4']$$

where M stands for the mass of the weight supported on the axle.

These equations describe quite generally the movement of the system in transitions, the equations being those of a forced oscillation, influenced by the outer forces P_c, R_w, and R_c. As a

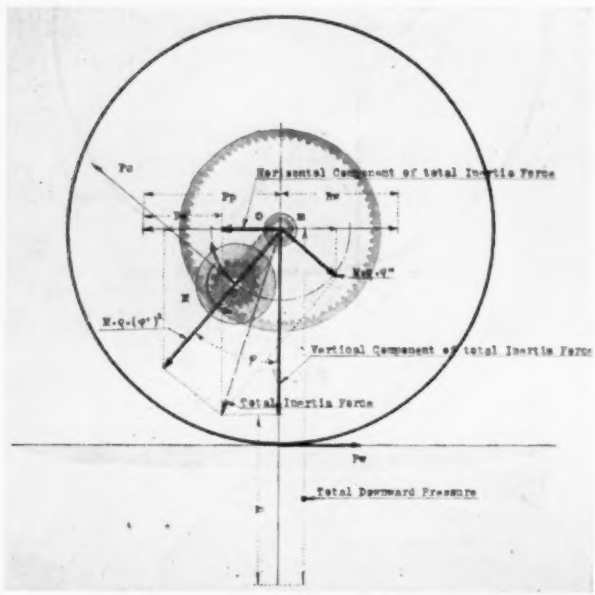


FIG. 9

matter of course, the relationship between the three angular velocities as described previously for the differential drive must be applied. This equation expresses the influence of the movement of the system on the angular velocity of the drive shaft.

Whenever a movement is restricted to a certain path, there are always inertia forces reacting perpendicularly to the resulting path. In the present case the wheel is supporting the system in the downward direction, and therefore the components of the inertia forces in this downward direction are of interest.

It follows from Equations [3'] and [4'] that there are two inertia forces acting at the wheel center, namely, the reaction $M\rho\varphi''$ to the accelerating force of the mass M in the direction of the momentary tangents of the relative path of the pendulum, and the centrifugal force $M\rho(\varphi')^2$ in the direction of the radius vector. The total vertical force at the support is the sum of their vertical components. This total pressure is in practically all cases positive, directed downward, and thus contributes to the weight supported on the wheel.

The conclusion can be drawn that *whenever there are changes in the constant running conditions of a vehicle equipped with an oscillating axle, there will be a downwardly directed force exerted on the wheel.*

Special cases of the movement of an oscillating system can be discussed on the basis of the general equations if certain assumptions as to the time change of the outer forces are made.

SPECIAL CASE OF TRANSITION WITH VEHICLE AT REST

Among the special cases of transitions is one of particular interest where the translatory movement of the wheel center is blocked—as for instance when there is extraordinary resistance R_w preventing the movement of the wheel. Then the vehicle can be represented by a reduced system as shown in Fig. 9. In this case also the horizontal components of the inertia force become of importance.

Assuming the wheel to be at rest with $x = 0$, $x' = 0$, and $x'' = 0$, and neglecting the front-wheel resistance, the fundamental equations appear then in the form

$$\frac{R}{R'}P_c + M\rho(\varphi')^2\sin\varphi - M\rho\varphi''\cos\varphi = P_p \quad [5']$$

$$M\rho\varphi'' + Mg\sin\rho = P_c \quad [6']$$

Equation [6'] refers to the movement of the carrier system, while Equation [5'] gives the relation of forces at the wheel center.

From Equation [5'] it can be seen that in addition to the propelling force due to the applied engine torque there are two forces existing due to the swinging carrier mass, the horizontal component of one acting in the direction of the propelling force,

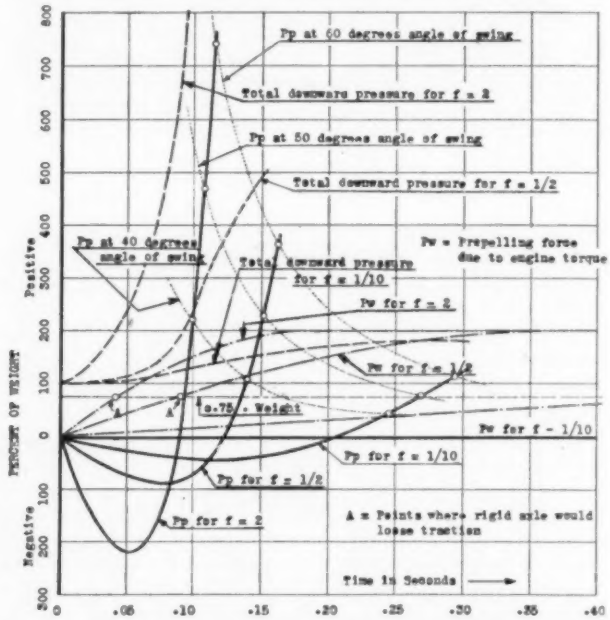


FIG. 10 TOTAL PROPELLING FORCE AND TOTAL DOWNWARD PRESSURE FOR DIFFERENT CLUTCH ENGAGEMENTS AS FUNCTIONS OF TIME

($P_c = 5Mg$; $f = a/a_m$.)

and the horizontal component of the other force in the opposite direction. The vertical components of the inertia forces $M\rho(\varphi')^2\cos\varphi$ and $M\rho\varphi''\sin\varphi$ are again downwardly directed, thus increasing the pressure on the ground due to the weight supported by the wheel.

The left-hand member of Equation [5'] determines the resultant propelling force. It is evident that a movement of the entire system cannot start before the resultant propelling force exceeds the resistance.

In order to determine the actual resultant propelling force, a time-change relation for the force P_c must be assumed. This is carried through in detail in the Appendix. The time change of the force P_c can be considered as representing the engagement of the clutch.

On this basis and for definite dimensions of the axle the total propelling force can be represented as a function of the time as is indicated in Fig. 10. This diagram shows that the resultant propelling force at the wheel center is negative at first and then increases through zero to positive values, which beyond a certain angle of swing becomes greater than the normal propelling force of the rigid axle. It shows further that the total downward pressure is greater than the weight at all times and, compared with the force P_w, which must be supported by the tire, will assure traction even for abnormally small coefficients of friction. The data appearing in this diagram invite an analysis of the various factors which influence the magnitude of the forces at the wheel center, but although interesting, such an analysis would extend this paper beyond its necessary limits.

For the preceding analysis it is assumed that the necessary relation of the planetary drive, as expressed by the formula $\omega_p r = \omega_c \rho$, is fulfilled, which will be true in practically all cases. Only in the event of an extremely great reduction in the power line would the relation of the angular velocities influence the forces at the wheel center.

Limitations of the Movement

Thus far the movement of the carrier has been considered unlimited, and in certain instances it would result in a rotary motion of the carrier with a periodically fluctuating angular velocity. Such an unlimited rotary motion would of course create inertia forces, of which the vertical as well as the horizontal components would be subjected to a periodical change. Since the vertical and horizontal components of the inertia forces must be positive in order to create favorable effects, the swinging movement of the carrier must be limited to approximately the first quadrant.

Practical reasons require a further limiting of the carrier swing, but since the additional propelling force, especially in starting, reaches greater values at greater angles, the angular limits for the movement should be extended as far as the construction permits.

It is unavoidable that under certain conditions the carrier encounters the limiting end stops. In order to keep the resulting impact force small, the elastic qualities of the end stops must be given full consideration. To obtain greatest flexibility the end stops are made of thick-walled rubber tubing and are supported in properly designed pockets. (Fig. 11.)

Starting

The previously detailed analysis makes it possible to discuss the process of starting an automotive vehicle equipped with an oscillating axle and differential drive, as compared with the starting of the same vehicle equipped with a conventional type of driving axle.

In order to start the movement of a vehicle, the circumferential force at the tire (or the propelling force at the wheel center) must be greater than the sum of all the resisting forces there. In a conventional type of vehicle with a positive connection between the power line and wheel, the required torque must be transmitted by the sliding clutch because of the fact that the engine delivers torque only above a certain minimum speed. In other words, the vehicle has to be started before the clutch can be fully engaged. If the torque required to start the movement of the vehicle is greater than the torque which can be transmitted by the sliding clutch, the engine will stall when the clutch is fully engaged.

It is a practice to start vehicles under difficult conditions by racing the engine and dropping in the clutch suddenly. Due to the inertia of the moving parts it is possible to exert momentarily more torque than the engine would provide continuously.

The torque that can be applied to the wheel is determined by the traction conditions. It is always possible, and in case of the "flywheel" start it is almost inevitable, that the applied torque exceeds the traction torque and causes wheel slippage.

On a vehicle with oscillating axle it is not necessary to slide the clutch and bring the vehicle to a start before fully engaging it. The entire starting process can be here subdivided into three periods.

The first of these periods has already been discussed. The torque, if the carrier stands in a vertical position, does not encounter any static resistance other than that of the front wheels, which because of the small load supported on the front

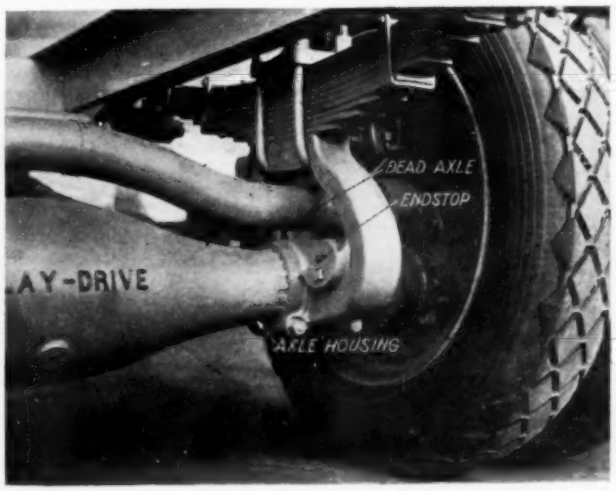


Fig. 11 Showing Axle Housing and End Stop

axle is negligible. The excess torque is consumed in accelerating the carrier (and with it the front portion of the vehicle), while the resisting weight component increases with the angle. It has been shown furthermore that while the carrier is swinging upward, inertia forces are built up, the vertical components of which press the wheel against the ground, while the horizontal components apply additional propelling forces to the wheel.

The second period of the starting begins when the sum of all the propelling forces overcomes the static resistances. The wheel begins to move—the carrier decelerates relatively to the wheel center and finally reverses its movement, and the vehicle and the wheel system then perform horizontal oscillations about the common center of gravity. These fluctuations of energy from one member to the other will, by virtue of the damping in the mechanism, die out after some time, and the third period, the constant running position, will begin.

The most important feature in the starting of a vehicle of this type is the fact that wheel slippage is practically eliminated. Because of the vertical components of the inertia forces caused by the up-swinging mass the weight utilized for traction is considerably increased, and only in exceptional cases will the applied torque exceed the available traction. Because of the additional propelling force, the vehicle with the oscillating axle can be started with a smaller amount of engine torque, and accordingly with less circumferential force on the tire.

To illustrate the difference between the two systems in starting, the following test was made. A motor truck equipped with an oscillating axle with differential drive was started on a soft sand bed. Then the oscillating action of the axle was blocked (thus making the differential drive a positive internal-gear drive) and the truck was started on the same ground. In both cases there was just enough power applied to put the vehicle in motion.

The tracks as photographed are shown in Fig. 12. The difference between these two tracks is obvious and substantiates the results previously deduced.

Encountering an Obstruction While Running

Any wheel when encountering an obstruction has a tendency to leave the ground, or at least to reduce its pressure on the ground to some extent, and in such a case the applied engine torque may not develop sufficient traction, thus causing the wheel to speed up while slipping or spinning freely. When this wheel regains traction after the impact, it will have a circumferential velocity which is different from the vehicle speed. This difference in speed imposes a torsional impact on the wheel, which in most cases causes the tire to slip.

If the wheel of a vehicle with an oscillating axle strikes an obstruction in the road the two systems—carrier and wheel—will begin to oscillate about the oscillation center. Since such an oscillation always accelerates the common center of gravity of both systems in the vertical direction, it is evident that the reaction of the accelerating force will press the wheel with an increased force against the ground. The increased pressure forcing the wheel downward counteracts the uplifting impact force. In practically all cases, therefore, the wheel of an oscillating axle will not lose traction in going over an obstruction, and thus will prevent the spinning of the tire on the road.

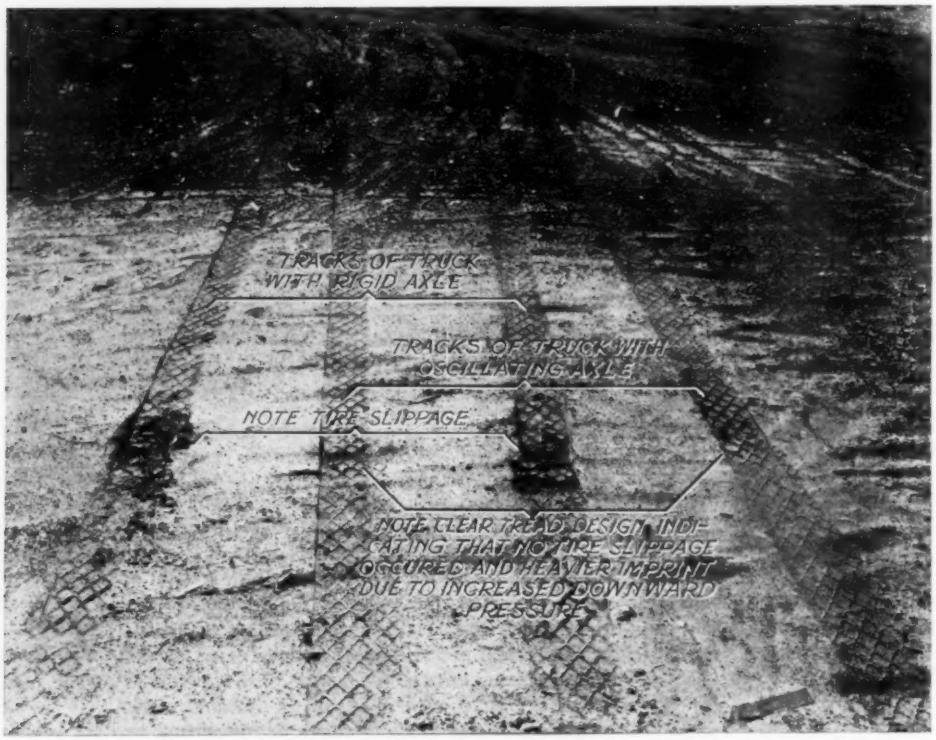


Fig. 12 Tracks Made by Tires of Truck With Rigid Axle and Truck With Oscillating Axle

Effects of Braking Upon an Oscillating Axle

When the brakes are applied to an oscillating axle, the effect is as though the carrier had been momentarily clamped to the wheel with an adhesion proportional to the braking force. This momentary fixed connection between carrier and wheel will become a sliding contact when either the weight component overcomes the brake adhesion or, as is more often the case, when the end stop is encountered.

Since the wheel is forced to follow the road, the common movement of wheel and carrier will result in an elevation of the center of gravity. In addition, the wheel is accelerated and the front portion decelerated during this period.

It is true that in this first period of braking no energy is consumed by the brakes, but the elevation of the center of gravity transforms a part of the kinetic energy of the vehicle into potential energy before the action of the brake begins. The amount of kinetic energy transformed into potential energy depends on the height to which the center of gravity is lifted, and for this reason it is desirable to extend the arc of the pendulum swing as far as the construction will permit.

It may therefore be stated that application of the brake to the oscillating axle will reduce the vehicle speed to a certain extent before the sliding action between brake shoe and drum takes place.

Summary

In recapitulation, it may be stated that the following definite effects have been determined as the result of the application of a differential drive to an oscillating axle:

1 By means of the differential drive, torque can be momentarily applied to the wheel without turning it, this gives the effect of a second clutch in the power line in which no energy is destroyed

2 The oscillating axle, by providing an elastic connection between the power line and rear wheel, cushions the torsional impacts that would be encountered with a rigid axle

3 Whenever there are changes in the running conditions of the vehicle, fluctuations of energy between the rear-wheel assembly and vehicle body take place; the reaction forces of these fluctuations provide an increased pressure against the ground

4 By virtue of the fluctuation between the rear-wheel assembly and the vehicle body, an additional accelerating force as well as an additional downward pressure is applied to the driving wheels in starting

5 Due to the transformation of part of the kinetic energy of the moving vehicle into potential energy, an additional retarding force is provided when the brakes are applied.

Appendix

THE OSCILLATING AXLE SYSTEM

THE vehicle with an oscillating rear axle has been reduced to the system shown in Fig. 13. For the calculation two independent coordinates are introduced, one linear, in the horizontal direction, called x and the other the angle φ the arm OC makes with the vertical direction.

In the system shown the mass m represents the mass of the parts of the rear axle which can swing relatively to the vehicle, while the mass M_1 represents the mass of the weight supported on the rear axle and M_2 the mass of the weight on the front axle. The mass m is assumed to be concentrated at the center O of the rear wheel, M_1 at the end of the arm in point C, and M_2 at the center of the front wheel. It is in the nature of the problem that the movement of m and M_2 is restricted to a horizontal path.

To deduce the equations of the movement of this system it is necessary to employ Lagrange's equations.

$$F_z = \frac{d}{dt}\left(\frac{\partial L}{\partial x'}\right) - \frac{\partial L}{\partial z}$$

The derivatives of the coordinates are to be represented as follows:

$$\frac{dx}{dt} = x'; \quad \frac{d^2x}{dt^2} = x''; \quad \frac{d\varphi}{dt} = \varphi' = \omega; \quad \frac{d^2\varphi}{dt^2} = \varphi'' = \epsilon$$

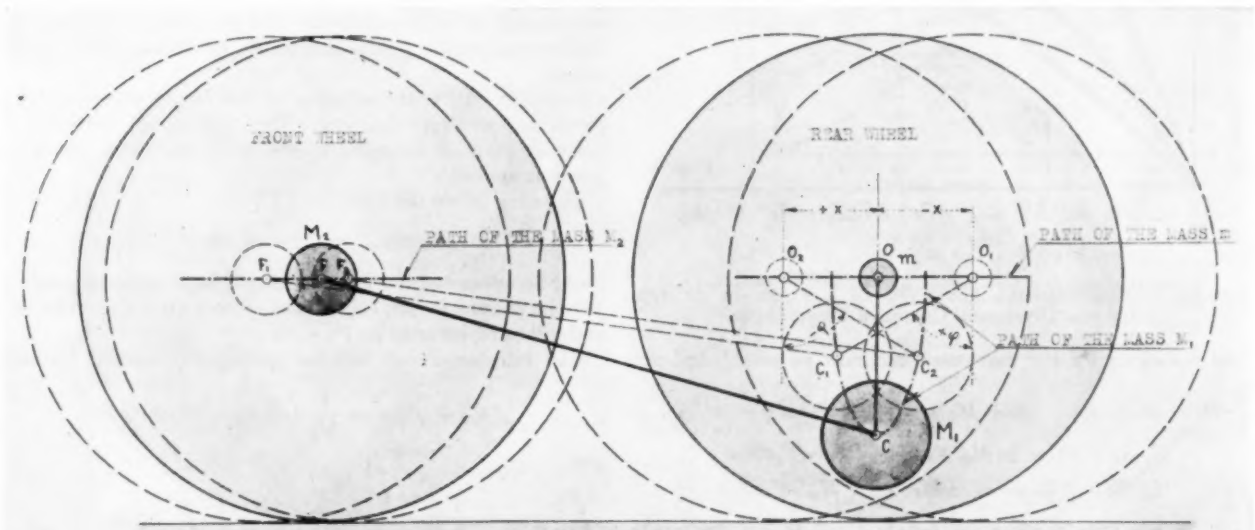


FIG. 13 DIAGRAMMATIC REPRESENTATION OF VEHICLE WITH OSCILLATING DRIVING AXLE

To make use of Lagrange's equations, however, an expression for the kinetic energy of the system must be first established.

The kinetic energy belonging to x can be expressed as

$$L_x = \frac{1}{2}m(x')^2$$

For the kinetic energy belonging to φ, both the velocity of the translatory motion x' and the circumferential velocity of the rotary motion $\rho\varphi'$ must be considered. These two components give a resultant velocity

$$V_\varphi^2 = (x')^2 + \rho^2(\varphi')^2 + 2\rho\varphi' x' \cos\varphi$$

Due to the fact that the mass M_2 is forced to travel in a horizontal path and the mass M_1 in a circular path, the kinetic energy belonging to φ will consist of two components, one in horizontal direction, namely,

$$L_{\varphi h} = \frac{1}{2}(M_1 + M_2)V_\varphi^2 \cos^2\varphi$$

and one in vertical direction,

$$L_{\varphi v} = \frac{1}{2}M_1 V_\varphi^2 \sin^2\varphi$$

The total kinetic energy of the system will therefore be

$$L = L_x + L_{\varphi h} + L_{\varphi v}$$

$$= \frac{1}{2}m(x')^2 + \frac{1}{2}(M_1 + M_2\cos^2\varphi)V_\varphi^2$$

or

$$L = \frac{1}{2}m(x')^2 + \frac{1}{2}M_1(x')^2 + \frac{1}{2}M_2\cos^2\varphi(x')^2 + \frac{1}{2}(M_1 + M_2\cos^2\varphi)\rho^2(\varphi')^2 + (M_1 + M_2\cos^2\varphi)\rho x'\varphi'\cos\varphi$$

If finally M_φ is substituted for $(M_1 + M_2\cos^2\varphi)$, the total kinetic energy of the system will be represented by:

$$L = \frac{1}{2}(m + M_\varphi)(x')^2 + \frac{1}{2}M_\varphi\rho^2(\varphi')^2 + M_\varphi\rho\varphi' x'\cos\varphi$$

For Lagrange's equations the following differential quotients are required:

$$\frac{\partial L}{\partial x'}; \quad \frac{d}{dL}\left(\frac{\partial L}{\partial x'}\right); \quad \frac{\partial L}{\partial x}$$

$$\frac{\partial L}{\partial \varphi'}; \quad \frac{d}{dt}\left(\frac{\partial L}{\partial \varphi'}\right); \quad \frac{\partial L}{\partial \varphi}$$

These as developed from the equation for L are:

$$\frac{\partial L}{\partial x'} = (m + M_\varphi)(x') + M_\varphi\rho\varphi'\cos\varphi$$

$$\frac{d}{dt}\left(\frac{\partial L}{\partial x'}\right) = (m + M_\varphi)(x'') + M_\varphi'\varphi' x' + M_\varphi\rho\varphi''\cos\varphi + M_\varphi'\rho(\varphi')^2\cos\varphi - M_\varphi\rho(\varphi')^2\sin\varphi$$

$$\frac{\partial L}{\partial x} = 0$$

$$\frac{\partial L}{\partial \varphi'} = M_\varphi\rho^2\varphi' + M_\varphi\rho x'\cos\varphi$$

$$\frac{d}{dt}\left(\frac{\partial L}{\partial \dot{L}}\right) = M_\varphi \rho^2 \varphi'' + M_\varphi' \rho^2 (\varphi')^2 + M_\varphi \rho x'' \cos\varphi + M_\varphi' \rho \varphi' x' \cos\varphi - M_\varphi \rho \varphi' x' \sin\varphi$$

$$\frac{\partial L}{\partial \varphi} = \frac{1}{2}(x')^2 M_\varphi' + {}^1/_2 M_\varphi' \rho^2 (\varphi')^2 + M_\varphi' \rho \varphi' x' \cos\varphi - M_\varphi \rho \varphi' x' \sin\varphi$$

The outer forces to be introduced in Lagrange's Equations are according to Equations [1] and [2] of the paper are

$$F_x = \frac{R}{R'} P_c - R_w - R_c \quad \ldots\ldots\ldots\ldots\ldots [1]$$

and

$$F_\varphi = (P_c - Mg \sin\varphi - R_c \cos\varphi)\rho \quad \ldots\ldots\ldots [2]$$

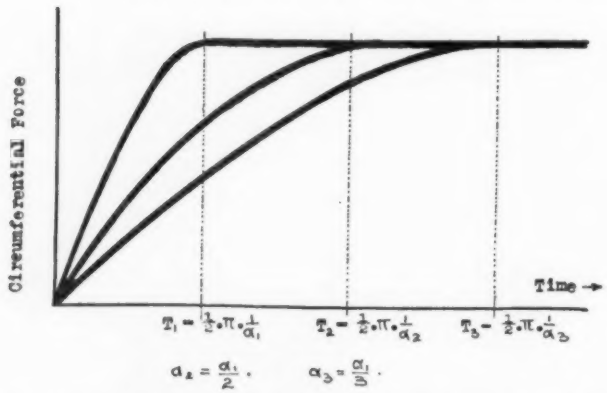


FIG. 14 CIRCUMFERENTIAL FORCE P_c AS A FUNCTION OF THE TIME FOR DIFFERENT CIRCULAR FREQUENCIES

The equations for the movement can now be established as:

$$\frac{R}{R'} P_c - R_w - R_c = (m + M_\varphi)x'' + M_\varphi \rho \varphi'' \cos\varphi - M_\varphi \rho (\varphi')^2 \sin\varphi + M_\varphi' x' \varphi' + M_\varphi' \rho (\varphi')^2 \cos\varphi \quad \ldots\ldots [3]$$

$$\rho(P_c - M_1 g \sin\varphi - R_c \cos\varphi) = M_\varphi \rho^2 \varphi'' + M_\varphi' \rho x'' \cos\varphi + M_\varphi' (\rho')^2 \rho^2 - \frac{1}{2}(x')^2 M_\varphi' - \frac{1}{2} M_\varphi' \rho^2 (\varphi')^2 \quad \ldots\ldots [4]$$

For a simpler discussion of these general equations of the movement of the system the front-wheel mass may be neglected. The equations then appear in the form:

$$\frac{R}{R'} P_c - R_w - R_c = (m + M)x'' + M_\rho \varphi'' \cos\varphi - M_\rho (\varphi')^2 \sin\varphi \quad \ldots\ldots [3']$$

$$P_c - Mg \sin\varphi - R_c \cos\varphi = M_\rho \varphi'' + Mx'' \cos\varphi \quad \ldots\ldots [4']$$

Here M stands for the mass of the weight which is supported on the axle. The equations describe the oscillations of the two-mass system, consisting of the masses m and M, where m is forced to follow a horizontal path while M is forced to keep at a constant distance from m under the influence of variable outer forces P_c, R_w, R_c, and Mg.

For the wheel at rest with $x = 0$, $x' = 0$, and $x'' = 0$, Equations [3] and [4] take the following forms:

$$\frac{R'}{R} P_c + M_\varphi \rho (\varphi')^2 \sin\varphi - M_\varphi \rho \varphi'' \cos\varphi + M_\varphi' \rho (\varphi')^2 \cos\varphi = P_p \quad \ldots\ldots [5]$$

$$M_\varphi \rho \varphi'' + M_\varphi' (\varphi')^2 \rho - \frac{1}{2\rho} M_\varphi' \rho (\varphi')^2 + M_1 g \sin\varphi = P_c - R_c \cos\varphi \quad \ldots\ldots [3]$$

To facilitate the discussion of Equations [5] and [6] the factor R_c, representing the front-end resistance, may be neglected in view of the magnitude of the other factors. Furthermore the front mass M_2 may be neglected also, which, introducing M as the mass which is supported on the axle, brings [5] and [6] to the forms:

$$\frac{R}{R'} P_c + M\rho(\varphi')^2 \sin\varphi - M\rho\varphi'' \cos\varphi = P\rho \quad \ldots\ldots [5']$$

$$M\rho\varphi'' + Mg \sin\varphi = P_c \quad \ldots\ldots [6']$$

Equation [6'], referring to the carrier system, describes a forced oscillation of a pendulum, while Equation [5'] gives the relation of the forces at the wheel center.

From Equation [5'] it can be seen that in addition to the propelling force due to the applied engine torque there are two forces existing due to the swinging carrier mass; the horizontal component of one of these acts, for starting conditions, in the direction of the propelling force, while the horizontal component of the other force is in the opposite direction. The vertical components of both forces are directed downward, thus increasing the pressure on the ground due to the weight supported by the wheel.

Equation [6'] is the equation of the forced movement of a pendulum without damping. The disturbing function P_c must be assumed according to the conditions under which the power is applied.

Equation [6'] in the form

$$M\rho\varphi'' = P_c - Mg \sin\varphi$$

could be integrated if P_c were a known function of the angle φ. For the present P_c will be assumed to be a known function of φ and will be represented by $P_c = f(\varphi)$.

The integration can now be symbolically carried through:

$$\varphi'' d\varphi = \varphi' d\varphi' = \frac{1}{M\rho}[f(\varphi)d\varphi - Mg \sin\varphi \, d\varphi]$$

and

$$\int_{\varphi_0'}^{\varphi'} \varphi' d\varphi' = \frac{1}{2}[(\varphi')^2 - (\varphi_0')^2] = \frac{1}{M\rho}\left[\int_{\varphi_0}^{\varphi} f(\varphi)d\varphi - Mg(\cos\varphi_0 - \cos\varphi)\right]$$

For $\varphi_0 = 0$ and $\varphi_0' = 0$ this equation reduces to

$$(\varphi')^2 = \frac{2}{M\rho}\left[\int_0^{\varphi} f(\varphi)d\varphi - Mg(1 - \cos\varphi)\right]$$

The left-hand member of Equation [5'] represents the total horizontal or propelling force P_p:

$$P_p = \frac{R}{R'} P_c + M\rho(\varphi')^2 \sin\varphi - M\rho\varphi'' \cos\varphi$$

Introducing the values for φ'', $(\varphi')^2$ and P_c into this equation gives the following expression for the total propelling force:

$$P_p = \frac{R}{R'} f(\varphi) + 2\left\{\int_0^{\varphi} f(\varphi)d\varphi - Mg(1 - \cos\varphi)\right\}\sin\varphi - \left\{f(\varphi) - Mg \sin\varphi\right\}\cos\varphi \quad \ldots\ldots [8]$$

In order to determine the unknown function $P_c = f(\varphi)$, it is necessary to assume a time relation of the circumferential force

TABLE 1

$\varphi = \frac{P_0}{M \cdot g \cdot (1-f^2)} \cdot [\sin(f \cdot \alpha_n \cdot t) - f \cdot \sin\alpha_n \cdot t] = 6.667 \cdot I$:

$P_0 = 5 \cdot M \cdot g$: $\alpha_n = 8.72$: $f = 1/2$: $(1-f^2) = 0.750$:

t	0.04	0.06	0.08	0.10	0.12	0.14	0.16	0.18	0.20
$\sin(1/2\alpha_n t)$	0.174	0.259	0.342	0.423	0.500	0.574	0.643	0.707	0.766
$1/2 \cdot \sin\alpha_n t$	0.171	0.250	0.322	0.383	0.433	0.469	0.493	0.500	0.493
I	0.003	0.009	0.020	0.040	0.067	0.105	0.150	0.207	0.273
$\varphi = 6.667 \cdot I$	0.020	0.060	0.133	0.266	0.447	0.700	1.000	1.380	1.840
φ (degrees)	1.1	3.5	7.5	15.2	25.6	40.2	57.3	79.2	105.5

P_c. This relation, which for the practical case on which the problem is based, represents the torque-time relation of the clutch engagement, can be described closely enough by a quarter of a sine curve, or

$$P_c = P_0 \sin(\alpha t)$$

where α represents the circular frequency and t the time.

For the following calculation the further assumption will be

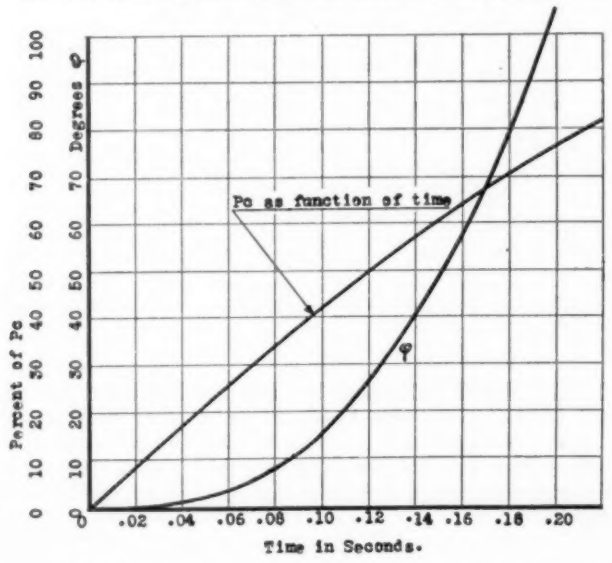


FIG. 15 P_c AND ANGLE φ AS FUNCTIONS OF TIME

made that no impact torque shall result from the clutch engagement.

Fig. 14 shows diagrammatically the different force-time relations for variable circular frequency. It is further necessary to establish a time relation for the angle φ. For this purpose Equation [6'] can be put into the following form:

$$M \frac{d^2\varphi}{dt^2} - M\varphi \frac{g}{\zeta} = \frac{P_0}{\rho} \sin(\alpha t) \ldots\ldots\ldots\ldots [9]$$

This equation is an approximation, since $\sin \varphi$ is replaced by the angle φ. This is the equation of a forced oscillation, the solution of which is given by

$$\varphi = \frac{P_0/\rho}{M(\alpha_n^2 - \alpha^2)} [\sin(\alpha t) - \frac{\alpha}{\alpha_n} \sin(\alpha_n t)] \ldots\ldots [10]^2$$

where $\alpha_n = \sqrt{g/\rho}$ = natural circular frequency. Introducing this equivalence into [10] gives:

[2] Dr. L. Geiger in "Mechanische Schwingungen."

$$\varphi = \frac{P_0}{Mg\left[1 - \left(\frac{\alpha}{\alpha_n}\right)^2\right]} \left[\sin(\alpha t) - \frac{\alpha}{\alpha_n} \sin(\alpha_n t)\right]$$

or, substituting f for α/α_n,

$$\varphi = \frac{P_0}{Mg(1-f^2)} [\sin(f\alpha_n t) - f \sin(\alpha_n t)] \ldots\ldots [11]$$

To carry the calculations further it may be best to deal with a special case, for which definite assumptions as to the dimensions and the factor f must be made. This factor represents

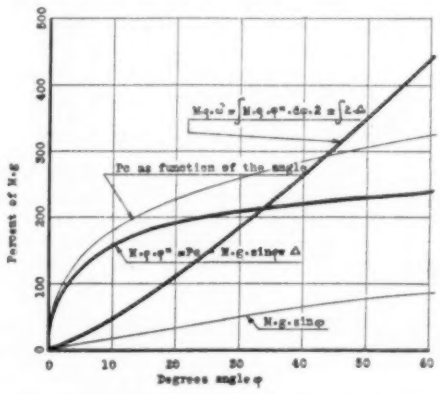


FIG. 16 DETERMINATION OF THE INERTIA FORCES ($f = 0.5 = \alpha/\alpha_n$.)

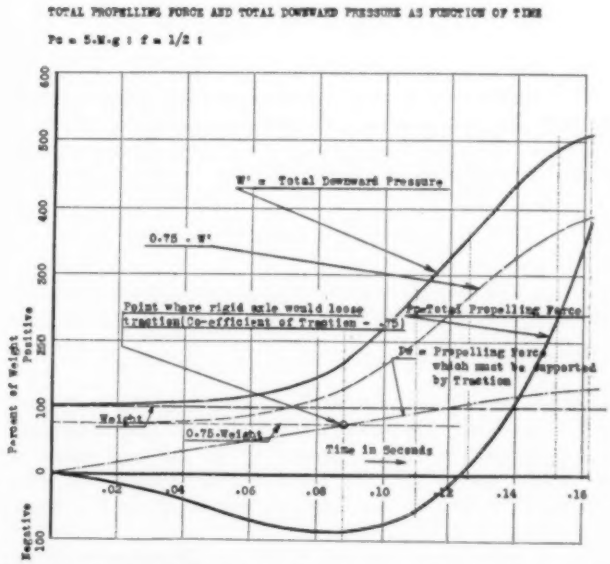


FIG. 17 TOTAL PROPELLING FORCE AND TOTAL DOWNWARD PRESSURE AS FUNCTIONS OF TIME ($P_c = 5Mg$; $f = 0.5$.)

the ratio of the imposed circular frequency to the natural circular frequency.

In the following calculations f is given the value 0.5, i.e., the harmonic application of torque to the mechanism is one-half the natural period of the pendulum. It is further assumed that $\rho = 5.18$ in. and $P_0 = 5\,Mg$. With these assumptions different values for the angle φ in relation to the time have been calculated and are given in Table 1.

The values of the angle φ as a function of time are plotted in Fig. 15, as well as the corresponding values of P_c. With the re-

TABLE 2

φ - Degrees	10	20	30	40	50	60
Pa	75.0	93.0	105.0	115.0	123.0	130.0
$M\cdot\rho\cdot\varphi''$	68.7	79.8	85.0	89.2	92.5	95.0
$M\cdot\rho\cdot\omega^2$	14.2	42.9	71.4	102.3	132.8	166.0
$M\cdot\rho\cdot\omega^2\cdot\sin\varphi$	2.5	14.7	35.7	65.7	01.8	143.8
$M\cdot\rho\cdot\varphi''\cdot\cos\varphi$	-67.5	-75.0	-73.6	-68.5	-59.5	-47.5
$Pa\cdot R/R'$	30.0	37.2	42.0	46.0	49.2	52.0
Pp	-35.0	-23.2	+ 4.1	+43.4	+91.5	+148.3
Pp in % of W	-87.5	-57.8	10.3	108.5	228.5	371.0
$M\cdot\rho\cdot\varphi''\cdot\sin\varphi$	11.9	27.3	42.5	57.4	70.8	82.4
$M\cdot\rho\cdot\omega^2\cdot\cos\varphi$	14.0	40.3	61.8	78.3	85.5	83.0
W	40.0	40.0	40.0	40.0	40.0	40.0
W'	65.9	107.6	144.3	175.7	196.3	205.4
W' in % of W	165.0	269.0	361.0	439.0	491.0	513.0
Time-Seconds.	.088	.110	.126	.140	.152	.162

sults obtained thus far it is possible to represent the circumferential force P_c as function of the angle. This is done graphically in Fig. 16.

With P_c as a known function of the angle φ it is possible to solve the integral in Equation [8]. The two inertia forces acting at the wheel center can be expressed as:

$$I_1 = M\rho(\varphi')^2; \quad I_2 = M\rho\varphi''$$

The calculation of these inertia forces under the previously stated assumption is carried through graphically, Fig. 16 being also used for this purpose. The total propelling force P_p is made up of the propelling force due to the engine torque and the horizontal components of the inertia forces:

$$P_p = \frac{R}{R'} P_c + M\rho(\varphi')^2 \sin\varphi - M\rho\varphi''\cos\varphi$$

The total downward pressure also comprises three members: the weight and the vertical components of the inertia forces; or

$$W' = Mg + M\rho(\varphi')^2 \cos\varphi + M\rho\varphi'' \sin\varphi$$

Under the assumption that R/R', the ratio of the track-gear pitch radius to the wheel radius, is 0.4, the total propelling force P_p and the total downward pressure W' for different values of the angle ρ are as given in Table 2. In figuring the total downward pressure in this table the weight Mg of the wheel assembly has been neglected.

Fig. 17 presents the values in Table 2 in diagrammatic form, and discloses fully the effect of the oscillating axle with power drive for the conditions previously assumed. The force P_w must be supported by the traction of the wheel. Assuming a tractor coefficient of 0.75, it is apparent from the diagram that in the case of the rigid axle, where only the full weight is available for traction, the force P_w becomes greater than the traction long before the full torque is applied and therefore the wheel will slide. In the case of the oscillating axle, however, where the total downward pressure W' is available for traction, the force P_w never reaches the limiting value and the wheel will not slide.

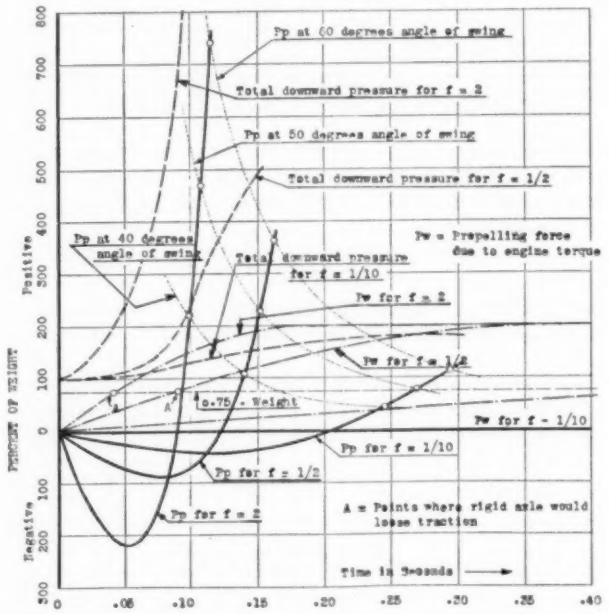


FIG. 18 TOTAL PROPELLING FORCE AND TOTAL DOWNWARD PRESSURE FOR DIFFERENT CLUTCH ENGAGEMENTS AS FUNCTIONS OF TIME

($P_c = 5Mg$; $f = a/\alpha_n$.)

Fig. 17 reveals furthermore the interesting fact that while at first the propelling force is negative, with an increased angle it later has positive values. The inclusion of the normal propelling force, i.e., the propelling force of a vehicle with rigid axle and drive, makes apparent the increased propelling force in starting.

Fig. 18 shows the total propelling force and total downward pressure for the same assumed vehicle based on different values of the factor f, and therefore for different clutch engagements.

For the preceding analysis it has been assumed that the necessary relation of the planetary drive, as expressed by the formula

$$\omega_p r = \omega_c \rho$$

is fulfilled, which will be true in practically all cases. Only in the event of extremely great reductions in the power line would the relation of the angular velocities influence the inertia forces at the wheel center.

APM-52-7

Crankless Reciprocating Machines

By ANTHONY G. M. MICHELL,[1] NEW YORK, N. Y.

The general features and properties of the crankless type of reciprocating machine, as compared with the crank-connecting-rod type, are described and illustrated. A few varieties of the crankless type are briefly described, including the author's slant-slipper form of the mechanism. The limitations of the crank type are pointed out and it is suggested that mechanical science is due to supply a type of machine for various present and probable future applications not efficiently served by either purely rotary machines or reciprocating machines of the crank type. A case is presented for the crankless type as satisfying this need.



WHILE the name "Crankless Reciprocating Machines" is not a quite accurate title for the subject matter of this paper, it has to be used in the absence of a better. It is intended to cover both prime movers and driven machines in which the reciprocating motion of pistons is interconverted with the rotary motion of a shaft without the use of cranks. But according to common and convenient usage the term is restricted to machines in which the motion of the pistons is parallel to the axis of the shaft.

The Crank and Crankless Types of Engines

The most fundamental distinction between a crankless engine, so defined, and a crank engine of the familiar type, is that the crank engine has a two-dimensional mechanism, while the crankless engine has a three-dimensional mechanism. That is to say, the essential parts of the crank engine move only in one or more planes at right angles to the axis of the engine shaft. In the crankless engine, on the contrary, motions of essential parts take place in all three of the coordinate directions of space. Another broad and significant difference in the two types of mechanism is seen when we compare multicylinder machines of each class. The typical crankless engine is an essentially symmetrical construction. Each cylinder, or piston, is geometrically indistinguishable from its fellows as regards its relation to them and to the mechanism. For this reason, crankless engines are sometimes referred to as "round engines" or "barrel engines." The crank-connecting-rod engine on the contrary is essentially a linear, or serial, arrangement, and has at most a bilateral symmetry. These differences are immediately apparent on comparison of Figs. 1 and 2, which show respectively the essential moving parts of a typical multicylinder crank-connecting-rod engine and of a crankless engine with the same number of pistons, namely, eight in each case.

We can, and often do, speak in the case of the crank-connecting-rod engine (Fig. 1) of the first, second, or third piston, or the nth piston from the front or rear end. The motion of any such piston is independent of that of each of the others in so far as it may be given any desired phase relation to those of the others by setting the crank corresponding to that particular piston at an appropriate angle.

In the typical crankless engine, on the contrary (Fig. 2), the several pistons are as truly indistinguishable from each other as the several blades of a turbine wheel, and the sequence and phases of the motions of the several pistons are unalterably determined.

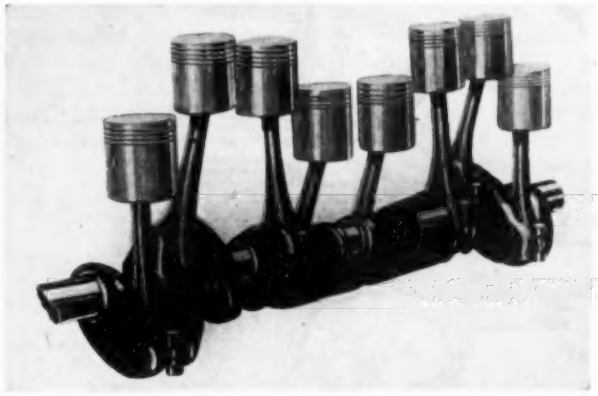


Fig. 1 Essential Moving Parts of a Typical Multicylinder Crank-Connecting-Rod Engine

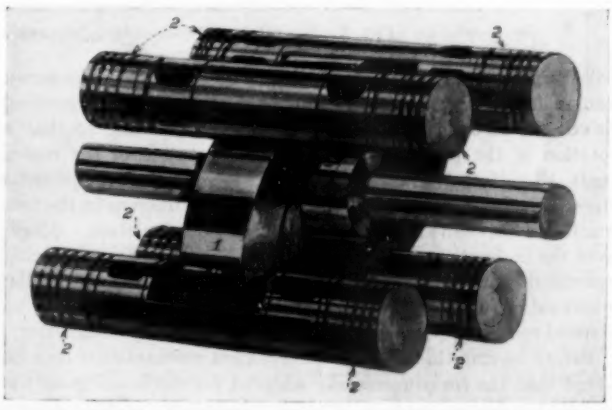


Fig. 2 Essential Moving Parts of a Crankless Engine Having the Same Number of Pistons as the Engine Shown in Fig. 1

Figs. 3, 4, and 5 show respectively three crankless machines, and illustrate three typical forms of crankless construction.

Numerous as are the forms of the crankless mechanism which have been proposed, designed, and patented, many of them extremely ingenious and interesting, these three illustrations cover, so far as the author knows, the only three species of crankless engines which are, or have been, in successful use.

The Ball-Race Type of Crankless Mechanism

Fig. 3 was the first design of the three to be perfected. It represents the well-known "Janney," "Williams," or "Waterbury" transmission mechanism.

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Presented at a meeting of the Metropolitan Section of the A.S.M.E., New York, April 19, 1929.

Substantially the same machine serves either as a pump or as a prime mover. Regarded as a pump, the rotating shaft 12 carries with it the ring 20, and the stationary inclined ball race 21, upon which the ring 20 rotates, compels the ring to travel in an inclined plane relatively to the shaft. Thus the outer ends of the connecting rods 19 are brought alternately in each revolution nearer to and farther away from the cylinder barrel 15, which comprises an annular series of cylinders such as 15a, containing corresponding pistons such as 18, each attached to the upper end of a rod 19. During one half of the rotation around the race 21, the pistons move outward and draw liquid into the cylinders, while during the other half they move inward and discharge the liquid under pressure, suitable valve action being assumed.

An almost exactly similar mechanism, which indeed may form a twin and attached counterpart of the pump, as at 11 in Fig. 3,

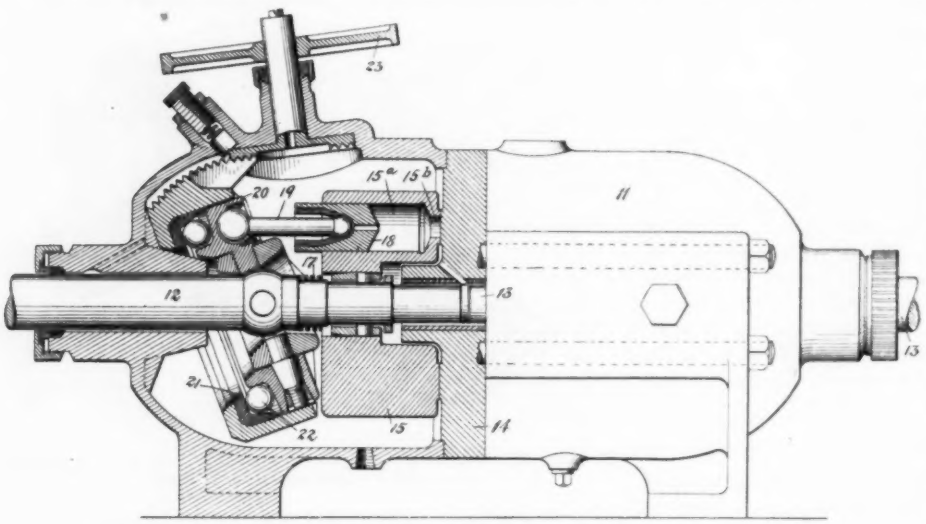


FIG. 3 BALL-RACE TYPE OF CRANKLESS MECHANISM

will function as a motor. The delivery and intakes of the pump and motor being reciprocally interconnected, the first machine drives the second through the medium of the fluid, so that a rotation of the pump shaft 12 causes a rotation of the motor shaft 13. If the pump and motor cylinders are of different sizes, or if the inclinations of the races 21 are different in the two machines, the shafts may have any desired speed relation. Moreover the inclination of the race 21 may be changed, even during operation, with an instantaneous corresponding change of the speed ratio, so that the combined machine serves not merely as a speed-reducing, but as a variable speed-reducing, transmitter.

Before leaving this form of the crankless mechanism it may be noted that the form necessarily adopted for the bearings at the ends of the connecting rods 19, while excellent for exerting a compression in the rod to resist the pressure in the fluid, would be quite unsuitable if there were considerable tension in the rods, such as would arise from inertia effects if the speed were high. The ball race 21, which is necessarily of a considerable diameter, is also only suitable for comparatively low speeds of rotation. This form of mechanism is therefore only suitable for machines operating at relatively low speeds.

THE WABBLE-PLATE TYPE OF CRANKLESS MECHANISM

Another practical form of the crankless mechanism is that shown in Fig. 4, the machine illustrated being the "Ali" two-stroke gasoline engine in use as an "outboard" boat engine. Allied forms go under the name of "Arvad engine" and other names. The cylinders 1, which contain the pistons 2 reciprocating in the vertical direction, are in this case stationary while the shaft 3, also vertical, revolves. Each piston is rigidly connected with a scavenge-air piston 4, working in the cylinder 5. Between the power cylinders and the pump cylinders lies the plate 6, which merely "wabbles" or nutates but does not rotate. A ball member 7, fixed to the plate, is socketed in the block 8 which is mounted in the piston unit and is capable of a small extent of transverse motion with respect to it. Thus the nutating motion of the plate corresponds with reciprocating motions of the various pistons in successive phases. Now the plate 6 is mounted by means of a pair of ball bearings, of the combined thrust and journal type, on a flanged collar mounted obliquely on the shaft. Thus as the shaft rotates, the plate, which is prevented from rotating by the ball-and-socket attachments to the pistons, is compelled to execute its nutating motion and so causes the pistons to reciprocate. Conversely, reciprocation of the pistons, compelled to take place in due sequence by their engagement with the plate, causes the latter to nutate and the shaft to revolve. The somewhat unfamiliar type of motion of the nutating plate can be readily followed and calculated when it is observed that its instantaneous motion at every instant is a rotation about an axis which is radially at right angles to the axis of the shaft, and in the same radial plane as the line of greatest inclination of the plate.

The angular velocity of rotation about this axis is, in fact, $\omega \tan \alpha$, where ω is the angular speed of the engine shaft, and α the inclination of the plate.

This nutating motion of the plate with respect to the pistons and frame of the engine may be called a uniform conical nutation, and is of course a three-dimensional motion. Relatively to the shaft, however, it is merely a pure rotation about an axis inclined to the shaft.

It is clear that the point of intersection of the axis of each piston with the central plane of the nutating plate reciprocates with a pure sinusoidal or harmonic motion. The piston itself, however, does not reciprocate in unison with this point of intersection, but with the axial coordinate of the center of the ball 7, which has an arcual and not a linear motion. The motion of the piston therefore deviates somewhat from a true harmonic motion.

Similarly in the Waterbury mechanism, since the radial distance from the shaft axis to the center of the outer ball ends of the connecting rods 19 varies as they traverse their inclined path, the rods 19 take up progressively varying inclinations to the shaft, and the reciprocating motion of the pistons differs somewhat from the true harmonic.

It will be noticed that in both the Ali and the Janney mechanisms there are three relatively moving parts interposed between the shaft and each piston: namely, in the Ali mechanism the ball race, the nutating plate, and the sliding block; and in the Janney mechanism, the ball race, the oblique plate, and the connecting rod. In the Ali engine the ball race is relatively of much smaller diameter than in the Janney gear, and the Ali sliding block, unlike the Janney connecting rod, is equally adapted to carry loads in both directions. The Ali engine is therefore better adapted to run at the high speeds of a multicylinder internal combustion engine than is the Janney mechanism. The load

limitations of the ball bearing in the Ali mechanism, however, seem to limit its use to engines of comparatively small size. Within its limits of application its mechanical efficiency should be very high.

The Slant-Slipper Type of Crankless Engine

The third of the three types of crankless engines which have been mentioned as having reached the stage of actual application is that which the author's own associates have developed.

Fig. 2, as already shown, illustrates one example of it. An outline view of the mechanism is shown in Fig. 5. From this

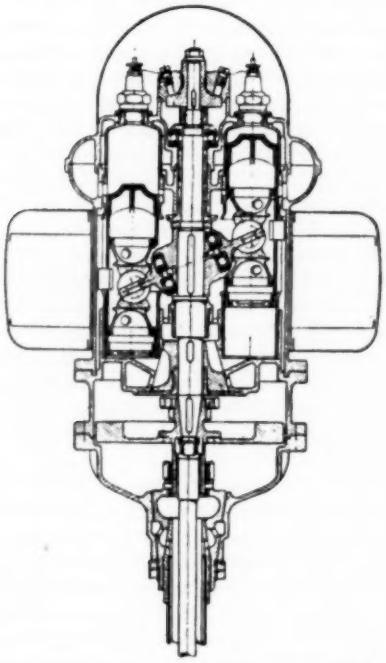


Fig. 4 "Wabble"-Plate Type of Crankless Mechanism

view it will be seen that the shaft has rigidly and permanently mounted upon it an oblique, plane-sided plate 1, which is called the "slant." It has been given this distinctive name since it is not a nutating plate, or "wabble" plate, like the oblique plates in the other two types. It has no other motion than a rotation in common with the shaft. From the slant, motion is transmitted to each of the pistons 2 by a slipper element 3, which serves the purposes of all three of the intermediate elements in the Ali, or in the Janney, mechanism.

Three views of a typical slipper are shown in Fig. 6. It has two bearing faces: a spherical face on one side and a flat face on the other. The spherical face fits into a socket formed in the piston. The flat face makes sliding contact with the flat face of the slant. By means of its spherical support in the piston, the slipper is enabled to follow the continually varying aspects of the inclined face of the slant as the latter revolves; the plane face of the slipper effecting, in fact, a uniform nutating motion of exactly the same kind as the motion of the nutating plate of the Ali engine. During its motion the center of the ball joint remains at a fixed point in the axis of the piston, and at a constant axial distance from the plane surface of the slant. The reciprocating motion of the piston is consequently truly sinusoidal or harmonic.

When in a multicylinder engine several such piston elements are arranged at equal angular intervals around the slant, there is a constant difference of phase between the motion of each piston and that of each of its neighbors.

Usually the piston elements are double, as seen in Fig. 7, pistons on each side of the slant having a common axis and being rigidly connected together by means of a bridge piece 4. A slipper 3 is socketed in each piston so as to effect both strokes of the complete reciprocating unit.

The success of such a mechanism evidently depends on the possibility of the relative sliding motion of the slant and slippers taking place with a minimum amount of friction. Particularly, if the pistons are to drive the shaft, the mechanism being employed as a motor, one recognizes instinctively that the machine would fail to run, or run very inefficiently, if the coefficient of friction of the slipper and plate had ordinary, every-day values.

The solution of the problem lies in the application of the Reynolds theory of lubrication, and the ball-socketed form of slipper which already has been described effects the application. As shown in Fig. 6, the ball is located unsymmetrically with respect to the flat face, to the extent of about one-tenth of the length of the slipper in the direction of the relative sliding motion of the slant. The effective point of application of the load on the slipper is of course the center of the ball. The slipper is thus enabled to function in the same way as the pads of the pivoted type of thrust bearings, which are well known in this country as Kingsbury bearings and which bear the name of the author in Europe. The coacting surfaces require of course an ample supply of lubricant of a viscosity suitable for the

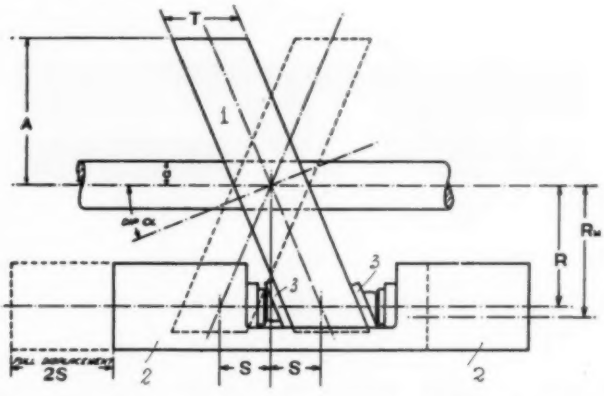


Fig. 5 Slant-Slipper Type of Crankless Mechanism Shown in Outline

ruling conditions of load and speed. With such provision, experience confirms calculation as showing that the coefficient of sliding friction can be extremely low, 0.0015 or 0.002 being usual values. As a result, the mechanism is enabled to operate with quite high efficiency and with practically the same efficiency, whether utilized as a motor or as a pump. The mechanism is regarded as operating as a motor when power applied to the pistons is transmitted through the slant-and-slipper mechanism to rotate the shaft, and as operating as a pump when power applied to rotate the shaft is transmitted through it to cause reciprocating motion of the pistons. Whichever is the direction of power transmission, some energy is of course lost in the frictional resistances of the mechanism; but if the angle of inclination of the slant to the shaft is not too nearly a right angle, such loss of energy in the crankless mechanism is small, and nearly the same in both cases. The crankless mechanism thus conforms to the rule that an efficient mechanism is reversible or, rather, is interconvertible, as motor or "movend;" and conversely, the practical demonstration that one of these machines is in this sense readily reversible is proof that it is mechanically efficient.

To illustrate by symbols: If μ is the coefficient of friction, and θ the inclination of an element of the slant over which sliding is

momentarily taking place, the efficiency operating as a motor is readily shown to be $E_M = 1 - \mu (\tan \theta + \operatorname{ctn} \theta)$; while the efficiency as a pump is

$$E_p = \frac{1}{1 + \mu (\tan \theta + \operatorname{ctn} \theta)}$$

Expanding the latter expression as

$$E_p = 1 - \mu (\tan \theta + \operatorname{ctn} \theta) + \mu^2 (\tan \theta + \operatorname{ctn} \theta)^2 - \mu^3 (\tan \theta + \operatorname{ctn} \theta)^3$$

we see that the efficiency as a motor is less than that as a pump by no more than the fraction

$$[\mu(\tan \theta + \operatorname{ctn} \theta)]^2$$

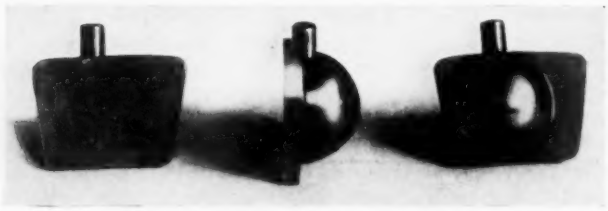


FIG. 6 THREE VIEWS OF A TYPICAL SLIPPER

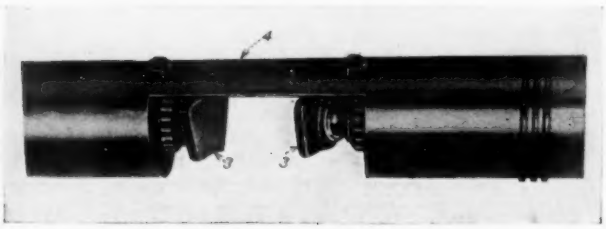


FIG. 7 DOUBLE PISTON ELEMENT OF SLANT-SLIPPER MECHANISM

If, for instance, $\tan \theta$ has the quite usual average value $^1/_4$,

$$\tan \theta + \operatorname{ctn} \theta = 4.25$$

$$(\tan \theta + \operatorname{ctn} \theta)^2 = 18$$

and if $\mu = 0.002$,

$$[\mu(\tan \theta + \operatorname{ctn} \theta)]^2 = (0.0085)^2 = 0.000072$$

or less than $^1/_{100}$ of 1 per cent, as the difference of the efficiencies in favor of the pump.

NEED OF THE CRANKLESS TYPE OF MACHINE

With such high efficiency, which has been amply realized in practice, and with the simplicity and compactness of its construction, it is to be supposed that the crankless type may be usefully applicable to a wide range of purposes. Clearly its merits will be most apparent in the case of multicylinder machines, which internal-combustion engines tend more and more to become. What actual advantages it possesses over the types of engines which at present practically hold the field, and in what applications those advantages may be most likely to secure its adoption, the author proposes to set forth by reviewing some special features of the crank engine.

DEFECTS OF THE CRANK TYPE OF ENGINE

Our familiarity with the crank type, the author believes, is apt to make us oblivious of the real nature of some of its characteristics, and perhaps unduly tolerant of some of its limitations.

In the first place, being essentially a two-dimension machine, the crank engine does not make efficient use of our space of three dimensions. Being neither symmetrical nor compact, it is apt to be unduly extensive, at least in some of its dimensions.

Furthermore, the inherent characteristics of the typical crank engine, namely, that it has no circumferential symmetry and that the motion of its pistons is not harmonic, render it impossible to make it free from vibration by giving it complete dynamic balance; and balance is surely an elementary virtue which the modern engineer, brought up on turbines and electric machines and demanding ever higher rotative speeds, very properly expects every rightly constituted machine to have.

Proceeding to the concrete, and to questions of construction, let us take the essential and characteristic organ of the crank engine, namely, the crankshaft. In an 8-cylinder crankshaft, as shown in Fig. 1, we have a member which is bent at right angles in 32 places. It has 8 crankpin bearings and, in this instance, 5 main bearings. Quite commonly it would have 9 main bearings, or 17 separate bearings in all. Such a shaft in a typical case requires about 160 distinct dimensions to describe it, all of which have to be adhered to with accuracy in all the operations of manufacture. If each of the journals is 3 in. in diameter and 3 in. wide, there will be about 500 sq. in. of bearing surfaces to be machined on these 17 journals, and an equal area of surface to be machined and fitted on the 17 corresponding bearings. This is more than three times the area that has to be machined on the parts performing the corresponding functions in the Michell crankless engine; namely, the slant's two faces, two journals, a thrust collar, and the corresponding bearings and slippers, 18 surfaces in all as compared with 34.

Regarded elastically, the elongated 8-cylinder crankshaft is obviously incapable without the support of its intermediate bearing of supporting the inertia forces of the reciprocating and revolving parts which load it transversely. These forces have therefore to be balanced individually as far as possible (which is to the extent of about 50 per cent), by adding adventitious masses to each crank; and the forces remaining unbalanced in each unit are transmitted through the bearings to the engine frame, to equilibrate as nearly as possible the resultants of corresponding stresses arising from the other units. In so doing, heavy stresses and vibrations are necessarily set up throughout the unit.

While it is true, therefore, that a 6- or 8-cylinder crank engine can be very approximately balanced as a whole, it is only at the cost of much added weight in balance weights, bearings, and frame, and complications to the design, which are entirely unnecessary in the typical crankless engine. What has been said as to the multicylinder crankshaft might be repeated, *mutatis mutandis*, with regard to the connecting rods, to the accommodation of which, by the way, the crank engine usually owes about one-third of its bulk.

Let us proceed, however, to a more vital element, the piston. It is for the sake of the piston, of course, and its unique capacity, with its sealing rings, for compressing in a cylinder a volume of fluid under pressure, that we have reciprocating engines. Turbine rotors will utilize the energy of expanding or falling fluid, with all desirable efficiency and with admirable convenience; but no efficient rotary or other substitute has been found, or seems likely to be found, for the cylindrical piston and its sealing rings, as a compressing agent. Efficient compression is an essential condition of successful operation of the internal-combustion engine, and it is the failure of the gas turbine in this respect which preserves the internal-combustion field for the reciprocating engine. One of the worst of the defects of the crank type is that it does not deal fairly with, or develop the full potentialities of, the piston, especially the piston of the internal-combustion engine.

The secret of the success of the cylindrical piston is, of course, its cylindricality, or the complete symmetry of the piston and the cylinder around their common axis; but in its two-dimensional motion transverse to the crankshaft, the connecting rods apply forces in one diametral plane only of the piston, and this to large extent destroys its symmetry. Contact of piston and cylinder takes place only along the two generating lines in the plane of the connecting rod, from which follows wear of piston and cylinder; while on other portions of the circumference the relatively considerable interspace remains filled with oil or gas, which effectually prevents the conduction of heat from piston to cylinder wall which is an essential function in the internal-combustion engine. An overheated and an unsymmetrically heated and consequently deformed piston is the result, unless the troublesome expedient—almost impracticable except in large engines—of fluid circulation for cooling the piston is resorted to.

In the typical crankless engine, on the contrary, as shown in Fig. 2, since the obliquity of the slant is turned in succession to all azimuths around the axis, every generating line of the piston makes contact at some phase of the cycle with the corresponding generating line of the cylinder. Fig. 8 shows the sequence of the directions of these contact pressures in a particular 4-stroke, 8-cylinder crankless engine, the radial distances to the curves at each azimuth being proportional to the radial pressure between piston and cylinder. This action makes the lubricant in the cylinders effective for reduction of friction, and it also effects symmetrical cooling of the whole piston wall.

In case it may be suggested that the mere "line contact," as it may be called, would not be effective for transmission of the heat, it may be pointed out that in the case of a piston having $^{5}/_{1000}$ in. diametral clearance in its cylinder, the clearance space is less than $^{1}/_{10,000}$ in. wide throughout an arc of more than 30 deg., and that the resistance of a film of oil of that thickness is no greater than that of $^{1}/_{8}$ in. of metal.

Thus while it is generally agreed that in the conventional internal-combustion engine only the gas rings of the piston are effective for conducting heat from the piston to the cylinder walls, in the crankless engine the whole piston barrel is effective for the purpose. In thus fully utilizing the virtues of the piston, the crankless type admits of higher compression pressures and temperatures, and higher piston speeds, just as its perfect balance admits of higher rotative speeds than the crank type.

Fields of Application of the Crankless Type of Engine

It is believed, therefore, that we can confidently expect a future, especially in the field of the internal-combustion engine, for the crankless type in whichever of its present or future forms may prove to be the best. Its success will come first where the defect and limitations of the crank type hinder its further progress. And the author ventures to suggest that there are clear indications, in several directions, of important new fields which the crank type cannot satisfactorily fill.

Diesel Locomotive. First, there is the oil-engine locomotive for main-line service. Very many attempts have been made with the crank type for the past 15 or 20 years. In the most recent examples 1200 to 1500 hp. has been reached, but the crank engine of this power attains the limit of height of most railway loading gages, and the weight and cost of the structure prevent economic competition with the steam locomotive.

The author visualizes the oil-engine locomotive of the future, of any required power, with a crankless engine of approximately the same size and shape and in much the same location on the locomotive as the present steam boiler, coupled to an electric generator in roughly the same location as the present firebox. The higher speeds which are acceptable for the crankless type will enable the weight and cost to be kept within reasonable limits.

Diesel Aero-Engine. Another field of even greater importance which he sees marked off for the crankless engine of the future is that of the heavy-oil engine for flying machines. The disabilities of the crank type of engine for this application are its inadaptability to the two-stroke cycle—at least without a second crankshaft—its excessive height or other dimension transverse to the shaft, its relatively great frontal area in all its lighter forms, and its deterrent weight. It is true that remarkable progress has lately been made in the reduction of weight of crank engines for this purpose, but whatever success is achieved in that direction will enable crankless engines to be constructed by the same methods at least 30 per cent lighter, and in this field the demand for the lightest possible engine will always be insistent.

Twelve-Cylinder Automobile Engine. Looking in another less lofty direction, if the fashion in automobilism should trend to a demand for an engine with twelve or more cylinders, as seems very probable in view of the diminution of torque fluctuation—which is now the most serious cause of vibration and unsteady running in automobiles of the best class—the crankless type must apparently be adopted. A 12-crank shaft can scarcely be contemplated with equanimity, and no other crank type than the straight-line type seems to be acceptable. A 12- or even a 16-cylinder crankless engine presents no difficulties, and would be no longer than an 8-cylinder engine.

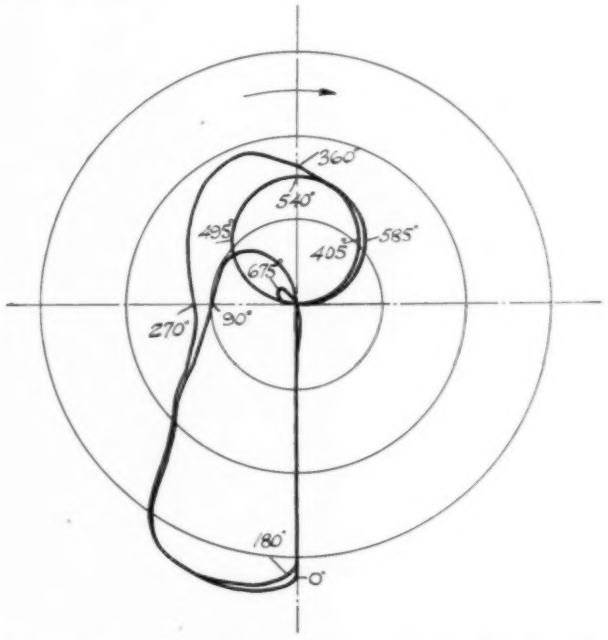


Fig. 8 Showing Sequence of Directions of Contact Pressures in a Particular 4-Stroke 8-Cylinder Crankless Engine

Compressors for Large Volumes. There are, the author thinks, numerous other fields in which the progress of internal-combustion engineering depends upon the adoption of an engine mechanism different from the crank and connecting-rod, but of those he will mention only one more. It is the field of compressors for very large volumes of air and gas required to be delivered at moderate or high pressures; that is, pressures from 5 lb. per sq. in. upward and volumes of several or many thousands of cubic feet per minute. Considerations of space occupied and the convenience of direct-electric driving demand that the machine shall have a high rotational speed.

The turbine and centrifugal types are negatived not only by their poor efficiency at any but low pressures, but by their inflexible control characteristics.

APM-52-8

Engineering Computations for Air and Gases

By SANFORD A. MOSS[1] and CHESTER W. SMITH,[2] LYNN, MASS.

In this paper the authors give methods of making the usual engineering computations for adiabatic compression, expansion, flow through measuring nozzles, and similar work concerned with the thermodynamic properties of air and gases, in the minimum amount of time and with the greatest accuracy possible with engineering methods. The theoretical formulas needed are included, together with extensive tables for use in computation.

THE object of this paper is to give methods of making the usual engineering computations for adiabatic compression, expansion, flow through measuring nozzles, and similar work concerned with the thermodynamic properties of air and gases, in the minimum amount of time and with the greatest accuracy possible with engineering methods.

In making such computations it is amply sufficient for engineering purposes to use the perfect-gas laws. This gives a well-known set of engineering formulas involving various constants, such as c_p and standard density. The values of these constants for air and other gases as given by different experimenters vary somewhat, and are usually thermodynamically inconsistent. Furthermore the constants as usually given apply to theoretically dry air or gas, which seldom exists for the engineer. It is of course just as easy to carry out computations with one set of constants as another. There is no good reason why a reputable, self-consistent set of constants should not be selected and used universally for ordinary humidity. The present common practice is the use by every engineer of a different set of constants, often mutually inconsistent, selected from an old edition of an engineering handbook which he happens to have in his desk. Blind faith in a set of constants which may have no justification except that an engineer is accustomed to them, will probably be the principal objection to adoption of the constants selected as reputable in the present paper.

The humidity of air in a compressor or flowing through a measuring nozzle varies from day to day with atmospheric conditions. Furthermore the specific heat and other properties of air vary slightly with temperature and with the amount of CO_2. However, most engineering work does not warrant exact measurement of humidity, or variation in the fundamental constants to take account of the change in the humidity from day to day, the change of specific heat with temperature, or the CO_2 content. A reputable set of constants is therefore selected for air with average temperature and CO_2 and with a usual humidity, which is here called "Normal Air." This "Normal Air" of the present paper is much more nearly representative of the air actually passing through a compressor or measuring nozzle than the dry air without CO_2 at 0 deg. cent., whose properties as obtained 60 years ago are often used by engineers.

Computations for the compression and flow of a perfect gas make a great deal of use of various exponential functions of pressure and temperature ratios, which are difficult to handle numerically. The theoretical formulas have therefore been recast by various authorities, and tables suited to these systems used for computations. The present paper selects what is believed to be the best system of the kind. This uses, as the most convenient function,

$$Y = \left(\frac{p_1}{p_2}\right)^{\frac{k-1}{k}} - 1$$

where $k = c_p/c_v$.

The paper gives a complete set of the commonly used thermodynamic expressions for air and gases, in terms of Y. An extensive numerical table is then given so that Y may be found for any case, on the basis of the selected reputable constants for Normal Air or perfect diatomic gases. This gives a complete system for computations in a convenient way which yields results somewhat more accurate on the average than is otherwise possible. No deductions are given for the various thermodynamic formulas, as these may be found in most thermodynamic textbooks.

The tables and values of constants are given with enough places to insure mathematical accuracy, so that checks with different forms of formula will agree, and so that the only uncertainties in computed values will be due to original observations, none being introduced by the computation work. For a great deal of engineering work, some of the last places in the constants and tables may be omitted.

Selection of Fundamental Constants. The fundamental constants often used for thermodynamic computations of air are based on values of c_p/c_v or k, obtained by direct experiments, which vary between 1.40 to 1.41. The corresponding values of $(k-1)/k$ vary from 0.286 to 0.291. These values are presumably for theoretically dry air. Such values and good experimental values of density do not exactly satisfy the fundamental relation for perfect gas, namely, $c_p - c_v = R/J$. This gives a discrepancy in the consequent relations,

$$Jc_pT(k-1)/k = 144\,pv$$

or

$$c_p = \frac{144\,p_0}{JT_0\rho_0(k-1)/k}$$

where ρ_0 is the density at p_0 and T_0. It follows that numerical results obtained from various combinations of thermodynamic equations, some of which have been transformed by means of the above relations, will not be self-consistent. A complete system for the selection of constants for Normal Air which avoided all such disabilities was used in a paper by one of the authors, "Measurement of Flow of Air and Gas With Nozzles,"[3] to which reference must be made for details. A brief résumé of this paper is as follows:

The start is made with a value of the density of atmospheric air at standard conditions with usual moisture, here called "Normal Air," which has been found satisfactory in ventilating work. This is 0.075 lb. per cu. ft., at 14.7 lb. per sq. in. and 68 deg. fahr. It is shown in the paper referred to that if there are used the best values of the densities of the various constituents of air, including the rare ones, or the values of the density of pure air (including the rare gases) with correction for the usual CO_2 content, the above figure for the density of Normal Air gives a humidity of about 36 per cent. This is a reasonable figure, so that the value of 0.075 lb. cu. ft. at 14.7 lb. and 68

[1] Thomson Research Laboratory, General Electric Co. Mem. A.S.M.E.

[2] Thomson Research Laboratory, General Electric Co.

Presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] S. A. Moss, Trans. A.S.M.E., vol. 50 (1928), no. 2, paper no. APM-3.

deg. fairly represents air with usual humidity and usual CO_2 content, which we call Normal Air. It is further shown in the paper referred to that a reputable experimental value for c_p for Normal Air, which as shown has 36 per cent humidity, is 0.243. This is higher than old values, because of Swann's experimental work by the "continuous flow" method (which here as usual, shows slightly higher values than by other methods, but which must be given due weight), because of the moisture content, and because of allowance for the slight increase of c_p with the temperatures at which air usually occurs in engineering work. Fig. 1 shows the various values, the data being obtained from "The Specific Heats of Gases," by Partington and Shilling. The authors of course take the responsibility for the weight given to the work of the various experimenters in selecting the value here used. As mentioned, the thermodynamic relations given above must be exactly satisfied by the fundamental constants used in order that numerical results obtained by equivalent thermodynamic formulas shall be consistent. Using the selected reputable values of c_p and density for Normal Air in these relations gives a value

$$(k-1)/k = 0.283$$

The exact value corresponding to the value $c_p = 0.243$ has more decimal places (0.28296), but one is not justified by the accuracy of the other work in using more than three places. The value 0.283 is therefore selected as being the exact fundamental basis of all of the work and the tables for Y computed from it. (This makes it necessary, in order to have exact consistency in any numerical work which demands it, to use a value of $c_p = 0.242967$. It is of course absurd to use so many places unless it is sought to show exact numerical identity of values computed by different methods.) A value of $(k-1)/k$ of 0.29 is often used, but the value here given of 0.283 is better for Normal Air. It leads to a value of $k = 1.3947$, which is better for Normal Air than the values 1.40 to 1.41 already referred to.

The use of 0.283 instead of 0.29 gives a difference in computed values of power for compression, etc., which, however, is completely justified by the data given herein.

The values of k and $(k-1)/k$ thus computed for Normal Air should also be practically correct for any perfect diatomic gas, and comparison with experimental values given in Partington and Shilling's work indicates that this is the case. The actual values by different experimenters differ appreciably, but reputable results are obtained by maintaining the fundamental values of the present paper for all nearly perfect diatomic gases with the assumption that the perfect-gas laws hold exactly.

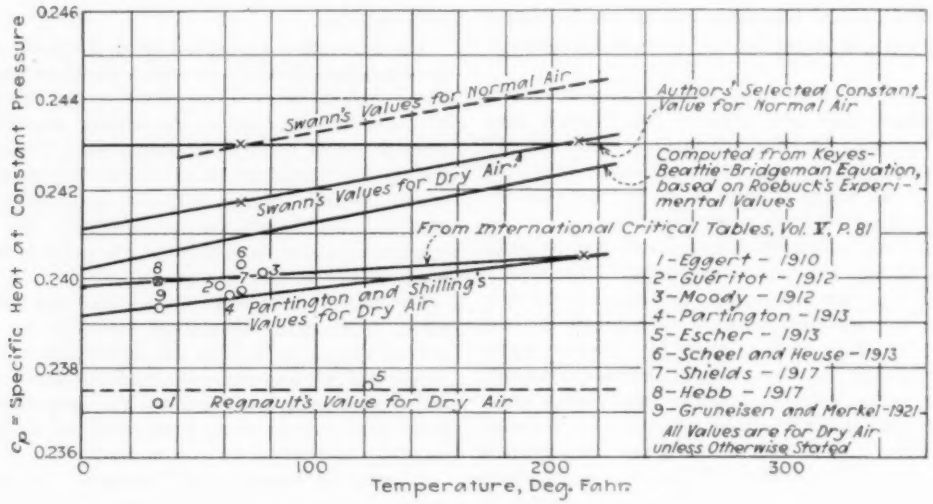


FIG. 1

NOTATION

p_1 = absolute value, lb. per sq. in., of the greater pressure
p_2 = absolute value, lb. per sq. in., of the lesser pressure
T_1, T_2 = corresponding values of absolute temperature, deg. fahr.
v_1, v_2 = corresponding values of volume, cu. ft. of a given weight, which may be any weight, but is usually 1 lb.
p_0 = 14.7 lb. per sq. in. = standard atmospheric pressure. This is almost exactly 760 mm. Hg, the usual physicists' standard. It is equal to 29.92 in. Hg at 32 deg. fahr.
T_0 = 527.6 deg. fahr. abs. = standard atmospheric absolute fahrenheit temperature. This is 68 deg. fahr. or 20 deg. cent., and is a convenient and commonly used value of standard temperature
M = molecular weight
ρ_{a0} = 0.075 lb. per cu. ft. = density of Normal Air at standard conditions, 14.7 lb. and 68 deg. fahr.
ρ_0 = density of any gas at p_0 and T_0

$$\rho_0 = G\rho_{a0} = M\rho_{a0}/28.896$$

G = specific gravity of any gas referred to normal air as 1.00

$$G = M/28.896 = \rho_0/\rho_{a0}$$

c = coefficient of discharge for a nozzle. For a nozzle with well-rounded approach, this may be taken as 0.97 for nozzles with diameters less than 1 in., 0.98 for diameters 1 in. to $2^1/_2$ in., 0.985 up to 4 in., 0.99 up to 12 in., and 0.995 for greater diameters
A = nozzle area, sq. ft.
V = velocity, ft. per sec.
p_m = mean effective pressure, lb. per sq. in.
P = horsepower
w = energy, ft-lb.
w = weight flow, lb. per sec.
q_s = volume, cu. ft. per min. at conditions p_s, T_s
J = 778 = mechanical equivalent of heat, ft-lb. per B.t.u.

c_p = specific heat at constant pressure, B.t.u. per lb. per deg. fahr.
= 0.243 for Normal Air with usual moisture
c_v = specific heat at constant volume
k = c_p/c_v. This is 1.3947 for normal air and perfect diatomic gases such as air, oxygen, nitrogen, etc., k is said to be about 1.67 for monatomic gases such as mercury vapor or helium, and 1.33 for triatomic gases such as CO_2 and for some gases with a greater number of atoms. The table for Y and most of the numerical values apply only for $k = 1.3947$.
Y = $(p_1/p_2)^{(k-1)/k} - 1$
R = perfect-gas constant in the relation $pv = RT$
= $p_0/\rho_0 T_0$

II—PARTICULAR VALUES FOR ALL PERFECT DIATOMIC GASES

Gas		Molecular weight	c_p	c_v	v_0 (Volume, cu. ft. per lb. at 14.7 lb. per sq. in., 68 deg. fahr.)	ρ_0 (Density, lb. per cu. ft. at 14.7 lb. per sq. in., 68 deg.)	G (Specific gravity referred to normal air)	R (R in $pv = RT$ with p in lb. per sq. in.)
Any perfect gas	...	M	$\frac{144\ p_0}{JT_0\rho_0(k-1)/k}$ or $\frac{1.987}{M(k-1)/k}$ or $\frac{0.06876}{G(k-1)/k}$	c_p/k	$\frac{385.29}{M}$	$0.002595M$	$\frac{M}{28.896}$	$\frac{10.735}{M}$
Any perfect diatomic gas	...	M	$\frac{0.01822}{\rho_0}$ or $\frac{7.021}{M}$ or $\frac{0.243}{G}$	$\frac{5.034}{M}$	$\frac{385.29}{M}$	$0.002595M$	$\frac{M}{28.896}$	$\frac{10.735}{M}$
Normal air (with usual moisture)		28.896	0.243	0.174	13.33	0.0750	1.000	0.3715
Dry air		28.989	0.242	0.174	13.29	0.0752	1.003	0.3703
Oxygen	O_2	32.000	0.219	0.157	12.04	0.0830	1.107	0.3355
Nitrogen	N_2	28.016	0.251	0.180	13.75	0.0727	0.970	0.3836
Carbon monoxide	CO	28.000	0.251	0.180	13.76	0.0727	0.969	0.3834

I—GENERAL VALUES FOR PERFECT DIATOMIC GASES

$$\frac{k-1}{k} = 1 - \frac{1}{k} = 0.283$$

Volume in liters of 1 gram of any perfect gas at 0 deg. cent. and 760 mm. of mercury

$$= 22.4115/M$$

Density of normal air (carrying usual moisture) at 14.7 lb. per sq. in. and 68 deg. fahr.

= 0.075 lb. per cu. ft.

These exact values and the perfect-gas laws are the entire basis of the present paper. The resultant numerical values are given throughout to a greater number of decimal places than usually needed in engineering work, for use in checks with different forms of formula, or in especially precise computations.

$$k = c_p/c_v = 1.3947;\quad \frac{2}{k+1} = 0.83518;\quad \frac{k}{k-1} = 3.53357$$

Critical-Pressure Ratio for Flow

$$= \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 0.52918$$

Volumetric Specific Heat, B.t.u. per deg. for 1 cu. ft. of any perfect diatomic gas

$$= 0.075 \times 0.243 = 0.0182$$

The properties of hydrogen do not quite fall in line with the constants of Table II. Diatomic gases other than those given depart appreciably from the perfect-gas laws. It is to be noted that the distinguishing characteristic of a particular gas may be given either by the value of R the gas constant, c_p specific heat, ρ_0 standard density, v_0 specific volume, M molecular weight, or G specific gravity, and that any one of these quantities may be given in terms of any of these quantities, but may be changed to terms of any others by the relations of the first two lines of Table II.

III—GENERAL FORMULAS

The term "adiabatic change" is used for the process of perfect compression or expansion, without loss of energy or addition of heat. It is sometimes called "Isentropic Change."

Pressure, Volume, and Temperature Relations for Adiabatic Change for Any Perfect Gas

$$\frac{\rho_1}{\rho_2} = \frac{v_2}{v_1} = \frac{p_1/p_2}{Y+1} \quad \text{..... [1]}$$

$$\frac{T_1}{T_2} = (Y + 1) \quad [2]$$

$$T_1 - T_2 = \left(\frac{Y}{Y+1}\right) T_1 = YT_2 \quad [3]$$

$$(p_1/p_2)^{(k-1)/k} = Y + 1 \quad [4]$$

$$\left(\frac{p_2}{p_1}\right)^{2/k} = \left(\frac{Y+1}{p_1/p_2}\right)^2; \quad \left(\frac{p_2}{p_1}\right)^{1+1/k} = \frac{Y+1}{(p_1/p_2)^2};$$

$$\left(\frac{p_2}{p_1}\right)^{2/k} - \left(\frac{p_2}{p_1}\right)^{1+1/k} = \frac{Y(Y+1)}{(p_1/p_2)^2} \quad [5]$$

IV—MEAN EFFECTIVE PRESSURE, ENERGY OF COMPRESSION AND EXPANSION, ETC., FOR ADIABATIC CHANGE OF NORMAL AIR AND PERFECT DIATOMIC GASES

Mean Effective Pressure, Lb. per Sq. In.:

(*a*) Under the adiabatic curve pv^k = constant, during compression or expansion only,

$$p_m = \frac{2.534\ p_1 Y}{p_1/p_2 - 1 - Y} = \frac{2.534\ p_2 Y}{1 - (p_2/p_1)(Y+1)};$$

$$2.534 = 1/(k-1) \quad [6]$$

(*b*) During a complete cycle or indicator card, and including work of delivery,

$$p_m = 3.534\ p_2 Y; \qquad 3.534 = k/(k-1) \quad [7]$$

Energy of Compression or Expansion, Ft-Lb. per Lb. of Gas:

(*c*) Under the adiabatic curve pv^k = constant, during compression or expansion only,

$$w = 364.8\ p_1 v_1 \frac{Y}{Y+1} = 364.8\ p_2 v_2 Y = 364.8R\ T_2 Y \quad [8]$$

$$364.8 = 144/(k-1);$$

(*d*) During a complete cycle or indicator card, and including work of delivery,

$$w = 508.8\ p_1 v_1 \frac{Y}{Y+1} = 508.8\ p_2 v_2 Y = 508.8R\ T_2 Y \quad [9]$$

$$508.8 = 144k/(k-1)$$

Energy of Compression or Expansion, Ft-Lb., for Gas Occupying a Volume of 1 Cu. Ft. at Low-Pressure Conditions p_2:

(*e*) Under the adiabatic curve pv^k = constant, during compression or expansion only,

$$w = 364.8\ p_2 Y; \qquad 364.8 = 144/(k-1) \quad [10]$$

(*f*) During a complete cycle or indicator card and including work of delivery,

$$w = 508.8\ p_2 Y; \qquad 508.8 = 144k/(k-1) \quad [11]$$

Power of Compression or Expansion, Horsepower During a Complete Cycle or Indicator Card and Including Work of Delivery:

$$P = 0.9252\ p_2 v_2 Y; \qquad 0.9252 = \frac{144}{550}\frac{k}{k-1} \quad [12]$$

For small pressure ratios,

$$P = 0.2618\ v_2(p_1 - p_2); \qquad 0.2618 = 144/550 \quad [13]$$

These are the horsepowers for any flow v_2 in cu. ft. per sec., at the low-pressure conditions, or the horsepowers for a flow of 1 lb. per sec. if v_2 is the specific volume, cu. ft. per lb.

The horsepower for a flow of 1 Lb. per Sec. of Normal Air is

$$P = 0.3437\ T_2 Y = 0.3437\ T_1 Y/(Y+1) \quad [14]$$

$$0.3437 = \frac{144Rk}{550\ (k-1)}$$

For small pressure ratios, for 1 Lb. per Sec. of Normal Air,

$$P = 0.0973\ T_2(p_1 - p_2)/p_2; \qquad 0.0973 = 144R/550 \quad [15]$$

Formulas for Compression in s Stages With Complete Intercooling to Initial Temperature Between, and With Equal Pressure Ratio in Each Stage (this theoretically gives minimum power):

There is first found the value Y_s for each stage. This is the value of Y corresponding to the sth root of the overall pressure ratio, or it is given by

$$Y_s = \sqrt[s]{Y+1} - 1$$

where Y corresponds to the overall pressure ratio. Then P_s the power for one stage is found from [12] or [14]. The overall power is

$$P = s\ P_s \quad [16]$$

The Equivalent Mean Effective Pressure for a multi-stage compressor based on the initial volume of the first stage is

$$p_m = 3.534\ s p_2 Y_s; \qquad 3.534 = k/(k-1) \quad [17]$$

V—FLOW OF GAS THROUGH A NOZZLE

If the greater pressure p_1 is taken with an impact tube, the formulas are to be used as given. If the greater pressure p_1 is taken as static pressure, all of the expressions for velocity and flow are to have included as a divisor the value

$$\sqrt{1 - \left(\frac{D}{D_p}\right)^4 \left(\frac{Y+1}{p_1/p_2}\right)^2} \quad [18]$$

Here D/D_p is the ratio of the diameter of the minimum section of the nozzle to the diameter of the pipe where the static pressure p_1 is taken. If D/D_p is small, the divisor becomes practically 1.00 and need not be used.

Velocity of Discharge Through a Nozzle With Well-Rounded Approach, Ft. per Sec, (see next paragraph):

(*a*) For any perfect gas,

$$V = 223.75\ c\ \sqrt{c_p}\ \sqrt{\left(\frac{Y}{Y+1}\right) T_1} = 223.75\ c\ \sqrt{c_p}\ \sqrt{T_2 Y} \quad [19]$$

$$223.75 = \sqrt{2gJ}$$

(*b*) For normal air

$$V = 110.30\ c\ \sqrt{\left(\frac{Y}{Y+1}\right) T_1} = 110.30c\ \sqrt{T_2 Y} \quad [20]$$

$$110.30 = \sqrt{2gJc_p}$$

Critical Pressure. For flow of a compressible fluid through a nozzle, there is a "Critical Pressure" which is about half of

VALUES OF Y FOR NORMAL AIR AND PERFECT DIATOMIC GASES [$Y = (p_1/p_2)^{0.283} - 1$]

p_1/p_2		0	1	2	3	4	5	6	7	8	9
1.00	0.00	000	028	057	085	113	141	169	198	226	254
1.01		282	310	338	366	394	422	450	478	506	534
1.02		562	590	618	646	673	701	729	757	785	812
1.03		840	868	895	923	951	978	006	034	061	089
1.04	0.01	116	144	171	199	226	253	281	308	336	363
1.05		390	418	445	472	500	527	554	581	608	636
1.06		663	690	717	744	771	798	825	852	879	906
1.07		933	960	987	014	041	068	095	122	148	175
1.08	0.02	202	229	255	282	309	336	362	389	416	442
1.09		469	495	522	549	575	602	628	655	681	708
1.10		734	760	787	813	840	866	892	919	945	971
1.11		997	024	050	076	102	129	155	181	207	233
1.12	0.03	259	285	311	337	363	389	415	441	467	493
1.13		519	545	571	597	623	649	675	700	726	752
1.14		778	804	829	855	881	906	932	958	983	009
1.15	0.04	035	060	086	111	137	162	188	213	239	264
1.16		290	315	341	366	391	417	442	467	493	518
1.17		543	569	594	619	644	670	695	720	745	770
1.18		796	821	846	871	896	921	946	971	996	021
1.19	0.05	046	071	096	121	146	171	196	221	245	270
1.20		295	320	345	370	394	419	444	469	493	518
1.21		543	567	592	617	641	666	691	715	740	764
1.22		789	813	838	862	887	911	936	960	985	009
1.23	0.06	034	058	082	107	131	155	180	204	228	253
1.24		277	301	325	350	374	398	422	446	470	495
1.25		519	543	567	591	615	639	663	687	711	735
1.26		759	783	807	831	855	879	903	927	951	974
1.27		998	022	046	070	094	117	141	165	189	212
1.28	0.07	236	260	283	307	331	354	378	402	425	449
1.29		472	496	520	543	567	590	614	637	661	684
1.30		708	731	754	778	801	825	848	871	895	918
1.31		941	965	988	011	035	058	081	104	128	151
1.32	0.08	174	197	220	243	267	290	313	336	359	382
1.33		405	428	451	474	497	520	543	566	589	612
1.34		635	658	681	704	727	750	773	795	818	841
1.35		864	887	910	932	955	978	001	023	046	069
1.36	0.09	092	114	137	160	182	205	228	250	273	295
1.37		318	341	363	386	408	431	453	476	498	521
1.38		543	566	588	611	633	655	678	700	723	745
1.39		767	790	812	834	857	879	901	923	946	968
1.40		990	012	035	057	079	101	123	145	168	190
1.41	0.10	212	234	256	278	300	322	344	366	389	411
1.42		433	455	477	499	521	542	564	586	608	630
1.43		652	674	696	718	740	761	783	805	827	849
1.44		871	892	914	936	958	979	001	023	045	066
1.45	0.11	088	110	131	153	175	196	218	239	261	283
1.46		304	326	347	369	390	412	433	455	476	498
1.47		520	541	562	584	605	627	648	669	691	712
1.48		734	755	776	798	819	840	862	883	904	925
1.49		947	968	989	010	032	053	074	095	116	138
1.50	0.12	159	180	201	222	243	264	286	307	328	349
1.51		370	391	412	433	454	475	496	517	538	559
1.52		580	601	622	643	664	685	706	726	747	768
1.53		789	810	831	852	872	893	914	935	956	977
1.54		997	018	039	060	080	101	122	142	163	184
1.55	0.13	205	225	246	266	287	308	328	349	370	390
1.56		411	431	452	472	493	513	534	554	575	595
1.57		616	636	657	677	698	718	739	759	780	800
1.58		820	841	861	881	902	922	942	963	983	003
1.59	0.14	024	044	064	085	105	125	145	165	186	206
1.60		226	246	267	287	307	327	347	367	387	408
1.61		428	448	468	488	508	528	548	568	588	608
1.62		628	648	668	688	708	728	748	768	788	808
1.63		828	848	868	888	908	928	948	968	988	007
1.64	0.15	027	047	067	087	107	126	146	166	186	206
1.65		225	245	265	284	304	324	344	363	383	403
1.66		423	442	462	481	501	521	540	560	580	599
1.67		619	638	658	678	697	717	736	756	775	795
1.68		814	834	853	873	892	912	931	951	970	990
1.69	0.16	009	028	048	067	087	106	125	145	164	184
1.70		203	222	242	261	280	299	319	338	357	377
1.71		396	415	434	454	473	492	511	531	550	569
1.72		588	607	626	646	665	684	703	722	741	760
1.73		780	799	818	837	856	875	894	913	932	951
1.74		970	989	008	027	046	065	084	103	122	141
1.75	0.17	160	179	198	217	236	255	274	292	311	330
1.76		349	368	387	406	425	443	462	481	500	519
1.77		538	556	575	594	613	631	650	669	688	706
1.78		725	744	762	781	800	818	837	856	874	893
1.79		912	930	949	968	986	005	023	042	061	079
1.80	0.18	098	116	135	153	172	191	209	228	246	265
1.81		283	302	320	339	357	376	394	412	431	449
1.82		468	486	505	523	541	560	578	596	615	633
1.83		652	670	688	707	725	743	762	780	798	816
1.84		835	853	871	890	908	926	944	962	981	999
1.85	0.19	017	035	054	072	090	108	126	144	163	181
1.86		199	217	235	253	271	289	308	326	344	362
1.87		380	398	416	434	452	470	488	506	524	542
1.88		560	578	596	614	632	650	668	686	704	722
1.89		740	758	776	794	811	829	847	865	883	901
1.90		919	937	954	972	990	008	026	044	061	079
1.91	0.20	097	115	133	150	168	186	204	221	239	257
1.92		275	292	310	328	345	363	381	399	416	434
1.93		452	469	487	504	522	540	557	575	593	610
1.94		628	645	663	681	698	716	733	751	768	786
1.95		804	821	839	856	874	891	909	926	944	961
1.96		979	996	013	031	048	066	083	101	118	135
1.97	0.21	153	170	188	205	222	240	257	275	292	309
1.98		327	344	361	379	396	413	431	448	465	482

Proportional Parts

29		28	
1	2.9	1	2.8
2	5.8	2	5.6
3	8.7	3	8.4
4	11.6	4	11.2
5	14.5	5	14.0
6	17.4	6	16.8
7	20.3	7	19.6
8	23.2	8	22.4
9	26.1	9	25.2

27		26	
1	2.7	1	2.6
2	5.4	2	5.2
3	8.1	3	7.8
4	10.8	4	10.4
5	13.5	5	13.0
6	16.2	6	15.6
7	18.9	7	18.2
8	21.6	8	20.8
9	24.3	9	23.4

25		24	
1	2.5	1	2.4
2	5.0	2	4.8
3	7.5	3	7.2
4	10.0	4	9.6
5	12.5	5	12.0
6	15.0	6	14.4
7	17.5	7	16.8
8	20.0	8	19.2
9	22.5	9	21.6

23		22	
1	2.3	1	2.2
2	4.6	2	4.4
3	6.9	3	6.6
4	9.2	4	8.8
5	11.5	5	11.0
6	13.8	6	13.2
7	16.1	7	15.4
8	18.4	8	17.6
9	20.7	9	19.8

21		20	
1	2.1	1	2.0
2	4.2	2	4.0
3	6.3	3	6.0
4	8.4	4	8.0
5	10.5	5	10.0
6	12.6	6	12.0
7	14.7	7	14.0
8	16.8	8	16.0
9	18.9	9	18.0

19		18	
1	1.9	1	1.8
2	3.8	2	3.6
3	5.7	3	5.4
4	7.6	4	7.2
5	9.5	5	9.0
6	11.4	6	10.8
7	13.3	7	12.6
8	15.2	8	14.4
9	17.1	9	16.2

17		16	
1	1.7	1	1.6
2	3.4	2	3.2
3	5.1	3	4.8
4	6.8	4	6.4
5	8.5	5	8.0
6	10.2	6	9.6
7	11.9	7	11.2
8	13.6	8	12.8
9	15.3	9	14.4

15		14	
1	1.5	1	1.4
2	3.0	2	2.8
3	4.5	3	4.2
4	6.0	4	5.6
5	7.5	5	7.0
6	9.0	6	8.4
7	10.5	7	9.8
8	12.0	8	11.2
9	13.5	9	12.6

13		12	
1	1.3	1	1.2
2	2.6	2	2.4
3	3.9	3	3.6
4	5.2	4	4.8
5	6.5	5	6.0
6	7.8	6	7.2
7	9.1	7	8.4
8	10.4	8	9.6
9	11.7	9	10.8

p_1/p_2		0	1	2	3	4	5	6	7	8	9
1.99	0.21	500	517	534	552	569	586	603	620	638	655
2.00		672	689	707	724	741	758	775	792	810	827
2.01		844	861	878	895	913	930	947	964	981	998
2.02	0.22	015	032	049	066	084	101	118	135	152	169
2.03		186	203	220	237	254	271	288	305	322	339
2.04		356	373	390	407	424	441	458	474	491	508
2.05		525	542	559	576	593	610	627	644	660	677
2.06		694	711	728	745	762	778	795	812	829	846
2.07		863	879	896	913	930	946	963	980	997	013
2.08	0.23	030	047	064	080	097	114	130	147	164	181
2.09		197	214	231	247	264	281	297	314	331	347
2.10		364	380	397	414	430	447	463	480	497	513
2.11		530	546	563	579	596	613	629	646	662	679
2.12		695	712	728	745	761	778	794	811	827	844
2.13		860	877	893	909	926	942	959	975	992	008
2.14	0.24	024	041	057	074	090	106	123	139	155	172
2.15		188	204	221	237	253	270	286	302	319	335
2.16		351	368	384	400	416	433	449	465	481	498
2.17		514	530	546	563	579	595	511	627	644	660
2.18		676	692	708	724	741	757	773	789	805	821
2.19		838	854	870	886	902	918	934	950	966	983
2.20		999	015	031	047	063	079	095	111	127	143
2.21	0.25	159	175	191	207	223	239	255	271	287	303
2.22		319	335	351	367	383	399	415	431	447	463
2.23		479	495	511	526	542	558	574	590	606	622
2.24		638	654	669	685	701	717	733	749	765	780
2.25		796	812	828	844	859	875	891	907	923	938
2.26		954	970	986	001	017	033	049	064	080	096
2.27	0.26	112	127	143	159	175	190	206	222	237	253
2.28		269	284	300	316	331	347	363	378	394	409
2.29		425	441	456	472	488	503	519	534	550	566
2.30		581	597	612	628	643	659	675	690	706	721
2.31		737	752	768	783	799	814	830	845	861	876
2.32		892	907	923	938	954	969	984	000	015	031
2.33	0.27	046	062	077	092	108	123	139	154	169	185
2.34		200	216	231	246	262	277	292	308	323	338
2.35		354	369	384	400	415	430	446	461	476	492
2.36		507	522	538	553	568	583	599	614	629	644
2.37		660	675	690	705	721	736	751	766	781	797
2.38		812	827	842	857	873	888	903	918	933	948
2.39		964	979	994	009	024	039	054	070	085	100
2.40	0.28	115	130	145	160	175	190	205	220	236	251
2.41		266	281	296	311	326	341	356	371	386	401
2.42		416	431	446	461	476	491	506	521	536	551
2.43		566	581	596	611	626	641	656	671	686	701
2.44		716	730	745	760	775	790	805	820	835	850
2.45		865	879	894	909	924	939	954	969	984	998
2.46	0.29	013	028	043	058	073	087	102	117	132	147
2.47		162	176	191	206	221	235	250	265	280	295
2.48		309	324	339	353	368	383	398	412	427	442
2.49		457	471	486	501	515	530	545	559	574	589
2.50		604	618	633	647	662	677	691	706	721	736
2.51		750	765	779	794	808	823	838	852	867	881
2.52		896	911	925	940	954	969	984	998	013	027
2.53	0.30	042	056	071	085	100	114	129	144	158	173
2.54		187	202	216	231	245	260	274	289	303	318
2.55		332	346	361	375	390	404	419	433	448	462
2.56		476	491	505	520	534	548	563	577	592	606
2.57		620	635	649	663	678	692	707	721	735	750
2.58		764	778	793	807	821	836	850	864	879	893
2.59		907	921	936	950	964	979	993	007	021	036
2.60	0.31	050	064	079	093	107	121	136	150	164	178
2.61		193	207	221	235	249	264	278	292	306	320
2.62		335	349	363	377	391	405	420	434	448	462
2.63		476	490	505	519	533	547	561	575	589	603
2.64		618	632	646	660	674	688	702	716	730	744
2.65		759	773	787	801	815	829	843	857	871	885
2.66		899	913	927	941	955	969	983	997	011	025
2.67	0.32	039	053	067	081	095	109	123	137	151	165
2.68		179	193	207	221	235	249	262	276	290	304
2.69		318	332	346	360	374	388	402	416	429	443
2.70		457	471	485	499	513	527	540	554	568	582
2.71		596	610	624	637	651	665	679	693	707	720
2.72		734	748	762	776	789	803	817	831	845	858
2.73		872	886	900	913	927	941	955	968	982	996
2.74	0.33	010	023	037	051	065	078	092	106	119	133
2.75		147	161	174	188	202	215	229	243	256	270
2.76		284	297	311	325	338	352	366	379	393	407
2.77		420	434	448	461	475	488	502	516	529	543
2.78		556	570	584	597	611	624	638	651	665	679
2.79		692	706	719	733	746	760	773	787	801	814
2.80		828	841	855	868	882	895	909	922	936	949
2.81		963	976	990	003	017	030	044	057	070	084
2.82	0.34	097	111	124	138	151	165	178	191	205	218
2.83		232	245	259	272	285	299	312	326	339	352
2.84		366	379	393	406	419	433	446	459	473	486
2.85		500	513	526	540	553	566	580	593	606	620
2.86		633	646	660	673	686	700	713	726	739	753
2.87		766	779	793	806	819	832	846	859	872	886
2.88		899	912	925	939	952	965	978	991	005	018
2.89	0.35	031	044	058	071	084	097	110	124	137	150
2.90		163	176	190	203	216	229	242	255	269	282
2.91		295	308	321	334	347	361	374	387	400	413
2.92		426	439	452	466	479	492	505	518	531	544
2.93		557	570	584	597	610	623	636	649	662	675
2.94		688	701	714	727	740	753	767	780	793	806
2.95		819	832	845	858	871	884	897	910	923	936
2.96		949	962	975	988	001	014	027	040	053	066
2.97	0.36	079	092	105	118	131	144	157	169	182	195
2.98		208	221	234	247	260	273	286	299	312	324
2.99		337	350	363	376	389	402	415	428	440	453

p_1/p_2	0	1	2	3	4	5	6	7	8	9
3.0	0.3647	0.3659	0.3672	0.3685	0.3698	0.3711	0.3723	0.3736	0.3749	0.3761
3.1	0.3774	0.3786	0.3799	0.3811	0.3824	0.3836	0.3849	0.3861	0.3874	0.3886
3.2	0.3898	0.3911	0.3923	0.3935	0.3947	0.3959	0.3971	0.3984	0.3996	0.4008
3.3	0.4020	0.4032	0.4044	0.4056	0.4068	0.4080	0.4091	0.4103	0.4115	0.4127
3.4	0.4139	0.4150	0.4162	0.4174	0.4186	0.4197	0.4209	0.4220	0.4232	0.4244
3.5	0.4255	0.4267	0.4278	0.4290	0.4301	0.4313	0.4324	0.4335	0.4347	0.4358
3.6	0.4369	0.4380	0.4392	0.4403	0.4414	0.4425	0.4437	0.4448	0.4459	0.4470
3.7	0.4481	0.4492	0.4503	0.4514	0.4525	0.4536	0.4547	0.4558	0.4569	0.4580
3.8	0.4591	0.4602	0.4612	0.4623	0.4634	0.4645	0.4656	0.4666	0.4677	0.4688
3.9	0.4698	0.4709	0.4720	0.4730	0.4741	0.4752	0.4762	0.4773	0.4783	0.4794
4.0	0.4804	0.4815	0.4825	0.4835	0.4846	0.4856	0.4867	0.4877	0.4887	0.4898
4.1	0.4908	0.4918	0.4928	0.4939	0.4949	0.4959	0.4970	0.4980	0.4990	0.5000
4.2	0.5010	0.5020	0.5030	0.5040	0.5050	0.5060	0.5070	0.5080	0.5090	0.5100
4.3	0.5110	0.5120	0.5130	0.5140	0.5150	0.5160	0.5170	0.5179	0.5189	0.5199
4.4	0.5209	0.5219	0.5228	0.5238	0.5248	0.5258	0.5267	0.5277	0.5287	0.5296
4.5	0.5306	0.5316	0.5325	0.5335	0.5344	0.5354	0.5363	0.5373	0.5382	0.5392
4.6	0.5401	0.5411	0.5420	0.5430	0.5439	0.5449	0.5458	0.5467	0.5477	0.5486
4.7	0.5495	0.5505	0.5514	0.5523	0.5533	0.5542	0.5551	0.5560	0.5570	0.5579
4.8	0.5588	0.5597	0.5606	0.5616	0.5625	0.5634	0.5643	0.5652	0.5661	0.5670
4.9	0.5679	0.5688	0.5697	0.5706	0.5715	0.5724	0.5733	0.5742	0.5751	0.5760
5.0	0.5769	0.5778	0.5787	0.5796	0.5805	0.5814	0.5822	0.5831	0.5840	0.5849
5.1	0.5858	0.5867	0.5875	0.5884	0.5893	0.5902	0.5910	0.5919	0.5928	0.5936
5.2	0.5945	0.5954	0.5962	0.5971	0.5980	0.5988	0.5997	0.6006	0.6014	0.6023
5.3	0.6031	0.6040	0.6048	0.6057	0.6065	0.6074	0.6082	0.6091	0.6099	0.6108
5.4	0.6116	0.6125	0.6133	0.6142	0.6150	0.6159	0.6167	0.6175	0.6184	0.6192
5.5	0.6200	0.6209	0.6217	0.6225	0.6234	0.6242	0.6250	0.6258	0.6267	0.6275
5.6	0.6283	0.6291	0.6300	0.6308	0.6316	0.6324	0.6332	0.6340	0.6349	0.6357
5.7	0.6365	0.6373	0.6381	0.6389	0.6397	0.6405	0.6413	0.6421	0.6430	0.6438
5.8	0.6446	0.6454	0.6462	0.6470	0.6478	0.6486	0.6494	0.6502	0.6509	0.6517
5.9	0.6525	0.6533	0.6541	0.6549	0.6557	0.6565	0.6573	0.6581	0.6588	0.6596
6.0	0.6604	0.6612	0.6620	0.6628	0.6635	0.6643	0.6651	0.6659	0.6666	0.6674
6.1	0.6682	0.6690	0.6697	0.6705	0.6713	0.6721	0.6729	0.6736	0.6744	0.6752
6.2	0.6759	0.6767	0.6774	0.6782	0.6789	0.6797	0.6805	0.6812	0.6820	0.6827
6.3	0.6835	0.6843	0.6850	0.6858	0.6865	0.6873	0.6880	0.6888	0.6895	0.6903
6.4	0.6910	0.6918	0.6925	0.6933	0.6940	0.6948	0.6955	0.6963	0.6970	0.6978
6.5	0.6985	0.6992	0.7000	0.7007	0.7014	0.7021	0.7028	0.7036	0.7043	0.7050
6.6	0.7058	0.7065	0.7073	0.7080	0.7087	0.7095	0.7102	0.7110	0.7117	0.7124
6.7	0.7131	0.7138	0.7145	0.7153	0.7160	0.7167	0.7174	0.7181	0.7189	0.7196
6.8	0.7203	0.7210	0.7217	0.7224	0.7232	0.7239	0.7246	0.7253	0.7260	0.7267
6.9	0.7274	0.7281	0.7288	0.7295	0.7302	0.7309	0.7316	0.7323	0.7330	0.7338
7.0	0.7345	0.7352	0.7359	0.7366	0.7373	0.7380	0.7386	0.7393	0.7400	0.7407
7.1	0.7414	0.7421	0.7428	0.7435	0.7442	0.7449	0.7456	0.7463	0.7470	0.7477
7.2	0.7483	0.7490	0.7497	0.7504	0.7511	0.7518	0.7524	0.7531	0.7538	0.7545
7.3	0.7552	0.7559	0.7565	0.7572	0.7579	0.7586	0.7592	0.7599	0.7606	0.7613
7.4	0.7620	0.7626	0.7633	0.7640	0.7646	0.7653	0.7660	0.7666	0.7673	0.7680
7.5	0.7687	0.7693	0.7700	0.7706	0.7713	0.7720	0.7726	0.7733	0.7740	0.7746
7.6	0.7753	0.7760	0.7766	0.7773	0.7779	0.7786	0.7792	0.7799	0.7806	0.7812
7.7	0.7819	0.7825	0.7832	0.7838	0.7845	0.7851	0.7858	0.7864	0.7871	0.7877
7.8	0.7884	0.7890	0.7897	0.7903	0.7910	0.7916	0.7923	0.7929	0.7936	0.7942
7.9	0.7949	0.7955	0.7961	0.7968	0.7974	0.7981	0.7987	0.7993	0.8000	0.8006
8.0	0.8013	0.8019	0.8025	0.8032	0.8038	0.8044	0.8051	0.8057	0.8063	0.8070
8.1	0.8076	0.8082	0.8089	0.8095	0.8101	0.8108	0.8114	0.8120	0.8126	0.8133
8.2	0.8139	0.8145	0.8151	0.8158	0.8164	0.8170	0.8176	0.8183	0.8189	0.8195
8.3	0.8201	0.8207	0.8214	0.8220	0.8226	0.8232	0.8238	0.8245	0.8251	0.8257
8.4	0.8263	0.8269	0.8275	0.8281	0.8288	0.8294	0.8300	0.8306	0.8312	0.8318
8.5	0.8324	0.8330	0.8336	0.8343	0.8349	0.8355	0.8361	0.8367	0.8373	0.8379
8.6	0.8385	0.8391	0.8397	0.8403	0.8409	0.8415	0.8421	0.8427	0.8433	0.8439
8.7	0.8445	0.8451	0.8457	0.8463	0.8469	0.8475	0.8481	0.8487	0.8493	0.8499
8.8	0.8505	0.8511	0.8517	0.8523	0.8529	0.8535	0.8541	0.8547	0.8552	0.8558
8.9	0.8564	0.8570	0.8576	0.8582	0.8588	0.8594	0.8600	0.8605	0.8611	0.8617
9.0	0.8623	0.8629	0.8635	0.8641	0.8646	0.8652	0.8658	0.8664	0.8670	0.8676
9.1	0.8681	0.8687	0.8693	0.8699	0.8705	0.8710	0.8716	0.8722	0.8728	0.8734
9.2	0.8739	0.8745	0.8751	0.8757	0.8762	0.8768	0.8774	0.8779	0.8785	0.8791
9.3	0.8797	0.8802	0.8808	0.8814	0.8819	0.8825	0.8831	0.8837	0.8842	0.8848
9.4	0.8854	0.8859	0.8865	0.8871	0.8876	0.8882	0.8888	0.8893	0.8899	0.8905
9.5	0.8910	0.8916	0.8921	0.8927	0.8933	0.8938	0.8944	0.8949	0.8955	0.8961
9.6	0.8966	0.8972	0.8977	0.8983	0.8989	0.8994	0.9000	0.9005	0.9011	0.9016
9.7	0.9022	0.9028	0.9033	0.9039	0.9044	0.9050	0.9055	0.9061	0.9066	0.9072
9.8	0.9077	0.9083	0.9088	0.9094	0.9099	0.9105	0.9110	0.9116	0.9121	0.9127
9.9	0.9132	0.9138	0.9143	0.9149	0.9154	0.9159	0.9165	0.9170	0.9176	0.9181
10.0	0.9187	0.9192	0.9198	0.9203	0.9208	0.9214	0.9219	0.9225	0.9230	0.9235
10.1	0.9241	0.9246	0.9252	0.9257	0.9262	0.9268	0.9273	0.9278	0.9284	0.9289
10.2	0.9295	0.9300	0.9305	0.9311	0.9316	0.9321	0.9327	0.9332	0.9337	0.9343
10.3	0.9348	0.9353	0.9358	0.9364	0.9369	0.9374	0.9380	0.9385	0.9390	0.9396
10.4	0.9401	0.9406	0.9411	0.9417	0.9422	0.9427	0.9432	0.9438	0.9443	0.9448
10.5	0.9453	0.9459	0.9464	0.9469	0.9474	0.9480	0.9485	0.9490	0.9495	0.9500
10.6	0.9506	0.9511	0.9516	0.9521	0.9526	0.9532	0.9537	0.9542	0.9547	0.9552
10.7	0.9558	0.9563	0.9568	0.9573	0.9578	0.9583	0.9589	0.9594	0.9599	0.9604
10.8	0.9609	0.9614	0.9619	0.9625	0.9630	0.9635	0.9640	0.9645	0.9650	0.9655
10.9	0.9660	0.9665	0.9671	0.9676	0.9681	0.9686	0.9691	0.9696	0.9701	0.9706
11.0	0.9711	0.9716	0.9721	0.9726	0.9732	0.9737	0.9742	0.9747	0.9752	0.9757
11.1	0.9762	0.9767	0.9772	0.9777	0.9782	0.9787	0.9792	0.9797	0.9802	0.9807
11.2	0.9812	0.9817	0.9822	0.9827	0.9832	0.9837	0.9842	0.9847	0.9852	0.9857
11.3	0.9862	0.9867	0.9872	0.9877	0.9882	0.9887	0.9892	0.9897	0.9902	0.9907
11.4	0.9912	0.9916	0.9921	0.9926	0.9931	0.9936	0.9941	0.9946	0.9951	0.9956
11.5	0.9961	0.9966	0.9971	0.9975	0.9980	0.9985	0.9990	0.9995	1.0000	1.0005
11.6	1.0010	1.0015	1.0019	1.0024	1.0029	1.0034	1.0039	1.0044	1.0049	1.0054
11.7	1.0058	1.0063	1.0068	1.0073	1.0078	1.0083	1.0087	1.0092	1.0097	1.0102
11.8	1.0107	1.0112	1.0116	1.0121	1.0126	1.0131	1.0136	1.0140	1.0145	1.0150
11.9	1.0155	1.0160	1.0164	1.0169	1.0174	1.0179	1.0184	1.0188	1.0193	1.0198
12.0	1.0203	1.0207	1.0212	1.0217	1.0222	1.0226	1.0231	1.0236	1.0241	1.0245

the greater pressure p_1. The exact ratio of critical pressure to p_1 for any perfect gas is

$$\left(\frac{2}{k+1}\right)^{k/(k-1)} \quad \text{.....[21]}$$

The value is 0.52918 for Normal Air and other perfect diatomic gases.

Affected Flow. If the lesser pressure p_2, or the pressure of the region into which the nozzle discharges, is above the Critical Pressure, so that the differential pressure $(p_1 - p_2)$ is less than half of p_1, the nozzle theoretically required has a well-rounded approach, ending in a parallel portion with the minimum section, and the above formulas for velocity and the succeeding formulas for flow apply directly. The pressure at the nozzle end is exactly the pressure p_2 of the region into which the nozzle discharges. This pressure, therefore affects the flow.

Unaffected Flow. If the lesser pressure p_2, or the pressure of the region into which the nozzle discharges, is below the "Critical Pressure," the nozzle theoretically required has a well-rounded approach, *succeeded by a divergent portion.* The theoretically required ratio of the area at the nozzle end to the area at minimum section for any perfect gas is

$$\frac{p_1}{p_2}\frac{\left(\frac{2}{k+1}\right)^{1/(k-1)}\sqrt{\frac{k-1}{k+1}}}{\sqrt{Y(Y+1)}} \quad \text{.....[22]}$$

For normal air or perfect diatomic gases, this becomes

$$\frac{p_1}{p_2}\frac{0.2572}{\sqrt{Y(Y+1)}}; \quad 0.2572 = \left(\frac{2}{k+1}\right)^{1/(k-1)}\sqrt{\frac{k-1}{k+1}} \quad [23]$$

If the divergent portion is properly supplied, the formulas for velocity above given apply directly, but if the divergent portion is not supplied, they do not hold.

If the lesser pressure p_2, or the pressure of the region into which the nozzle discharges, is less than the "Critical Pressure," this pressure does not affect the flow, and the formulas for flow below do not apply, whether or not the nozzle has the divergent portion. The pressure at the minimum section, if there is a divergent portion, or the pressure at the nozzle end, if there is no divergent portion, is the critical pressure.

Flow With Small Differential Pressure. If the differential pressure $(p_1 - p_2)$ is less than 10 per cent of the greater pressure p_1 the formulas below are unduly cumbersome, and while they are correct, they should not be used. For this case, and for the case of flow where the lesser pressure p_2 is less than half of the greater pressure, proper formulas for flow are given in the paper "Measurement of Flow of Air and Gases With Nozzles" (S. A. Moss, Trans. A.S.M.E., vol. 50 (1928), no. 2, paper no. APM-3). The formulas for flow below therefore only apply to the case where the differential pressure $(p_1 - p_2)$ is between 10 per cent and 50 per cent of the greater pressure p_1. Even in this region, the use of the formulas below is *merely optional*, and they may be replaced by a set of formulas not using Y, given in the paper mentioned above.

Any of the expressions for velocity or flow for any gas may be expressed in terms of the value for the gas in question of any one of the quantities R, c_p, p_0, v_0, M, or G. The relations between these are given in the first two lines of Table II.

Weight Flow Through Nozzle, Lb. per Sec.:

(*a*) For perfect diatomic gases,

$$w = 296.88\frac{cAp_2}{\sqrt{T_1}}\sqrt{G}\sqrt{Y(Y+1)} \quad \text{.....[24]}$$

$$296.88 = \sqrt{\frac{144\ 2gT_0\rho_{a0}}{p_0(k-1)/k}}$$

(*b*) For normal air,

$$w = 296.88\frac{cAp_2}{\sqrt{T_1}}\sqrt{Y(Y+1)} \quad \text{.....[25]}$$

$$296.88 = \sqrt{\frac{144\ 2gT_0\rho_{a0}}{p_0(k-1)/k}}$$

Volume Flow Through Nozzle, Cu. Ft. per Min., expressed in terms of volume at given conditions p_3, T_3:

(*a*) For perfect diatomic gases,

$$q_3 = 6618\frac{T_3}{p_3}\frac{cAp_2}{\sqrt{G}\sqrt{T_1}}\sqrt{Y(Y+1)} \quad \text{.....[26]}$$

$$6618 = 60\sqrt{2gJGc_p}$$

(*b*) For normal air,

$$q_3 = 6618\frac{T_3}{p_3}\frac{cAp_2}{\sqrt{T_1}}\sqrt{Y(Y+1)} \quad \text{.....[27]}$$

$$6618 = 60\sqrt{2gJc_p}$$

Outline of Tables for Y

Seven-place Vega logarithm tables were used in computing the tabular values. These are given for every pressure ratio p_1/p_2 from 1.000 to 3.000 in steps of 0.001, with some proportional parts to enable interpolation to 0.0001. For ratios from 3.00 to 12.00, steps of 0.01 are used. It is to be noted that if values of $(p_1 - p_2)/p_2$ are known, the corresponding values of p_1/p_2 for use of the table are given by

$$\frac{p_1}{p_2} = \frac{p_1 - p_2}{p_2} + 1$$

The values of the adiabatic factor Y are given to five decimal places for pressure ratios from 1.000 to 3.000, and to four places for ratios from 3.0 to 12.0. It is to be noted that the Y-table applies only to air and perfect diatomic gases.

Errata. In the previously mentioned paper "Measurement of Flow of Air and Gases With Nozzles" [S. A. Moss, Trans. A.S.M.E., vol. 50 (1928)], there is a typographical error in the formula immediately preceding the section headed "Temperature Corrections for Mercury Columns." In the denominator of the last fraction in the formula, p_2 should be p_1.

Discussion

Barton H. Coffey.[4] The authors' 1927 and 1929 papers are really one, and the writer therefore experienced double pleasure in going over them for two reasons: first, the broadminded and liberal spirit of public service exhibited by one of our great corporations in giving to the profession this extremely valuable experience and original research; second, the clear and thorough manner in which the authors have presented the data.

The writer believes the method of choosing the mathematical constants so as to harmonize the links in the chain of perfect-gas relations leads without doubt to maximum overall accuracy, as the Y-function and its elaborate table of values most certainly lead to important labor savings in computation. The writer believes that dry air plus the normal or annual average quantity of water vapor is the proper and scientific basis for rating capacity of compressed-air apparatus as advocated by the author under the caption "Normal Air." The chosen psychrometric state is 68 deg. dry-bulb and 36 per cent relative humidity, which appears somewhat low for a mean value based upon population affected. On a population basis the sparsely settled country east of the Cascade and Sierra Nevada Ranges and west of the Rockies averaging 50 per cent mean annual relative humidity comes out leaving a rough average for the rest of the country of 75 per cent relative humidity at 56 deg. mean average temperature and about 50 grains of water vapor per pound of dry air.[5] At 68 deg. air temperature, identical water vapor would call for 50 per cent relative humidity, or an increase of 14 per cent over the authors' figure, a trifling difference in affecting the constants.

Thomas E. Butterfield.[6] The table contained in this paper will be valuable in the computation of the work, horsepower, and mean effective pressure for adiabatic compression of air.

For the maximum pressure ratio of 12, the assumption of constant specific heat is justified for engineering computations. The maximum variation of specific heat for the corresponding temperature range is less than 2 per cent.

For expansions with the same maximum pressure ratio, the approximation involves a considerably greater error. This is because the specific heat for the corresponding temperature range is considerably lower than that assumed by the authors, and also because the air composition constantly varies due to separation of moisture. For nozzle work the effect of supersaturation will diminish the error.

Though the usual methods of interpolation were presumably used in computing the table, the work must have taken much time. Nevertheless, the question arises whether the computation of additional tables for compressions other than adiabatic would not yield results valuable enough to compensate for the labor involved.

The writer feels that the great value of the table is ease in finding the temperature ratio, which is $Y + 1$. This gives the final temperature. The work of the compression cycle, discussed under IV (*d*), is then readily found as $w = 778 \times 0.243\ (T_1 - T_2) = 189\ (T_1 - T_2)$ in ft-lb. per lb. air. The work of compression alone is $778 \times 0.174\ (T_1 - T_2) = 135.37\ (T_1 - T_2)$.

In calculating the mean effective pressure of compression, the most convenient procedure is to find the volume ratio v_2/v_1 as the quotient of the pressure ratio p_1/p_2, which is the argument, divided by the temperature ratio, equal to $(Y + 1)$.

Dividing the work of compression by the reduction in volume $v_2 - v_1$, the mean effective pressure is obtained, or

$$\text{m.e.p.} = \frac{135.37\ (T_1 - T_2)}{144\ (v_2 - v_1)} = 0.940\frac{T_1 - T_2}{v_2 - v_1}\ \text{lb. per sq. in.}$$

$$\text{The m.e.p. for the compression cycle is} = \frac{189(T_1 - T_2)}{144\ v_2} = 1.312\frac{(T_1 - T_2)}{v_2}$$

M. G. Robinson.[7] This paper gives a clearly stated set of formulas for use in the thermodynamic computations dealing with compression, expansion, and flow of "normal" air, as defined by the authors. The pains taken in framing the formulas to make them mutually consistent, easy to work with on an ordinary slide rule, and yet capable of yielding numerical accuracy, as well as the care exercised in stating the basis for selecting the fundamental constants involved, should make this paper helpful, within the scope of its application. Although the paper contains frequent references to other "perfect" diatomic gases, the remarks following will be confined to air.

The graphical summary of the available data for c_p of air,

[4] Consulting Engineer, Elizabeth, N. J. Mem. A.S.M.E.

[5] Bulletin Q, 1916, p. 61, U. S. Weather Bureau, Washington, D. C.

[6] Professor of Heat Power Engineering, Lehigh University, Bethlehem, Pa. Mem. A.S.M.E.

[7] Centrifugal Compressor Department, General Electric Company, Lynn, Mass. Assoc-Mem. A.S.M.E.

as given on the second page, is of particular interest. Several years ago, in connection with heat-balance testing of centrifugal compressors (Paper No. 1991, Trans. A.S.M.E., 1925), the speaker went to considerable trouble in an effort to select the most satisfactory value for c_p of air near atmospheric pressure and temperature. The number of textbooks and handbooks still quoting Regnault's value of 0.2375 B.t.u. per lb. per deg. fahr. was amazing. The study pointed to a value of 0.240–0.243 as more proper for average conditions. The well-known work of W. F. G. Swann (1909) was adopted at that time as fully trustworthy, and, if anything, somewhat on the conservative side for the application in question. Referring to the graph on the second page of the paper, it might be well to add the value 0.2415 given by J. R. Roebuck ("The Joule-Thomson Effect in Air," Proc. American Academy of Arts and Sciences, 1925).

It should be definitely understood that the fundamental constants recommended by the paper are for ordinary engineering work with air at or near atmospheric pressure and temperature. For special engineering problems involving the use of very high or very low temperatures, or high pressure, or demanding extreme accuracy, these constants are considerably modified. Excellent data are now available. Probably the equation of state as developed by Dr. Keyes and reported recently in the *Journal of the American Chemical Society*, would furnish satisfactory data for most cases.

The paper applies to air with an average humidity of about 36 per cent. There are cases when the effect of the actual humidity contents of air must be taken into account. Although there are no great theoretical difficulties in estimating this effect, experience shows that the average test engineer, particularly one who is sent to witness an acceptance test, is quite hazy about it, inclined either toward overestimating or underestimating. The title and contents of this paper make the writer feel justified in the following remarks.

At the same pressure and temperature, water vapor is about five-eighths as heavy as air. On the other hand, the c_p of water vapor is approximately double that of air. The adiabatic exponent for water vapor is less than that for air. Hence, in general, it follows that as the relative humidity of air increases, its density (or specific gravity referred to dry air) and adiabatic exponent decrease and its specific heat at constant pressure increases. Applying these observations to flow measurements, it is noted that for a given pressure differential, the weight flow of moist air will be less, it being almost exactly proportional to the square root of specific gravity of the air. On the other hand, the energy required for compressing a given weight of air from a given pressure and temperature to a specified pressure will be greater the greater the relative humidity, it being very nearly inversely proportional to the specific gravity only. In both cases the effect of the exponent variation is practically negligible, and correct results will be obtained by allowing for the specific-gravity change only.

At low relative humidity, and particularly at low temperatures, say, below 75 deg. fahr., the magnitude of the effects mentioned is not great. With high humidity and high temperatures, and especially in case of exhauster or low-pressure air operation, the effects may be quite serious. For instance, air at normal atmospheric pressure, at 100 deg. fahr. and 100 per cent relative humidity, will require almost $2^1/_2$ per cent more energy for compression and will have a c_p value almost $3^1/_2$ per cent greater than dry air under the same conditions of pressure and temperature; while air at one-half atmospheric pressure at 100 deg. fahr. and 100 per cent relative humidity will require nearly 5 per cent more energy for compression to a given pressure and will have a c_p value some 7 per cent greater than dry air at the same pressure and temperature.

A. PETERSON.[8] The authors deserve credit for their efforts to establish uniform constants for use in calculations covering adiabatic compression and expansion of gases, and particularly for the considerable amount of painstaking work involved in calculating the Y-values for the many pressure ratios given in the table. The writer has worked up such tables and therefore knows what it means.

In reading the first part of the paper one obtains the impression that the use of constants established about 60 years ago might lead to results that are not quite reliable. The most important one of these constants is the exponent 0.29, found in many old engineering handbooks. The writer therefore proceeded to calculate what difference in results would be obtained by the use of the exponent 0.29, corresponding to $c_p/c_v = 1.4085$, and the exponent 0.283 given in the paper, and selecting a pressure ratio $p_1/p_2 = 2$ and calculating the theoretical horsepower for this ratio, found a difference of only 0.22 per cent, which is insignificant for all practical purposes.

While the difference in the value of Y is considerably greater (2.65 per cent) for the two exponents, this difference is almost entirely offset by the factor $k/(k-1)$ in the horsepower formula, which for $k = 1.4085$ is 3.448 and for $k = 1.3947$ in the paper is 3.534.

Therefore no serious errors will result by the use of the old constants. However, it would be very desirable if uniform constants as recommended in the paper be adopted in all handbooks if for no other reason than the excellent and most complete table for values of Y, which alone will save considerable time for any one who may have occasion to make calculations on adiabatic compression or expansion of gases, and who may not have available such complete tables.

The writer is a little surprised to note that air of 68 deg. fahr. is referred to as normal air. Practically all blower and compressor manufacturers use a temperature of 60 deg. fahr. for standard air, and this also applies to many users of such equipment. The writer would like to know if there was any particular reason for selecting a temperature of 68 deg. fahr.

H. A. EVERETT.[9] A paper such as this represents a large amount of repetitive computation, and all should be grateful for the authors' willingness to share with us the results of these arduous labors. Such a tabulation may be extremely helpful when used within its limitations, but these limitations must be carefully kept in mind. The authors have clearly stated the basic values on which the table was reared; yet it may not be out of place to point out in a somewhat different manner the assumptions accepted when using the tables.

That hypothetical substance the "perfect gas" is assumed to be a substance which not only conforms completely to the gas laws of Boyle, Charles, and Joule, but also one in which the specific heats remain constant. This assumption of constancy of specific heats is in addition to and independent of that of conformity with the first-mentioned gas laws. In actual gases the variation of specific heat with pressure is but trifling for wide ranges of pressure, but the variation of specific heat with temperature is pronounced, and for temperature ranges met in adiabatic compression or expansion of even moderate amounts its variation is appreciable. For example, in the very common volumetric compression of 5 to 1, with an initial pressure of 14.7 lb., the final temperature of the perfect diatomic gas figured from the customary expression, $pv^{1.40} = \text{constant}$, is equal to 1005 deg. fahr. abs. when the initial temperature is 68 deg. fahr. If,

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however, the final temperature be figured taking cognizance of the variation in specific heat in accordance with Goodenough's equation $Mc_p = 6.93 + 0.1200 \times 10^{-3}T^2$, the final temperature is 999 deg. fahr. abs. There is thus a difference of $^1/_2$ per cent due to the effect of the variable character of the specific heat.

If the specific heat varies, the value of k also varies. It would of course be entirely possible to assume a value of k that would bring the results of the computation by the first method into harmony with that taking cognizance of the variable specific heat. For example, if we were to take the value of k as 1.396, we should obtain the same final temperature by the first method as was computed by using the variable-specific-heat expression. But this would be good for this particular range of temperatures, and only for this particular range of temperatures.

The kinetic theory of gases clearly shows that perfect gases having monatomic molecules should have the value of k equal to $^5/_3$, or 1.6667, since these molecules have but 3 degrees of freedom. For the diatomic molecules where the degrees of freedom are increased to 5, the value of k is $^7/_5$, or 1.4000. For the tri- or poly-atomic molecules, the degrees of freedom become 6, and the corresponding value of k should be $^8/_6 = 1.3333$. If, then, one is to discuss the performance of diatomic perfect gases, it would seem that the value of k must be 1.40. If the authors choose to select a value of k differing from this sufficiently to approximately correct for the effect of the variation in specific heat, that is their privilege, but it should be clearly understood that such an approximation has been made, and furthermore that this value is strictly correct only for that one range where the average value of the true k's as affected by specific-heat variation is equal to that assumed. The authors do not make it quite clear that this point is involved as they have accepted a value of specific heat for what is termed "normal air" and hung the rest of the structure on that value. It may be that this is a reasonable value to assume for air, but if one is to discuss really perfect diatomic gases, then such an assumption of specific heat is unwarranted, and it seems to the writer that the effect of variable specific heat should be taken account of as such, rather than included in the acceptance of an arbitrary value of k.

The writer has checked temperature rises computed both by the tables and by the rather laborious variable-specific-heat method for several of the larger ranges covered by the table and finds their agreement fairly close. He does not wish to detract in any way from the obvious merits of such a paper as the authors have given, but thinks it would be of interest to call attention to the fact that the accepted values really take into account the variation in specific heat, though in an indirect manner. He doubts, however, if the results are dependable to the number of significant figures quoted in the table.

Frank O. Ellenwood.[10] The authors have presented a paper that gives in convenient form certain quantities that are likely to be of service to those making extensive calculations for air and other gases. There are a few statements in the paper, however, that appear to the writer to be slightly misleading at least, and he will therefore call attention to them.

In the first place, considerable stress is laid upon what the authors call the "normal air," and the statement is made on the first page that "this air is much more nearly representative of the air actually passing through a compressor or measuring nozzle than dry air without CO_2 at 0 deg. cent., whose properties as obtained 60 years ago are often used by engineers." In another place on the same page reference is made to the correction for the usual CO_2 content and the "rare gases" to justify the values of density and relative humidity in the so-called "normal air." By looking up the previous paper of Dr. Moss, to which reference is made, the amount of CO_2 in the normal air is found to be 0.03 per cent by volume, and the sum of all the rare gases is much less than 0.002 per cent. Therefore these factors, it seems to the writer, are very much overemphasized in the paper because they are not the ones that are affecting the gas constants appreciably.

On the other hand, consider Fig. 1, and it will be observed that Swann's values of c_p for dry air are nearly 1 per cent higher than Partington and Shilling's and those of other recent experimenters. In this connection it should be mentioned that the Goodenough and Felbeck values of c_p for dry air would give a curve passing almost through points 6 and 8 of Fig. 1. This would be about $^3/_4$ of 1 per cent less than Swann's values for dry air. Thus a difference of nearly 1 per cent is involved in our most reliable values of c_p for dry air, whereas the effect of the CO_2 in the atmosphere on this specific heat is about $^1/_{100}$ part as great. Furthermore, a relative humidity of about 0.70 is very common for the atmosphere in various parts of this country, and when the relative humidity is changed from 0.36 to 0.70, the value of c_p is thereby increased $^1/_2$ of 1 per cent.

Considering the facts that have just been given, the writer believes that one may therefore very properly question the wisdom of defining "normal air" as has been done by the authors of this paper. Since the density of air is far more dependent upon its temperature and humidity than are c_p and k, it seems logical that these two terms should form the starting point, rather than density, from which the other gas constants of "normal air" may be found after agreement is reached on the composition of the latter.

The value of k for normal air has very commonly been used as 1.4, and with a relative humidity of either 36 or 70 per cent it is probably still closer to that figure than it is to the much longer one used in the paper. Since 1.3947 is only about $^1/_3$ of 1 per cent less than 1.4, the question naturally arises as to how much this small difference affects the values of Y given in the table. The answer is that for the pressure ratios used in the table the values of Y are about $1^1/_4$ per cent less than they would be with $k = 1.4$, so this difference is certainly not a serious one in engineering applications.

Authors' Closure

The authors are gratified that so many competent engineers have been interested enough in the paper to discuss it and to give words of commendation. In reply to Mr. Peterson, a standard temperature of 68 deg. fahr., rather than 60 deg., was selected because this is believed to be more nearly the average temperature for most engineering work, and because it corresponds exactly to the value of 20 deg. cent. commonly used.

Following the suggestion given by Mr. Robinson, there has been added to Fig. 1 a line which represents the values of specific heat computed on the basis of the work of Keyes, Beattie, and Bridgeman, with constants based on Roebuck's experimental values. This gives additional credibility to Swann's specific-heat values, as compared with the previous lower ones.

There are a number of detail criticisms regarding the exact numerical values which have been finally selected. The selected value, $(k - 1)/k = 0.283$, applies to air at 68 deg. fahr., 29.92 in. of mercury, and 36 per cent relative humidity if the value of c_p is as high as Swann's value. It applies to a relative humidity of 70 per cent, if the value of c_p is approximately as determined by the recent experiments of Roebuck (Proc. Am. Acad. Arts and Sci., vol. 60 (1925), p. 537). It applies to a higher relative humidity if the lower values of c_p shown in Fig. 1 are correct.

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However, the authors feel that the set which they have selected is at least as good as any other set of values which would accord on the average with the various items of experimental data, which would be thermodynamically self-consistent, and which would give an even value to three significant figures of the quantity $(k - 1)/k$, which is the direct number used in computing the tables for Y. The rounding off of some of the other constants, with retention of thermodynamic consistency, would require more places than $(k - 1)/k = 0.283$, and this is not justified under the circumstances.

APM-52-9

Elastic and Inelastic Behavior in Spring Materials

Progress Report No. 5 of the A.S.M.E. Special Research Committee on Mechanical Springs (Technological)

By M. F. SAYRE,[1] SCHENECTADY, N. Y.

SINCE the Fourth Progress Report of the committee was published, work has been continued at Union College on fundamental studies of the elastic behavior of spring materials. The purpose of this work has been to lay a sound theoretical basis for the whole problem of spring design, taking up those underlying problems which it would be uneconomical for any one manufacturer by himself to tackle, but which are vitally necessary to a real understanding of the whole problem. During the past six months it has been possible to clarify several points which were uncertain at the time the last report was written, and to open up several new points regarding the elastic behavior of spring metals in tension. These results may prove to be of value outside as well as within the field of spring design.

1 In the phosphor-bronze and spring-steel samples tested the modulus of elasticity has been shown to be greatly affected by overstressing. Overloading in tension beyond the yield point lowers the modulus of elasticity with respect to future lighter loads. Within limits, the greater the permanent elongation, the greater the reduction in modulus. Two weeks' rest at room temperature causes only a slight recovery. Specimens are being held for future tests at longer intervals. Such overloading as this, as in surging a spring, operates in two ways to increase the energy-storing capacity of a spring, first by lowering the modulus of elasticity, and second by raising the apparent elastic limit of the material. Further tests are desirable in order to show to what extent this gain is counterbalanced in other ways.

2 For years Hooke's law, that stress is proportional to strain, has been almost a fixture in engineering practice. Numerical results obtained seem to indicate a second-order correction to this law, not great enough to be perceptible for materials commonly used at low stresses, such as structural steels, but great enough to be noticeable at customary spring stresses in phosphor bronze and spring steel. With sufficiently accurate measuring instruments, the stress-strain line appears to be continuously curved from zero load upward, not because of minute departures from elastic behavior, but because the elastic line itself is curved. The modulus of elasticity in tension decreases as the load increases, the decrease in slope of the stress-strain line being of the general order of magnitude of 2 to 4 per cent. This effect was noted in last year's work, but an attempt was made to fit successive straight lines to the stress-strain curve, so obtaining an upper and a lower proportional limit, and two different moduli of elasticity. The work done since then has emphasized this curvature as a real elastic property, and indicated that the proportional limit properly so-called is not evidence of the elastic limit of the material, but tests have not yet been made on enough specimens to show whether the rate of curvature will vary with individual specimens, and whether such variation, if any, is indicative of the character of the material.

From theoretical reasons it is expected that in compression the modulus will increase as the load increases. Work is planned on the much more difficult problem of checking this phase of the work.

The practical effect of this curvature of the stress-strain line upon the stresses in a power or cantilever spring will be to shift the neutral axis toward the compression side, to slightly decrease the maximum stresses on the tension side, and to increase them on the compression side. The magnitude of the effect will be small.

3 In last year's report, mention was made of a discrepancy in the law of variation of internal friction, or elastic hysteresis, between tension and bending specimens. Later tests have shown that this discrepancy resulted from a sharp break in the law of hysteresis in tension which occurs at about the yield point. The precision with which it was possible to work last year was such that practically all our significant results in tension were limited to the upper load ranges, near or above the break in the curve. By multiplying the precision of the extensometer by five, so that one scale division of the extensometer now represents 1 part elongation in 500,000, and working with an extensometer with almost negligible instrumental hysteresis, it has been possible to obtain results on the lower part of the curve as well. For stresses below 160,000 lb. per sq. in., the actual width of the hysteresis loop (i.e., the amount of "backlash" which was shown by the specimen on descending loads as compared to ascending loads, at half full load) increased in these tests as some power of the maximum load varying between 2.5 and 3. The energy loss in hysteresis would vary as the maximum load raised to the 3.5 or 4 power. Above 160,000 lb. per sq. in., which is a point about midway between elastic limit and yield point and not far from where appreciable permanent set first begins, and up to maximum stresses of 180,000 lb. per sq. in., the hysteresis varies more nearly as the eighteenth power of the maximum stress. In specimens in bending, the outside fibers reach the yield point only one by one as the load is increased, and consequently no such sharp change in the law of hysteresis will be evident.

4 Any overloading, to stresses which exceed this same figure of 160,000 lb. per sq. in., acted to increase the future energy loss in hysteresis, even for small load ranges. Within limits, the greater the amount of overloading, the greater was the effect on the hysteresis. Recovery from the effect of the overloading, after two weeks at room temperature, was relatively slight. Further tests are planned to study the amount of recovery after longer periods.

The fact that permanent or quasi-permanent changes in both modulus of elasticity and hysteresis begin at about the same unit stress indicates that the two are tied together.

5 The data obtained as to the actual shape and character of the hysteresis loop are not yet complete. This information is of course likely to be of academic rather than of practical interest to spring manufacturers and users, but a number of questions have been asked about it. It seems to be more nearly of string-bean shape than like an ellipse, with pointed, eccentric ends, and with often a slight in-curve on the under side. The exact shape will

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depend upon the magnitude of the stress range, upon the speed with which the readings are taken, and upon how nearly the material has reached a cyclical state.

As stated in the preceding report, the experimental problems have not been entirely simple. The elastic changes in shape of a metal, particularly in tension or compression, are themselves small. Much of the work done has required the measurements of slight discrepancies in this elastic behavior of the order of magnitude of 0.00001 in., so calling for the highest available degree of precision, with repetition of work to allow for elimination of erratic results and for use of average values. It has been necessary to work in all cases practically to the limit of accuracy of the instruments used, and to apply all practicable precautions.

Details of Work

The testing machine used was an Amsler universal machine with 50,000 lb. maximum capacity, but with seven load ranges. This was a new machine just installed by the college, calibrated in Switzerland and again calibrated here, up to a load of 4000 lb., using standard 50-lb. weights. The results of that calibration are shown in Table 1. It was installed in a large ground-floor

TABLE 1 DEAD-LOAD CALIBRATION OF TESTING MACHINE

Weight, lb.	Actual error of machine in pounds at the load indicated — Scale setting — 5,000 lb.	10,000 lb.	25,000 lb.
0	+3 lb.	−2 lb.	−28 lb.
500	−2	−4	0
1000	+1	0	−10
1500	+5	+3	− 3
2000	−3	0	0
2500	+3	+8	+15
3000	−1	0	0
3500	−1	0	0
4000	+4	0	+ 5

room, practically free from vibration. The temperature at the machine was relatively uniform during the entire period of the tests with a maximum range from 68 to 78 deg. fahr. The range during any one day did not exceed 4 deg., and was usually not over 1 deg. A cloth curtain around the specimen and extensometer shielded them from direct drafts.

The extensometer used was of the Riehlé-Sayre type, developed by the author. For this work it was adjusted for an 8-in. gage length. Each of the 500 divisions on the scale represented an elongation of 1 part in 500,000, so that the full scale range represented 1 part in 1000, or a load range (for steel specimens) of 28,000 to 30,000 lb. per sq. in. Errors in absolute calibration of the extensometer are believed to be within 0.5 per cent. The extensometer is relatively free from instrumental hysteresis. The combined sensitivity of the testing machine and extensometer was such that successive readings, taken at the same nominal load reading (about 5000 lb.) on the machine, checked within one division on the extensometer, equivalent to about 3 lb. load on the specimen. This, of course, was true only provided the load variation on the machine was kept small enough so that the internal hysteresis of the specimen itself was not a factor.

The method of operation was so planned as to eliminate experimental errors as far as possible. The accuracy of the hysteresis figures depended upon the sensitiveness only, and not upon the absolute accuracy of the testing machine and extensometer, since they came as a difference of two readings taken at the same load and at nearly the same point on the extensometer scale. Values of modulus of elasticity were based on sets of readings, each taken over practically full scale range of the extensometer. As a result, any errors of calibration of the extensometer as a whole or in individual parts of the scale affected only the absolute value of the modulus of elasticity and did not appreciably affect the difference in readings, upon which the conclusions are based. Any index error in the zero setting of the testing machine would not appreciably affect the conclusions. A consistent error in the testing machine, of the same percentage at all loads, would affect the absolute values of the modulus but not appreciably the differences in modulus. Erratic errors, varying in magnitude from point to point, remain as a possible source of error.

Most of the work was done on samples of the same $^1/_2$-in. × $^1/_8$-in. oil-tempered 0.68 per cent carbon flat steel wire used for the work of the last report. The description of this wire as given by the manufacturers was as follows:

> A tempered annealed rod 0.5 in. × 0.162 in. is rolled and straightened and then lead-annealed at 0.160 in., rolled to 0.147 in., then to 0.140 in., lead-annealed at this stage, and then rolled successively to 0.135, 0.130, and 0.125 in.
>
> In hardening and tempering this wire, the furnace for heating is at 1550 deg. fahr., and the temperature of the tempering furnace is 900 deg. fahr.

The analysis of this material is as follows:

	Per cent
Carbon	0.68
Manganese	0.67
Sulphur	0.046
Phosphorus	0.036
Silicon	0.201

The hardness numbers obtained were:

Shore	51.7
Brinell	37.4
Rockwell	C39–C40

It has a yield point of about 175,000 lb. per sq. in. and an ultimate strength of 198,000 lb. per sq. in. Tests were also made on flat spring tempered sheet and round rod of phosphor bronze, and on a bar of nickel steel, both obtained locally.

In studying hysteresis the specimen was put through a series of cycles of loading and unloading between a nominal load and the desired maximum load in tension. After the specimen had approached a cyclical state, sets of readings were taken with the extensometer for three complete cycles, or more if necessary to obtain consistent values. In the work covered by the last report readings were taken at even load intervals over the entire cycle. Since then the sensitivity of the extensometer has been multiplied by 5 and this increased sensitivity had to be paid for in the form of reduced range. On the flat-wire steel specimens an actual load of 1600 lb. gave almost a full scale reading. Consequently, when working for instance within a load range from 100 lb. to 6400 lb., after running some preliminary cycles, the extensometer was set at zero for a 2400-lb. load. Readings were taken at 2800, 3200, 3600, and 4000 lb. Leaving the extensometer in place the load was increased to 6400, then lowered and readings taken at 4000, 3600, 3200, 2800, and 2400 lb. as the load decreased. The load was then lowered to 100 lb., raised to 2400 lb., and the cycle of readings repeated. In this way the width of the loop as measured by the differences in readings, ascending and descending, was taken at five points near the middle of its height. The sides of the loop are so nearly straight at this point that the average of these five differences, or rather the mean of three sets of five differences each, could be taken as the maximum width of the loop itself.

Such measurements were made using successively increasing maximum loads and the results are given in Table 2 and are shown graphically in Fig. 1. Repeat runs, dropping back to a lower maximum load after a cycle at a higher load had been completed, gave no appreciably changed results until the maximum load had reached nearly the yield point of the material. The results of sets of test runs taken after the yield point had been reached are also given in Table 2. The results are given in millionths of an inch per inch length of specimen, and the measurements involved are so minute and so close to the limits of accuracy of the instrument used that too great consistency must not be expected in the lower load ranges. Some tests are planned

for the future using a 50-ft. gage length under carefully controlled temperature conditions, and it is hoped that it will be possible to obtain higher precision in these tests.

In all of these measurements the extensometer readings were taken over the same load range of 1600 lb. except in those few cases where the hysteresis of the specimen was so great that some of the readings came off the scale. A study of these elongation readings gave the first definite indication that the true modulus of elasticity of the material changed as the load increased. Table 3 gives a set of the readings for one bar. It will be noticed that the change in length for a 1600-lb. load range with ascending and descending load checks within a reasonable limit of error, but as the actual load itself increases, the elongation also fairly consistently increases. In the second series of runs, taken after the specimen had been loaded to the yield point, the elongation readings were consistently higher than in the first series; and in the third series, after still greater overload, the readings were still higher.

With this as a clue a program of measurements was planned with the special purpose of studying the true elastic stress-strain curve. The most obvious method was to take measurements over the complete stress-strain curve itself, from zero load to the yield point. This immediately introduced the factor of elastic hysteresis, and also made it impossible to study the elastic elongation of the bar as separated from the effect of minute plastic yield which might be simultaneously occurring. To avoid these effects, the total curve was broken up into a series of short sections, each of short enough range so that the elastic hysteresis within that range would be small. For the first run a new specimen was loaded to 100 lb. and the extensometer set at zero. A series of load cycles was then run between 100 and 1600 lb. in order to develop any permanent elongation which was likely to occur within the 1600-lb. load range and to set the extensometer firmly in place on the specimen. The load was again brought to 100 lb., the extensometer again set at zero, and readings taken at 800 and 1600 lb. on ascending and descending loads for at least three cycles. The load was next brought to 1600 lb., and the extensometer rest to an 8-in. gage length at this loading. After a series of preliminary cycles as before, readings were now taken over the load ranges from 1600 to 3200 lb., ascending and descending. In that way the true slope of the stress-strain curve was found between the 1600- and the 3200-lb. load, corresponding to unit stresses of 26,800 and 53,600 lb. per sq. in., respectively. This was followed by a similar run from 3200 to 4800 lb. In some cases intermediate runs from 800 to 2400, and from 2400 to 4000 lb., were interpolated.

The load was now dropped to 100 lb., and the runs 100–1600 lb., 1600–3200 lb., and 3200–4800 lb. were repeated, both as a check and to determine whether the cycle from 3200 to 4800 lb. had perceptibly affected future behavior at lower loads. This second series was continued with runs at 4000–5600 lb. and at 4800–6400 lb., and then a third series was started beginning at

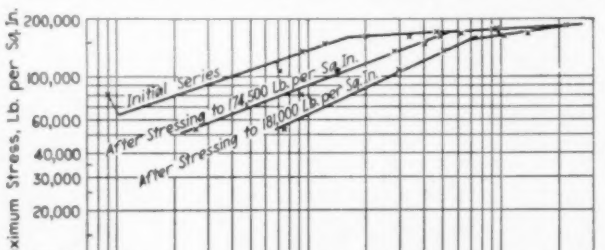


FIG. 1 MEAN WIDTH OF LOOP FOR HYSTERESIS IN TENSION
(Bars Nos. 17, 22, and 28, heat-treated flat steel wire.)

TABLE 2 MEAN WIDTH OF HYSTERESIS LOOPS IN TENSION TESTS OF HEAT-TREATED FLAT STEEL WIRE. 1/8 IN. × 1/2 IN.

Maximum load		—Width of loop, millionths of an inch per inch—			
Lb.	Lb. per sq. in.	Bar No. 17	Bar No. 22	Bar No. 28	Mean
First Series					
3200	53590	4.4	0	0	0
4800	80390	1.6	0	1.2	0.9
6400	107180	10.8	6.4	4.0	7.1
8000	133980	12.4	12.4	3.2	9.3
8800	147380	14.8	12.4	9 2	12.1
9600	160780	25.2	15.2	19.6	20.0
10000	167480	46.4	61.6	30.0	46.0
10400	174180	126.4	56.4	100.0	94.3
Second Series, After Loading to 174,000 Lb. per Sq. In.					
3200	53590	3.6	...	1.6	2.6
4800	80390	12.4	10	5.2	9.2
6400	107180	17.6	16.4	9.2	14.4
8000	133980	27.2	35.2	21.2	27.9
8800	147380	...	40.8	...	40.8
9600	160780	56.0	52.8	36.0	48.3
10000	167480	...	62.8	...	62.8
10400	174180	114.0	93.2	68.8	92.0
10800	180870	280.0	189.2	182.0	217.1
Third Series, After Loading to 181,000 Lb. per Sq. In.					
3200	53590	6.8	6.8	8.8	7.5
4800	80390	14.0	18.4	14.4	15.6
6400	107180	30.8	32.0	24.8	29.2
8000	133980	51.6	59.6	40.8	50.7
9600	160780	98.8	133.2	80.4	104.1
10400	174180	142.0	150.8	120.0	137.6
Fourth Series, After Two Weeks' Rest					
3200	53590	0	7.6	...	3.8
4800	80390	12.4	16.0	...	14.2
6400	107180	22.8	29.2	...	26.0
8000	133980	47.2	59.6	...	53.4
8800	147380	68.0	74.8	...	71.4
9600	160780	90.4	106.8	...	98.6
10000	167480	102.4	124.0	...	113.2
10400	174180	125.2	132.4	...	128.8

TABLE 3 MODULUS-OF-ELASTICITY DETERMINATIONS AS TAKEN FROM HYSTERESIS LOOPS; HEAT-TREATED FLAT STEEL WIRE, BAR NO. 17

Maximum stress	Stress for readings	Temp., deg. fahr.	Strain readings			Corrected readings (See key below)			Modulus of elasticity
			Up	Down	Mean				
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)
First Series									
53	13–40	72.5	459.0	454.3	456.6	456.3	456.1	456.3	29.31
80	27–53	71.5	456.7	453.7	455.2	455.0	454.7	455.0	29.40
107	40–67	72	457.0	456.3	456.6	456.3	455.9	456.3	29.31
134	53–80	72	457.0	456.3	456.6	456.3	455.8	456.3	29.31
147	60–87	73	457.0	456.7	456.8	456.5	455.9	456.5	29.30
160	67–94	74	458.7	460.7	459.7	459.3	458.7	459.3	29.12
167	70–97	73	462.3	462.0	462.1	461.8	460.9	461.8	28.96
174	74–94	73.5	472.9	470.7	471.8	471.4	470.1	470.6	28.42
Second Series, After Loading to 174,000 Lb. per Sq. In.									
53	13–40	74	465.7	465.0	465.3	464.9	464.1	464.1	28.80
80	27–53	74.5	464.0	464.0	464.0	463.6	462.7	462.8	28.90
107	40–67	75	465.0	465.0	465.0	464.5	463.5	463.7	28.84
134	53–80	75.5	466.0	467.3	466.6	466.1	465.0	465.3	28.74
160	67–94	76	470.3	469.3	469.8	469.3	468.1	468.5	28.55
174	74–94	74	469.3	476.9	473.1	472.7	471.3	471.9	28.34
181	77–90	75	486.0	495.0	490.5	490.5	488.0	488.6	27.37
Third Series, After Loading to 181,000 Lb. per Sq. In.									
53	13–40	75	477.7	476.3	477.0	476.5	474.5	474.5	28.18
80	27–53	76	472.3	472.0	472.1	471.6	469.4	469.6	28.48
107	40–67	76	475.7	474.8	475.3	474.8	472.5	472.8	28.29
134	53–80	76	478.0	477.3	477.6	477.1	474.7	475.1	28.15
160	67–87	76	480.0	483.6	481.8	481.3	478.8	479.3	27.90
174	74–94	74	492.0	494.6	493.3	492.9	490.4	490.9	27.24
Fourth Series, After Two Weeks' Rest									
53	13–40	68	468.0	472.7	470.3	470.3	468.3	468.3	28.55
80	27–53	68	471.7	475.0	473.4	473.4	471.2	471.4	28.37
107	40–67	68	475.7	479.0	477.4	477.4	475.1	475.4	28.13
134	53–74	68	477.3	475.6	476.4	476.4	474.0	4.4	28.19
147	60–80	69	479.1	484.0	481.6	481.5	479.1	479.5	27.89
160	67–87	70	480.0	485.7	482.8	482.7	480.2	480.7	27.82
167	70–90	70	487.5	487.5	487.5	487.4	484.9	485.4	27.55
174	74–94	70	485.0	486.7	485.8	485.7	483.2	483.7	27.65

KEY TO TABLE 3

Columns 1 and 2 give the maximum stress in the cycle, and the stress range over which readings were taken, both in thousands of pounds per square inch.

Columns 4 to 6 give the measured elongation, in parts per 500,000, resulting from a load increase or decrease of 1600 lb. or 26,746 lb. per sq. in. Each figure represents the mean of at least three results.

Column 7 gives the same figures as corrected to 68 deg. fahr. for the effect of temperature upon modulus of elasticity.

Column 8 includes also a correction for the effect of the total change in cross-section, both elastic and plastic. Poisson's ratio was assumed as 0.27.

Column 9 is corrected for temperature and for permanent, plastic change in cross-section only.

Column 10 gives moduli of elasticity in millions of lb. per sq. in. based on the results in column 9.

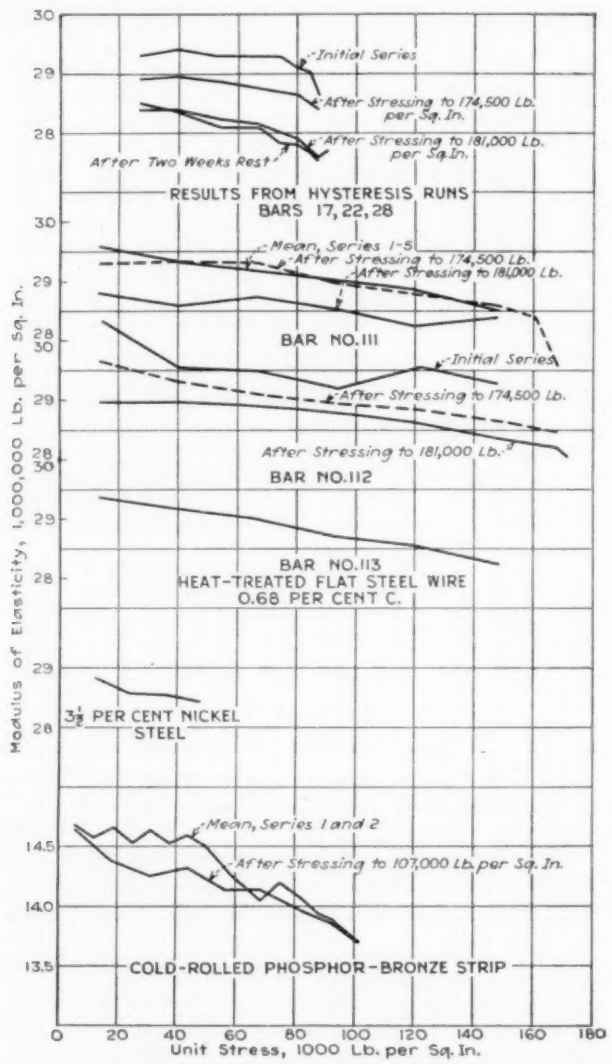


FIG. 2 RESULTS OF MODULUS-OF-ELASTICITY DETERMINATIONS

unit strain can be obtained by dividing it into the constant stress range of 26,800 lb. per sq. in.

A study of these figures reveals a relatively steady increase with increasing load in any one series of runs. This of course indicates a decreasing modulus of elasticity. From series to series, the variations are within experimental limits of error until after the fifth series. At the end of the fifth series, the load was raised to 10,400 lb., equal to 174,500 lb. per sq. in., and held there for some time. This was sufficient to cause permanent elongation of the order of magnitude of two-tenths of 1 per cent; and resulted in a slight decrease in the average values of modulus of elasticity. A further overload, to 181,000 lb. per sq. in., equivalent to an elongation of one-half per cent, resulted in a much more perceptible decrease in modulus. Summarized values are given in Table 5.

Before computing moduli of elasticity, it will be well to make certain corrections.

1 It has been known for some time that the modulus of elasticity decreases, and the flexibility increases, with increasing temperature. All readings were therefore reduced to find the equivalent reading as of 68 deg. fahr., correcting[2] 0.014 per cent per deg. fahr.

2 Any permanent elongation of the specimen results in an equivalent decrease of area of cross-section, so that larger elongations would naturally be expected as a result of a given actual load. To compensate for this, all readings made after the speci-

TABLE 5 MODULUS-OF-ELASTICITY DETERMINATIONS; HEAT-TREATED FLAT STEEL WIRE, BAR NO. 111

Load range, 1000 lb. per sq. in.	Temp., deg. fahr.	Mean strain as read	Reading corrected for Temp.	Reading corrected for Elong.	Modulus of elasticity × 10⁶ Bar No. 111	Values for comparison Bar No. 112	Values for comparison Bar No. 113
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
Mean, First to Fifth Series							
1.6– 27.0	76	453.6	453.1	453.1	29.58	30.33	29.37
27.0– 54.0	76	457.3	456.8	456.8	29.34	29.54	29.17
54.0– 80.5	76	460.0	459.5	459.5	29.17	29.47	29.01
80.5–107.5	76.5	462.8	462.3	462.3	29.00	29.19	28.71
107.5–134.0	77.3	464.9	464.3	464.3	28.87	29.54	28.45
134.0–161.0	76.2	471.0	470.5	470.5	28.50	29.25	28.23
Sixth Series, After Loading to 174,500 Lb. per Sq. In.							
1.6– 27.0	74	458.5	458.1	457.4	29.30	29.63	
27.0– 54.0	74.5	458.0	457.6	456.9	29.33	29.32	
54.0– 80.5	74.5	458.3	457.9	457.2	29.31	29.10	
80.5–107.5	75	463.8	463.4	462.7	28.96	28.95	
107.5–134.0	75	470.8	470.4	469.7	28.53	28.83	
134.0–161.0	75.5	470.0	469.5	468.8	28.59	28.63	
147.5–174.5	75.5	473.3	472.8	472.1	28.38	28.52	
154.5–181.0	76	489.3	488.8	486.2	27.57	28.46	
Seventh Series, After Loading to 181,000 Lb. per Sq. In.							
1.6– 27.0	78	468.8	468.1	465.5	28 79	28.97	
27.0– 54.0	78	472.0	471.3	468.7	28.59	28.96	
54.0– 80.5	78	469.5	468.8	466.2	28.74	28.91	
80.5–107.5	78	473.3	472.6	470.0	28.51	28.79	
107.5–134.0	78	478.0	477.3	474.7	28.23	28.63	
134.0–161.0	78	474.5	473.8	471.2	28.44	28.34	
154.5–181.0	..	...	...	...	...	28.20	
158.0–184.5	..	...	...	...	...	28.03	

TABLE 4 MODULUS-OF-ELASTICITY DETERMINATIONS; HEAT-TREATED FLAT STEEL WIRE, BAR NO. 111

[Actual elongation, in 500,000ths, resulting from a stress increase or decrease of 26,650 lb. per sq. in. (1600 lb. actual load).]

Load range 1000 lb. per sq. in.	Series 1 Up	Series 1 Down	Series 2 Up	Series 2 Down	Series 3 Up	Series 3 Down	Series 4 Up	Series 4 Down	Series 5 Up	Series 5 Down	Mean of Series 1–5 Up	Mean of Series 1–5 Down	Series 6 Up	Series 6 Down	Series 7 Up	Series 7 Down	Load range, 1000 lb. per sq. in.
1.6– 27.0	455.5	454.4	452.3	451.5	453.7	453.3	453.0	452.3	455.5	454.4	454.0	453.2	458.7	458.3	469.0	468.7	1.6– 27.0
13.5– 40.5	454.3	453.3	...	...	...	...	...	...	...	...	...	...	...	...	...	...	13.5– 40.5
27.0– 40.5	457.7	456.7	455.7	455.7	461.0	460.8	455.7	455.7	457.7	457.0	457.6	457.1	458.0	458.0	472.3	471.7	27.0– 54.0
40.5– 67.2	462.8	462.7	...	...	...	...	...	...	...	...	...	...	...	...	...	...	40.5– 67.2
54.0– 80.5	462.0	461.0	459.3	458.3	460.7	460.3	460.0	460.0	459.3	458.7	460.3	459.7	458.7	458.0	470.0	469.0	54.0– 80.5
67.2– 94.0	...	...	461.0	459.7	...	...	...	...	...	...	...	...	...	...	...	...	67.2– 94.0
80.5–107.5	...	...	461.7	460.3	462.7	462.0	465.0	464.3	463.3	463.0	463.2	462.4	464.3	463.3	473.7	473.0	80.5–107.5
94.0–120.8	...	...	...	...	461.7	460.7	...	...	...	...	...	...	...	...	...	...	94.0–120.8
107.5–134.0	...	...	...	...	466.7	465.0	465.0	463.7	465.0	464.0	465.6	464.2	473.3	468.3	478.7	477.3	107.5–134.0
120.8–147.5	...	...	...	...	...	...	466.7	464.7	...	...	...	...	...	...	...	...	120.8–147.5
134.0–161.0	...	...	...	...	...	...	475.7	469.7	471.7	466.7	473.7	468.2	471.7	468.3	477.3	471.7	134.0–161.0
147.5–174.5	...	...	...	...	...	...	...	...	...	...	...	...	474.0	472.7	...	...	147.5–174.5
154.2–181.0	...	...	...	...	...	...	...	...	a	...	...	...	492.0 b	486.7	...	...	154.2–181.0

a Load carried to 174,500 lb. per sq. in. at end of this series.
b Load carried to 181,000 lb. per sq. in. at end of this series.

100–1600 lb., and so on. Complete sets of readings for one bar are given in Table 4. Each value given there represents a mean of results from three or more cycles taken for that load range adjusted to represent the elongation corresponding to an actual load increase or decrease of 1600 lb. or 26,800 lb. per sq. in. These values of unit strain are in parts per 500,000, as actually read, and should be multiplied by two to convert to parts per million. The actual modulus of elasticity for any one figure of

[2] This factor is taken from work done by A. L. Kimball, Jr. and D. E. Lovell, and reported in the *Physical Review* of July, 1925, p. 121.

TABLE 6 MODULUS OF ELASTICITY BASED ON HYSTERESIS-LOOP RUNS

(Mean of results from bars Nos. 17, 22, and 28, heat-treated flat steel wire.)

Load range, 1000 lb. per sq. in.	Modulus of elasticity × 10^6 — Initial series	After loading to 174,500 lb. per sq. in.	After loading to 181,000 lb. per sq. in.	Rerun after 2 weeks
13.5– 40.5	29.31	28.93	28.41	28.52
27 – 54	29.42	28.98	28.41	28.39
40.5– 67	29.31	28.88	28.25	28.11
54 – 80.5	29.28	28.78	28.16	28.11
60.5– 87.5	29.29	...	...	27.85
67 – 94	29.10	28.65	27.92	27.79
70.5– 97	29.05	...	...	27.67
74 –100.5	28.65	28.40	27.57	27.53
77 –104	...	...	27.70	...

TABLE 7 MODULUS OF ELASTICITY OF COLD-ROLLED PHOSPHOR-BRONZE STRIP NO. 306

Load range, 1000 lb. per sq. in.	Modulus of elasticity × 10^6 — Initial series	After stressing at — 75,600 lb. per sq. in.	107,100 lb. per sq. in.
1.5– 12.6	14.67	14.70	14.64
6.3– 18.9	14.53	14.61	...
12.6– 25.2	14.69	14.62	14.36
18.9– 31.5	14.49	14.55	...
25.2– 37.8	14.63	14.62	14.25
31.5– 44.1	14.52	14.52	...
37.8– 50.4	14.62	14.52	14.32
44.1– 56.7	14.47	14.50	...
50.4– 63.0	14.24	14.36	14.15
56.7– 69.3	14.11	14.19	...
63.0– 75.6	14.02	14.09	14.14
69.3– 81.9	...	14.19	...
75.6– 88.2	...	14.07	13.97
81.9– 94.5	...	13.94	...
88.2–100.8	...	13.85	13.86
94.5–107.1	...	13.68	13.69

TABLE 8 MODULUS OF ELASTICITY OF 3½ PER CENT NICKEL-STEEL BAR NO. 476

Load range, 1000 lb. per sq. in.	Modulus of elasticity × 10^6 — First series	Second series	Mean
0.8–24.0	28.79	28.85	28.82
12.0–36.1	28.68	28.43	28.55
24.0–48.1	28.54	28.53	28.53
36.1–60.1	28.50	28.33	28.42

men had been overstrained were corrected so as to give the elongation corresponding to the original area.

3 Any elastic elongation results in an equivalent elastic decrease in cross-section, but not in the same ratio. The actual amount depends upon the value of Poisson's ratio. It is not felt that this correction properly should be made. It would in any event not be likely to exceed one-quarter of 1 per cent.

4 The extensometer was reset at an 8-in. gage length before each run, whereas ordinarily an extensometer would be set at the given gage length at zero load and left on while the gage length gradually increased with increasing load. This correction may reach a maximum of one-half per cent at the highest loads. If made, it would still further emphasize the decrease in modulus of elasticity at high load.

Values of moduli of elasticity at various loads, with the first two corrections applied, are given for various spring-steel samples in Tables 3 to 6. Moduli of elasticity for hard-rolled spring-tempered phosphor-bronze samples $^1/_8$ in. thick × $^1/_2$ in. wide are given in Table 7 The values are uncorrected for temperature or set. In these cases the first correction is uncertain in magnitude, and the correction would in any event not be large. Additional tables and full data are on file ready for inspection.

In closing, reference should again be made to the fact that a large amount of time and attention has been required for each series of tests, and that therefore only a small variety of samples have been tried. The results, as representing a general law, are consequently tentative. On the other hand, the evidence in favor of the curved stress-strain line is increased by the fact that for some time past the theories regarding the nature of atomic attraction and repulsion have called for just such curvature. This aspect of the case can well be taken up at some other time.

Acknowledgment should be made to Union College, for the use of its facilities in this investigation, and to Mr. Henry W. Sory, through whose skill and care in experimental operations these sets of readings have been made possible.

Discussion

W. J. Sweetser.[3] At the University of Maine some work has been done on soft steel of approximately 0.15 carbon and in investigating some of the properties after the steel has been stretched beyond the proportional limit and well into the plastic region. This work has been done in torsion on an Amsler testing machine of 1000 ft-lb. capacity. A troptometer has been used of such radius that 1 in. of arc on the scale was equal to 1 deg. of twist, and this arc has been divided into 0.1 deg. Great care was taken in laying out this arc and in making the divisions; 30 deg. being laid off and the chord measured to check for accuracy. The 30-deg. scale was then divided into single degrees by spacing with dividers, and each degree was subdivided into tenths. A magnifying glass was used in making the latter divisions and in marking the radial lines.

The modulus of elasticity of this steel was determined after it had been stressed to approximately 62,500 lb. per sq. in. maximum fiber stress in torsion. The load was then released to zero and the test piece allowed to rest for a week to recover its normal elasticity, as it was found that this steel will change very materially during that period, but will not make much further change after a longer rest. The data of Table 9 were taken a month and a half after the steel was first overstressed, so that it had over six weeks in which to recover. The readings were over an 8-in. gage length, and the load was applied to give equal angles of twist, then the readings of the twisting moment were taken from the machine. These latter readings are on a cylinder which is divided into divisions of 10 ft-lb., a vernier gives the unit figure, and one can estimate to one-quarter of the unit figure, due to the large divisions on the scale. This machine, however, has not been accurately calibrated. The settings on the troptometer are determined by sighting over the end of a pointer through a peep sight over 8 in. distant, and as the pointer moves close to the troptometer scale quite accurate settings for a given angle of twist can be made when the twisting force is applied. These settings of the troptometer are in error less than one-hundredth of a degree on the scale, and as the increments taken were 0.8 deg., which represented a twist on the piece of 0.1 deg. per inch of length, it is believed that the results are within the degree of accuracy which the machine will permit. The three sets of data quoted are from the same specimen and were taken over different divisions of the troptometer so that individual errors on the troptometer scale as well as errors due to the personal equation are included in the settings for equal angles of twist.

TABLE 9 MODULUS-OF-ELASTICITY DETERMINATIONS TAKEN FROM OVERSTRESSED MILD STEEL OF ABOUT 0.15 CARBON, MEAN DIAMETER 0.8738 IN.

Degrees of twist per inch of length	Twisting moment, ft-lb. — 1	2	3	Mean	Torsional stress, lb. per sq. in.	Diff.	Torsional modulus of elasticity × 10^6
0.0	0.0	0.0	0.0	0.0	...	...	...
0.1	97.0	97.0	98.0	97.3	8610	8910	11.69
0.2	193.5	194.0	193.75	193.8	17750	8840	11.59
0.3	289.0	290.0	289.75	289.6	26530	8780	11.51
0.4	384.0	382.25	382.5	382.9	35060	8530	11.18
0.5	476.0	474.25	473.0	474.4	43460	8400	11.02
0.6	563.25	563.0	562.0	562.8	51560	8100	10.62

The maximum stress while taking this data was more than a 1000 lb. per sq. in. less than the maximum to which the material had been subjected, and no creep could be detected when the piece was stressed to this value. The piece was strained in increments of 0.1 deg. per inch of length up to a maximum of 0.6

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deg., then released to 0 deg. This latter setting was determined by bringing the machine to zero twisting moment and then reading the troptometer. In only one of the three sets of data did the troptometer apparently fail to return to the original zero reading and then by not over 0.00125 deg. per inch of length.

As will be noted from the results given in Table 9, the modulus of elasticity in torsion decreases with the increase in stress. Over this range the specimen complies with the definition of an elastic material in that it shows no permanent set after the load is released.

R. L. Templin.[4] The author has presented some interesting data pertaining to some spring materials. The data undoubtedly have been obtained by carefully considered methods and well-designed apparatus. Unfortunately, however, some of the conclusions reached by the author are not in agreement with similar data obtained in tests of other materials.

In the laboratories of the Aluminum Company of America tests have been made of a number of different alloys of aluminum in both the wrought and cast conditions, using a Martens mirror extensometer with which unit deformations could be determined to 0.000005 in. In Table 10 are shown the observed changes in the tensile modulus after an initial test carried to the stress value indicated for a few specimens.

TABLE 10 CHANGES IN THE TENSILE MODULUS AS A RESULT OF SECOND LOADING

First loading stress, lb. per sq. in.	Proportional limit (from plotted data) lb. per sq. in.	Change in modulus,[a] per cent
8,100	2,900	— 1.9
6,100	1,900	0
7,000	1,800	0
28,000	10,000	0
22,000	6,000	0
24,000	7,000	— 10.9
13,000	3,000	— 2.7
24,000	7,000	— 0.9

[a] Second loading value compared to first loading value.

As the observed changes were of the same magnitude as the probable errors involved, it is doubtful that any changes in modulus actually occurred.

Generally speaking, one usually expects an increase in the modulus value with successive loadings, provided the material is stressed appreciably above the proportional limit or above the yield point. The initial stress-strain deformations include bending deformations, deformations due to relief of internal strains resulting from cold working or heat treatment, deformations due to misalignment of testing-machine heads and grips, non-axial application of load, etc. These non-uniform strains are largely equalized by overstressing, with the result that an increase in modulus would be anticipated.

If the conclusions of the author to the effect that cold working, which is a form of overstressing, decreases the tensile modulus by as much as 10 to 15 per cent are valid, how can the fact be explained that copper and aluminum both have the same modulus values after reductions in area, by cold drawing or rolling, of more than 90 per cent?

In view of these points it would appear unfortunate if the author attempted to draw too fundamental conclusions from his data and assume that they were generally applicable to all or even to most metals.

Dr. A. Nadai.[5] So far as the phenomenon of elastic hysteresis is concerned opinions have been expressed that such non-elastic properties of the materials of construction may be explained by certain processes which take place in the atomic structure of the crystal grains. A mechanical model which had been proposed some years ago in a paper published in England seems to account for a number of observations which are known under the heading mentioned as the phenomena of elastic hysteresis (loops in stress-strain diagram). In this mechanical model, ordinary friction was an essential part. It is well known that such materials as cast iron or sandstone show a pronounced elastic hysteresis. This type of material must be imagined as consisting either of grains with loose connections or of grains of two materials with very different properties of strength. A well known example is cast iron which has the soft and weak graphite particles imbedded in a more solid metallic mass. In a piece of such a material under compressive stress many grains will be constrained to slide a little under the pressure. For example, in a cast-iron piece the soft graphite inclusions will permit small tangential movements of the grains. Hence, ordinary friction will be present between the grains and will retard the relative movement of groups of grains, causing the phenomena of elastic hysteresis.

John M. Lessells.[6] In this paper the work on mechanical springs is divided into two parts. There is an engineering side, which is concerned with the effect on the fatigue strength of such factors as surface defects and decarburization, as brought to notice by Mr. Eaton. There is also a scientific side, as represented by the work of Professor Sayre. Both are important, and to allow of sufficient rapidity of progress both must be studied in parallel.

Coming to Professor Sayre's report, the impression has been formed that there is some uncertainty regarding hysteresis at low stresses. There is evidence of this from other fields. If we consider the work of Mr. Kimball, we find that internal friction is expressed as a function of (stress)^2 and apparently this holds for zero stress. There is further evidence from atomic theory. The generalized equation for stress and strain is

$$e = \frac{1}{E}(p + b_1 p^2 + b_2 p^3 + \ldots\ldots\ldots\ldots\ldots)$$

where p = stress in lb. per sq. in.
e = strain
E = modulus of elasticity
b_1, b_2 = constants

Hence it is seen that Hooke's law is only true if the other terms but the first be neglected.

The work as outlined in the report is nevertheless of great interest to engineers.

Jos. Kaye Wood.[7] Although the experimental results presented in the paper have no immediate value in so far as direct application to engineering is concerned, they are extremely important to the solution of the more fundamental problems of fatigue, creep, proportional limit, and other similar properties. This fact will be demonstrated more completely in future reports of the committee. In direct practical application, Hooke's law may still be considered as a straight-line characteristic. This is true even for high stresses in which the continuous and minute deviation from Hooke's law causes an ultimate cumulative change in the modulus of elasticity varying from 2 to 4 per cent for the materials tested. This change might be of sufficient importance, however, to warrant serious consideration in the design of high-precision springs, such as those used in sensitive length-measuring instruments.

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Both Joffe and Griffith speak of a slight curvature in Hooke's law, the former claiming to have made highly sensitive tests which showed a deviation of small magnitude. He attempts a theoretical explanation of this deviation based upon the electrostatic theory of crystal lattices. The writer will describe Joffe's theory briefly and without apology, because future progress in the fundamental knowledge of the physical properties of metals can be attained only by an understanding of the nature of atomic forces. According to this theory, the force exerted by two atoms constitutes a balance established between the attractive and repulsive forces. Symbolically, the force or stress

$$f = \frac{\delta E}{\delta r} = \frac{mA}{r^{m+1}} - \frac{nB}{r^{n+1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

in which m is the exponent of attraction and n is the exponent of repulsion; A is the attraction and B is the repulsion; while r is the distance between the atoms. Griffith calls f_0 the intrinsic pressure, due to the thermal vibrations of the atoms and of which the total heat of vaporization is a measure. That is, Equation [1] may be presented by an infinite convergent series of the form

$$f = f_0 + f'\Delta r + \frac{1}{2!} f''(\Delta r)^2 + \frac{1}{3!} f'''(\Delta r)^3 + \ldots\ldots [2]$$

in which Δr is the strain. This is not a straight-line relation. Hooke's law is represented by the single term $f' \Delta r$ only, which assumes erroneously that all other terms are equal to zero. In the normal state (that is, at absolute zero temperature) the thermal pressure f_0 does equal zero, so that the repulsive force just balances the attractive force at an equilibrium distance r_1 that is

$$\frac{mA}{r_0^{m+1}} = \frac{nB}{r_0^{n+1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

Applied stress, if it is tensile, assists the repulsive force much as the thermic pressure does, while if the applied stress is compressive, it assists the attractive force. Hence we should expect to find an increase in the modulus of elasticity in compression instead of a decrease, as was found for tension in this investigation. Griffith checks with this view and speaks of the curvature of Hooke's law as being due to the diminution of the modulus at very high stresses.

In the first tests conducted for the committee by A. Hoadley,[8] the testing equipment was arranged so as to induce a uniform flexural stress in the specimen. The resulting stress gradients therefore showed no shift of the neutral plane from the center of gravity. Now in the cold coiling of helical springs by the inside-mandrel method,[9] a considerable tension is put on the wire which combined with flexure causes an appreciable shift of the neutral plane from the center of gravity. This fact should be given some consideration in the further study of trapped stresses. In the mass production of cold-coiled helical springs by the outside-coiling method, however, very little tension is put on the wire, with the result that no shift of the neutral plane occurs. Consequently, this method by which 90 per cent of all cold-coiled springs are made, should be more suitable from the trapped-stress standpoint.

[8] Research Associate of the Spring Research Committee.

[9] "Design of Helical Springs," J. K. Wood, *Amer. Mach.*, 1921, vol. 54, pp. 854–7.

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The Effect of Occasional Overstress and Subsequent Annealing on Chain Steel and Welded Chain Steel

By H. IVAN ANDREWS,[1] ENFIELD, ENGLAND

The results of the tests reported in this paper indicate that occasional overload tends to increase the static and fatigue strength of unwelded chain steel at the expense of the ability to resist the energy of shocks. They also indicate that the effect of annealing after overstress is to reduce the static and fatigue strength of the unwelded metal to values as low or lower than those for the metal before overstress, and that the benefit of annealing consists in the complete or partial restoration of the shock-resisting capacity of the metal.

ONE of the most pertinent problems with which the chain-making industry is now faced is the heat treatment of chains in service. It is well known that chain material suffers considerably under constant handling or under occasional severe loading, and it has become customary for many chain users periodically to remove chains from service for heat treatment, usually an anneal following a slight overstress (stress above working load).

Several investigations of service damage to chains and of the effect of annealing have been made, of which two may be noted here. C. G. Lutts[2] made tests which showed a distinct development of brittleness in single links after overstress. Studying wrought-iron chains, H. J. Gough,[3] of the British National Physical Laboratory, made an extensive study of the damage due to repeated localized stress at points of contact of links.

The tests described in this paper were undertaken in order to study the effect of a single overload on chain material, and the effect of subsequent annealing. Tests were also made on welded material. The test results do not give any answer to the question as to the effect of repeated stresses at points of contact of links, but do give some information as to the effect of a few overloads, such as most chains receive in service, both on chain stock and on welded chain stock.

The tests were carried out by the writer in the Materials Testing Laboratories of the University of Illinois, under the direction of Prof. H. F. Moore, and formed a part of the work of the Engineering Experiment Station of that University.

A supply of representative chain material was obtained through the courtesy of the American Chain Company, who also supplied a number of lengths of the same material containing welds made by the electric-contact method. The material was a high grade of low-carbon steel, commonly used in the manufacture of chains. The unwelded chain material was supplied in the form of $^1/_2$-in.-diameter round rod, bright drawn, and wound into an open coil. From this coil a number of pieces each 22 in. long were cut, and numbered in the order in which they were taken from the coil. As these pieces were still curved they were heated to approximately 1600 deg. fahr. in an electric furnace and straightened, first by hammering, and finally in a press. To eliminate the effects of unequal hot working as far as possible, the straightened pieces were normalized by heating to 1600 deg. fahr. and cooling in air. This normalizing was intended to assure that the whole of the material was reduced to a fairly standardized condition for further treatment and testing.

The welded pieces were straight as received, and it was unnecessary to treat them further, particularly as they were in the condition of actual chain welds. The excess of metal at the weld was, however, removed by turning or grinding down to the diameter of the rod.

Before deciding on the amount of load to be used in overstressing specimens it was necessary to determine the yield point and the tensile strength of the material as normalized. This was done by means of a 100,000-lb. Amsler testing machine fitted with a semi-autographic[4] rig for drawing a stress-strain diagram. Several specimens of normalized rod and of welded rod were thus tested. For the normalized rod the mean value for yield point was 34,200 lb. per sq. in., and for tensile strength, 48,200 lb. per sq. in. For the welded rod the yield point (poorly marked) was 50,800 lb. per sq. in. and the tensile strength 60,200 lb. per sq. in. (see Table 1).

In order to compare the effects of various treatments, the specimens of normalized rod were divided systematically into five groups as follows:

Group 1 Specimens tested in normalized condition
Group 2 Specimens loaded in Amsler machine to a stress of 35,500 lb. per sq. in. (slightly above yield point)
Group 3 Specimens loaded to a stress of 45,700 lb. per sq. in. (nearly to tensile strength)
Group 4 Specimens loaded to a stress of 35,500 lb. per sq. in. and annealed at 1700 deg. fahr.
Group 5 Specimens loaded to a stress of 45,700 lb. per sq. in. and annealed at 1700 deg. fahr.

The welded specimens were divided into three groups as follows:

Group W1 Specimens as received
Group W2 Specimens loaded to a stress of 45,700 lb. per sq. in.

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[2] Proc. Am. Soc. for Test. Matls., vol. 24 (1924), part II, p. 651.

[3] Proc. British Inst. Mech. Engrs., 1928, vol. I, p. 293.

[4] For a description of the semi-autographic method of drawing stress-strain diagrams, see Proc. Am. Soc. Test. Matls., vol. 17 (1917), part II, p. 589.

Group W3 Specimens loaded to a stress of 45,700 lb. per sq. in. and annealed at 1840 deg. fahr.

The stress of 45,700 lb. per sq. in. was used for the welded material to correspond with the heaviest loading used for the normalized rod. The metal in the weld was found to be markedly brittle and the annealing was carried to the high temperature of 1840 deg. fahr. in order that it might be as thorough as possible.

The tests carried out included static tension tests, made on the Amsler machine, impact tension tests made on a Charpy pendulum machine fitted with attachment for tension specimens,

The general results of the tests are given in Table 1. In considering the results of the tests it must be noted that, with the exception of the direct tension tests the test conditions in the laboratory do not correspond exactly with conditions met in service. However, it must be remembered that the object of the tests was to make a comparative study of the effects of different conditions of overstress and annealing, and for such a study the test results are believed to have significance. For example, it is obvious that the fatigue tests consisted in the application of a large number of *reversals* of stress, whereas in service, chain material is subjected to repetitions of stress in the same direction.

TABLE 1 RESULTS OF TESTS OF CHAIN STEEL AND WELDED CHAIN STEEL

Group	Material and treatment	Static tension tests: Yield point, lb. per sq. in.	Static tension tests: Tensile strength, lb. per sq. in.	Fatigue test; endurance limit, lb. per sq. in.	Impact tensile test; energy for fracture, ft-lb.	Charpy notched-bar test; energy for fracture, ft-lb.
1	Chain steel without welds; normalized	34,200	48,200	26,000	111.4	38.0
2	Chain steel without welds; normalized and stressed to 35,500 lb. per sq. in.	46,000	56,300	29,000	101.2	36.1
3	Chain steel without welds; normalized and stressed to 45,700 lb. per sq. in.	60,400	62,100	33,000	100.6	28.6
4	Chain steel without welds; normalized, stressed to 35,500 lb. per sq. in. and annealed at 1700 deg. fahr.	23,200	40,000	23,000	111.9	26.4
5	Chain steel without welds; normalized, stressed to 45,700 lb. per sq. in., and annealed at 1700 deg. fahr.	24,700	40,600	23,000	103.6	28.3
W1	Welded chain steel as received	50,800	61,800	26,000	66.8	..
W2	Welded chain steel stressed to 45,700 lb. per sq. in.	....	....	25,000	64.8	..
W3	Welded chain steel stressed to 45,700 lb. per sq. in. and annealed at 1840 deg. fahr.	....	....	24,000	81.9	..

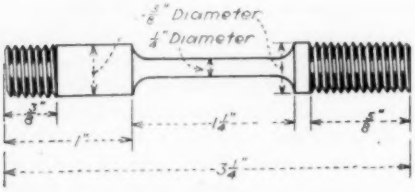


FIG. 1 TENSILE IMPACT SPECIMEN

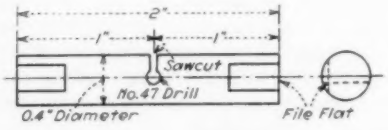


FIG. 2 SPECIAL CHARPY NOTCHED-BAR SPECIMEN FOR CHAIN STEEL

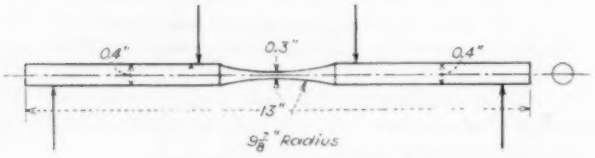


FIG. 3 FATIGUE SPECIMEN

Charpy notched-bar tests, made on the unwelded rod material only, and fatigue tests in reversed bending made on rotating beam machines of the Sondericker (or "Farmer") type. The static tension-test specimens were pieces of rod about 18 in. long, unmachined, and held in wedge grips. The impact tension specimens were of the form and size shown in Fig. 1, and for the welded specimens the weld was at the middle of length of the specimen. The notched-bar specimens were of the form and size shown in Fig. 2. The fatigue specimens were of the form and size shown in Fig. 3.

The test data for the fatigue tests are shown graphically in Fig. 4, which gives the *S-N* (or stress-cycle) diagrams for the fatigue tests plotted on semi-logarithmic paper. A glance at these *S-N* diagrams shows how the endurance limit was determined as the ordinate of the horizontal part of the diagram.

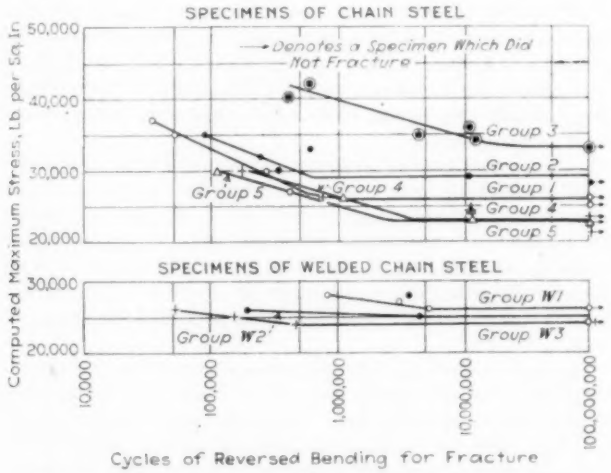


FIG. 4 *S-N* (STRESS-CYCLE) DIAGRAMS FOR FATIGUE TESTS

It does not seem unreasonable to assume that the resistance of a material to reversals of stress is a fair index of its resistance to repetitions of stress in one direction.

Again the impact tension tests, using unnotched tension specimens, are assumed to give a fair index of the resistance of the material to the energy of an impact load, such as would be caused by the slipping of hitches and jammed links. The results of the Charpy notched-bar tests are very difficult to interpret, but probably indicate something of the susceptibility of the metal to damage by notches, nicks, and cuts. It is to be noted that the results of the tension tests and of the fatigue tests are measured in pounds per square inch (stress), while the results of the impact tests are measured in foot-pounds (energy).

For the unwelded rod the effect of overstress without subsequent annealing is to increase both the yield point and the tensile strength, the increase of yield point being much more marked than the increase of tensile strength. Severe overstress increased both these values distinctly more than did stress just beyond the yield point. Overstress without subsequent annealing reduced the resistance to shock, as shown by the impact-tension test re-

sults. The endurance limit was increased by overstress in about the same proportion as the static tensile strength. Overstress without subsequent annealing slightly reduced the Charpy notched-bar values.

For the specimens overstressed and subsequently annealed the yield point, tensile strength, and endurance limit were found at somewhat lower values than for the normalized material. For the specimens subjected to slight overstress (just beyond the yield point) the shock-resisting value (as measured by the results of the tensile impact tests) was restored by annealing to that of the normalized material before overstress. For the heavily overstressed metal (stress nearly up to the tensile strength) the restoration of shock-resisting value was only partial.

Owing to the number of welded specimens available it was, unfortunately, not possible to get a full set of test results. Under a stress which severely overstressed the base metal, but probably did not seriously overstress the weld, the endurance limit was but very slightly affected. It is worthy of note that the endurance limit of the welded metal was about the same as that of the normalized rod. The resistance to shock, as measured by the results of the tensile impact tests, was not greatly affected for the specimens stressed to 45,700 lb. per sq. in. without subsequent annealing. The resistance to shock of welded specimens was markedly increased by annealing after the application of stress.

Conclusions

1 The tests here reported were made to study the effect of occasional overload and of subsequent annealing on chain steel and welded chain steel. They throw no light on the effect of many repetitions of working load in producing damage at points of contact of link.

2 The results of the tests here reported indicate that occasional overload tends to increase the static and fatigue strength of unwelded chain steel at the expense of the ability to resist the energy of shocks.

3 The results of the tests here reported indicate that the effect of annealing after overstress is to reduce the static and fatigue strength of the unwelded metal to values as low or lower than those of the metal before overstress, and that the benefit of annealing consists in the complete or partial restoration of the shock-resisting capacity of the material. This increase of shock-resisting capacity was especially noticeable in the annealed welded specimens.

APM-52-12

The State of Stress in Full Heads of Pressure Vessels

By W. M. COATES,[1] ANN ARBOR, MICH.

After a brief survey of the significant experiments upon which a comparison with the theory developed is to be based and a statement of the problem, the author derives expressions for stresses and displacements at any point of a thin-walled pressure vessel. Assuming that the shell offers no resistance to bending, an analysis is made considering it as a membrane. A study of the resulting equations indicates that in the neighborhood of the junction of head and cylinder, bending couples result, so that the plates may not be considered as membranes, at least in this locality. This leads to a consideration of the local effects of the reactions set up at the junction, the head and cylinder being considered as surfaces capable of carrying bending stresses with loads which are uniformly distributed edge shearing forces and bending couples. The resultant effect at any point is obtained by superposition. An illustrative example is given.

FROM the accounts of meetings at which pressure vessels with full heads have been discussed, the emphasis put upon the empirical and the older experimental methods would seem to suggest that the existence of a theory which will predict these findings is not sufficiently known. This is possibly due to the form in which the theory has appeared, and it is hoped here to make some contribution which will better the situation in that respect. Inasmuch as improvements in experimental methods permit a triple check, one set of measurements with another and both with the theory, the question has a new appeal. A brief survey of the most significant experiments will bring out the facts with which it is expected to work.

Vessels which carry inner pressure have, as is well known, a zone of high stress concentration at the inner surface of the knuckle, both in the flat crimped head and the dished head whos meridian is built up of two circular arcs. This fact was established by measurements of displacements at the outer surface, in the direction of the axis and perpendicular thereto.[2] From the picture thus obtained of the deformed shape of the head, conclusions are possible concerning the state of stress in it.

Later experiments showed[3] the following:

The full dished head, when subjected to increasing inner pressure, approaches a shape which has elliptical meridians.

A series of heads which have the same radius of curvature at the crown and the same diameter of base, is arranged to have meridians built up of two circular arcs, with the knuckle radius of curvature larger for each successive head. Then the pressure at which the first significant permanent deformation occurs will also be larger for each successive head.[4]

If the head be ellipsoidal, such that for its elliptical meridian

$$\frac{\text{semi-major axis}}{\text{semi-minor axis}} = \frac{\text{diameter of cylinder}}{\text{depth of head}} = \frac{2}{1}$$

then it will resist a pressure sufficient to cause the first slight permanent deformation to be in the cylinder.

Experiments made by the Swiss Association of Boiler Owners[5] verified the above findings on full-size vessels and noted especially the greater ability of the ellipsoidal head (axis ratio 2) to resist permanent deformation even at high pressures. In this group of experiments not only was the deformed shape of the head found, but also extensometers were introduced to procure the strain distribution along meridians of the outer surface of the vessel. This was found to be highly irregular in all cases,[6] but markedly more favorable for the ellipsoidal head above, and for heads whose meridians had two radii of curvature with a sufficiently large knuckle radius.

A series of experiments at the Testing Materials Laboratory in Stuttgart[7] showed again the superiority of the ellipsoidal shape (axis ratio about 2) over those with smaller knuckle radii. Here the elastic and permanent deformations of the heads were found, and the pressures at which the yield point was reached.

All of the previous investigations were confined to phenomena at the outer surface of the vessel. With no possibility of applying instruments at the inner surface of the knuckle where the maximum stresses were to be expected, recourse was had to obtaining the angular displacement of normals to the surface. These measurements, combined with those of elongations at the outer surface, permitted establishing the strain due to bending, and this permitted the computation of strains, hence also stresses, at both outer and inner surfaces.[8,9]

The study of yield lines on the surface was also extended

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[1] Instructor in Engineering Mathematics, University of Michigan.

[2] C. Bach, "Untersuchung über die Formänderungen und die Anstrengung flacher umgekrempter Kesselböden," *Z.V.D.I.*, 1897, p. 1157. Also "Anstrengung gewölbter Böden," *Z.V.D.I.*, 1899, p. 1585.

[3] C. Diegel, "Versuche über die Beanspruchung des Materials geschweisster zylindrischer Kessel mit nach aussen gewölbten Böden." *Forschungsarbeiten auf dem Gebiete des Ingenieurwesens*, Sonderreihe M, Heft 2. The work was done in the period from 1905 to 1909 and from 1913 to 1914 in an industrial concern, and the ellipsoidal head was patented. The above publication was made in 1920.

Contributed by the Applied Mechanics Division and presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] The series was so arranged as to approach from below the ellipsoidal shape with axis ratio 2/1. Tests were on models of about 10 in. diameter.

[5] E. Höhn, "Ueber die Festigkeit elektrisch geschweisster Hohlkörper." Springer, Berlin, 1924. See short account in *Mechanical Engineering*, vol. 49, no. 2, Feb., 1927; also *Engineering* (London), July 5, 1929.

[6] Some of the diagrams are given in *Mechanical Engineering*, loc. cit. See also Barraclough and Gibson's "Boiler Explosions as Affected by Unsymmetrical Riveted Joints," Proc. Eng. Assn. N.S.W., vol. 26 (1910–11), p. 271.

[7] C. Bach, "Versuche über die Widerstandsfähigkeit und die Formänderung gewölbter Kesselböden." *Forschungsarbeiten, V.D.I.*, Heft 270, 1925. See abstract in *Mechanical Engineering*, loc. cit. The original contains some excellent photographs of yield lines on the surface.

[8] Mitteilungen aus dem Kaiser-Wilhelm-Institut für Eisenforschung zu Düsseldorf, Abhandlungen 59, 60. Siebel und Körber, "Versuche über die Anstrengung und die Formänderungen gewölbter Kesselböden u.s.w.," I und II, Bericht, 1926.

[9] Höhn, "Über die Festigkeit der gewölbten Böden und der Zylinderschale," Springer, Berlin, 1927.

to include yield phenomena within the material, by means of etched sections.[10]

Thus we find a development of three experimental methods:

1 The investigation of the state of stress in a pressure vessel before the elastic limit of the material is passed, by means of fine extensometers and instruments for procuring angular deflections. (The use of dials for measuring axial or normal displacements is also possible here, but is of more interest in the next group, due to the small magnitudes of these displacements.)

2 The investigation of elastic and plastic deformation of the vessel, principally of the head, along its normals and in the direction of or perpendicular to the axis.

3 The investigation of yield phenomena, on the surface or within the material. This may be combined with tests of specimens from the destroyed vessel.

The first of these methods especially recommends itself for industrial purposes, since it may be carried out without injuring the vessel, and, with slight provision, may be made part of the routine test of a new vessel. Also it is the only one of the three which gives the quantities of fundamental interest, namely, the stresses which are set up.

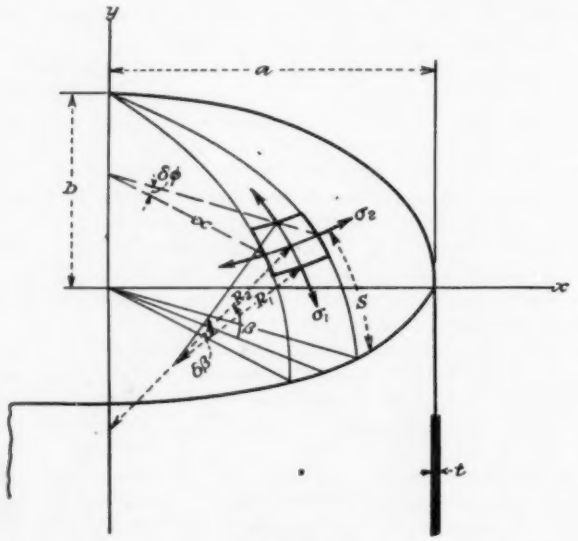


FIG. 1

The logical outcome of these experiments in the form of empirical methods has not been missing. These will not be discussed here since the interest is in how far the findings can be predicted without such considerations. Along with experiment, in part independently of it, went an improvement in the theoretical attack; and this was used to good advantage in at least one set of experiments.[11] The following is an attempt to give a simplified presentation and method of application of this theory. As long as the head may be considered thin and its shape is near that shown to be desirable by experiment, it is possible to compute all the quantities which can be measured.

STATEMENT OF THE PROBLEM

The pressure vessel is assumed to be as follows: Its side walls are in the form of a right circular cylinder whose length exceeds its diameter. Each head is a half-ellipsoid of revolution of base radius a and depth b; or it may also be a surface of revolution whose meridian is an approximation to an ellipse by two or more circular arcs, with that at the knuckle tangent to the element of the cylinder where they join. This last shape has several technical designations. It will be called here "quasi-ellipsoidal." The thickness t is taken to be constant throughout the vessel. The load is a uniformly distributed inner pressure, of intensity p. This will therefore give a normal stress perpendicular to the surface, of amount 0 at the outer surface and p at the inner. Since p is small in comparison with the meridional and hoop stresses developed, it will be assumed that a two-dimensional state of stress exists in which the material is not stressed beyond the elastic limit. Then it is desired to find expressions for the stresses and the displacements at any point of the vessel.

THE MEMBRANE EQUATIONS

The simplest assumption is that the shell[12] offers no resistance to bending. It is then called a "membrane." For a surface of revolution under uniform inner pressure, the membrane stresses take the well-known form

$$\left.\begin{aligned} \sigma_1 &= \frac{pR_2}{2t} \\ \sigma_2 &= \sigma_1\left[2 - \frac{R_2}{R_1}\right] \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

in which σ_1 = meridional stress, uniformly distributed over the thickness of the shell

σ_2 = hoop stress

R_1 = radius of curvature of the meridional section

R_2 = radius of curvature of the section perpendicular to the meridian at the point in question (see Fig. 1).

Use will be had for the following special cases:

(*a*) The Hemisphere: $R_1 = R_2 = a$

$$\left.\begin{aligned} \sigma_1 &= \frac{pa}{2t} \\ \sigma_2 &= \sigma_1 \end{aligned}\right\} \quad \ldots\ldots [1a]$$

(*b*) The Cylinder: $R_1 \longrightarrow \infty$, $R_2 = a$

$$\left.\begin{aligned} \sigma_1 &= \frac{pa}{2t} \\ \sigma_2 &= 2\sigma_1 \end{aligned}\right\} \quad \ldots\ldots [1b]$$

(*c*) The Ellipsoid: $R_1 = kav^3$, $R_2 = kav$

$$\left.\begin{aligned} \sigma_1 &= \frac{pka}{2t}\, v \\ \sigma_2 &= \sigma_1\left[2 - \frac{1}{v^2}\right] \end{aligned}\right\} \quad \ldots\ldots [1c]$$

$$\text{where } k = \frac{a}{b} \text{ and } v = \frac{1}{\sqrt{1 + (k^2 - 1)\cos^2\beta}}$$

The displacement of any point P of the shell, due to the applied load, is resolved into a component u along the tangent to the meridian, and a component w along the normal to the surface. The zero value of the *meridional displacement* u is taken at the plane in which head and cylinder join, and for each surface u is positive away from this plane. This will cause no confusion since the two surfaces are discussed separately. The zero value of the *normal displacement* w is taken at the unstrained middle

[10] Mitt. a.d. K.W.I. für Eisenforschung zu Düsseldorf, Abhandlung 62. Siebel und Pomp: "Das Zusammenhang zwischen der Spannungsverteilung und der Fliesslinienbildung an Kesselböden u.s.w.," 1926.

[11] See especially Part II of the pamphlet mentioned in footnote 9.

[12] The term "shell" is used to define a curved plate of thickness small in comparison with its other dimensions. Here it may be either the head or the cylinder. A certain technical usage would limit it to the cylinder.

surface, and w is positive outward. These components for a membrane under uniform inner pressure become:

(*a*) For the hemisphere:

$$\left.\begin{aligned} u &= 0 \\ w &= \frac{pa^2}{2Et}(1-\mu) \end{aligned}\right\} \quad \ldots\ldots\ldots [2a]$$

(*b*) For the cylinder:

$$\left.\begin{aligned} u &= \frac{pas}{2Et}(1-2\mu) \\ w &= \frac{pa^2}{2Et}(2-\mu) \end{aligned}\right\} \quad \ldots\ldots\ldots [2b]$$

(*c*) For the ellipsoid:

$$\left.\begin{aligned} u &= \frac{pa^2}{2Et}\frac{\sqrt{k^2-1}}{2k}\mathbf{L}\cos\beta \\ w &= \frac{pa^2}{2Et}\left\{\frac{\sqrt{k^2-1}}{2k}\mathbf{L}\sin\beta + k^2[(2-\mu)v^2-1]\right\} \end{aligned}\right\} \quad \ldots\ldots [2c]$$

in which μ = ratio of unit lateral strain to unit longitudinal strain and is here taken as 0.3

s = distance measured along the meridian from the junction plane to the point in question and is positive away from the junction; and

$$\mathbf{L} = \left\{[2k^2-(1-2\mu)]\log(k+\sqrt{k^2-1}\sin\beta)v - [(1-2\mu)k\sqrt{k^2-1}\,]v^2\sin\beta\right\}$$

in which the square-bracketed expressions are constant for any given case.

By use of these expressions for the displacements, we may now examine the pressure vessel with regard to its behavior as a membrane.

For a vessel with hemispherical heads, and p, a, E, t, and μ constant, the normal expansion w of the cylinder (Equation [2*b*]) exceeds that of the head (Equation [2*a*]) in the ratio $(2-\mu)/(1-\mu) = 2.4$. The assumption that these shells behave as membranes leads to the state shown by heavy lines in Fig. 2. The light lines represent the unstrained position. In the actual vessel, the occurrence of such a discontinuity is prevented by the ability of each shell to resist forces normal to its middle surface or bending couples acting in meridional planes. The head and cylinder experience reactions from each other as shown in Fig. 4, and consequently, at least in the neighborhood of the junction, they may not be considered purely as membranes.

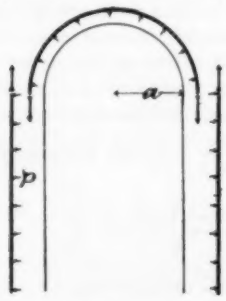


Fig. 2

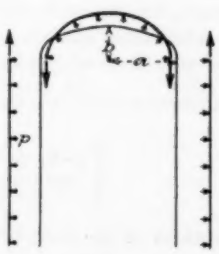


Fig. 3

In case the vessel has an ellipsoidal head, a still greater discontinuity results from the assumption that the shells behave as membranes. From Equation [2*c*], at the edge, $\beta = 0$, and $v = 1/k = b/a$. Then

$$\mathbf{L} = (\log 1) - 0 = 0 \quad \text{and} \quad \left\{\begin{aligned} u &= 0 \\ w &= \frac{pa^2}{2Et}[(2-\mu)-k^2] \end{aligned}\right.$$

For the hemisphere, $k = 1$ and $w = \frac{pa^2}{2Et}(1-\mu) = \frac{pa^2}{2Et}(0.7)$ in agreement with Equation [2*a*]. With a constant base radius a, as the depth b decreases, k increases. When $2-\mu = k^2$, that is, when $k = 1.3$, the normal displacement w at the edge is zero. For $k > 1.3$, w is negative, hence the edge of the ellipsoid contracts instead of expands. In the usual design $k > 2$, consequently the condition of contraction shown in Fig. 3 is expected in such vessels on the basis of their behavior as membranes. It is recalled that the cylindrical side wall always expands (Equation [2*b*]), therefore the edge reactions of the sort shown in Fig. 4 increase in importance as the depth of the head is reduced from that for a hemisphere.

Similar results may be obtained in the case of a quasi-ellipsoidal head. A discontinuity in the normal membrane displacement w arises there not only at the junction of head and cylinder, but also at each junction of zones of different ring surfaces.

Analysis of the General Problem

It has been shown that a discontinuity exists in the values of the normal displacement w at the junction of head and cylinder. The equations upon which the discussion was based, [2*a*, *b*, *c*], were obtained on the assumption that the shells behave as membranes. For thin shells under uniform inner pressure with no restraints at the edges, the bending stresses which are set up during these displacements are negligible in comparison with the normal stresses. Then the predicted discontinuity would exist in the actual vessel if it were not prevented by another set of forces, namely, the reactions shown in Fig. 4. From their nature, they cannot be carried by a membrane. They will set up in the shell normal, shearing, and bending stresses, which are of local character for shells of a certain design. The consideration of these effects forms a second part of the analysis of the state of stress in the vessel.

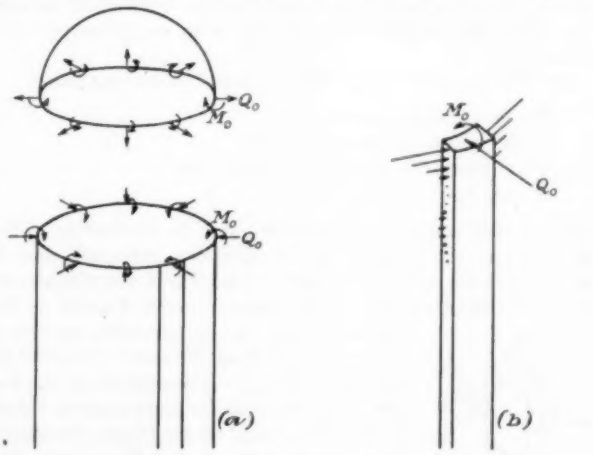


Fig. 4

It is therefore found convenient to make the following division:

1 *Membrane Analysis* of the shells, under uniform inner pressure. (See Fig. 2 or 3.)

2 *Analysis of Local Effects.* The shells are considered as surfaces capable of carrying bending stresses, and the loads are taken to be the uniformly distributed edge shearing forces and bending couples shown in Fig. 4. Then the resultant effect at any point is obtained by superposition. The equations necessary for the first part of the analysis have been given, and it remains to develop those for the second part. In discussing the shell as a membrane, it was found possible to deter-

mine the stresses directly, since the problem is a statically determinate one. From the stresses, the strains, and thence the displacements, were found. In the case of a shell which supports bending stresses, this method cannot be used since the statical conditions of equilibrium give only three equations in five unknowns, viz., the quantities T_1, T_2, Q, M_1, and M_2, shown in Fig. 6. Consequently, as is usually done, the elastic properties of the material are introduced. These involve the displacements, which will be most conveniently determined first. The solution for the stresses will then follow.

The Cylindrical Shell With Edge Loads Only

The shell is as shown in the lower part of Fig. 4. The intensities of the loads, which are uniformly distributed along the edge circle, are Q_0 and M_0, respectively, each per area $t \times$ unit arc. The cylindrical shell of a vessel is usually sufficiently long that the effects of these loads at the end do not extend into the neighborhood of the other end; therefore it may as well be assumed that the length is infinite. The strip shown in Fig. 4 is isolated as a free body in Fig. 4(*b*). The forces acting on this body are the loads Q_0 and M_0 at the end, and the forces and bending couples which are set up on the longitudinal sections. The latter are omitted from the figure to avoid complications there.

An element of the strip (Fig. 5) is in equilibrium under:

T_2, the normal hoop force;
Q, the meridional shearing force;
M_1, the meridional bending couple; and
M_2, the hoop bending couple.

Other forces and couples drop out, due to symmetry. Those given above are positive in the direction indicated, and the magnitudes are for an area unity $\times$ t. The force which resists the radial displacement of the element varies directly as that displacement; that is, the strip receives an elastic support from the rest of the shell (see Appendix No. 1). The analysis of the local effects in the cylinder consequently reduces to that of a beam on an elastic foundation, loaded by a force and a bending couple at one end.[13]

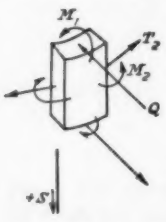


Fig. 5

The normal displacement of any element at distance s from the edge is then given by the equation:

$$\frac{d^4w}{ds^4} + 4\lambda_0^4 w = 0 \quad \ldots\ldots [3]$$

in which

$$\lambda_0 = \sqrt[4]{\frac{3(1-\mu^2)}{a^2t^2}} \quad \ldots\ldots [4]$$

For the assumed values of the edge loads, this equation has the solution:

$$w = -\frac{1}{2\lambda_0^3 D} e^{-\lambda_0 s}[Q_0 \cos \lambda_0 s + \lambda_0 M_0(\cos \lambda_0 s - \sin \lambda_0 s)] \quad \ldots [5]$$

in which

$$D = \frac{E}{1-\mu^2} I = \frac{E}{1-\mu^2} \frac{1 \times t^3}{12} = \frac{Et^3}{12(1-\mu^2)} \quad \ldots\ldots [6]$$

is the flexural rigidity of the shell and has the role here of EI in the case of a simple beam.

From the above value of w are obtained the expressions for the normal and shearing forces and the bending couples on any element of the shell.

[13] See Timoshenko and Lessells, "Applied Elasticity," p. 143.

$$\left.\begin{aligned}
T_2 &= \frac{Et}{a} w = -2a\lambda_0^2 e^{-\lambda_0 s}[Q_0 \cos \lambda_0 s \\
&\qquad + \lambda_0 M_0(\cos \lambda_0 s - \sin \lambda_0 s)] \\
Q &= D\frac{d^3w}{ds^3} = -e^{-\lambda_0 s}[Q_0(\cos \lambda_0 s - \sin \lambda_0 s) \\
&\qquad - 2\lambda_0 M_0 \sin \lambda_0 s] \\
M_1 &= -D\frac{d^2w}{ds^2} = \frac{1}{\lambda_0} e^{-\lambda_0 s}[Q_0 \sin \lambda_0 s \\
&\qquad + \lambda_0 M_0(\cos \lambda_0 s + \sin \lambda_0 s)] \\
M_2 &= \mu M_1
\end{aligned}\right\} \quad \ldots\ldots [7]$$

This solution has the advantage that the variables enter only in the forms:

$$e^{-\lambda_0 s} \cos \lambda_0 s, \quad e^{-\lambda_0 s} \sin \lambda_0 s, \quad e^{-\lambda_0 s}(\cos \lambda_0 s \pm \sin \lambda_0 s)$$

These functions have all been tabulated,[14] and are displayed in Fig. 11. Each is oscillatory, and its amplitude is very rapidly damped out by the factor $e^{-\lambda_0 s}$. The zero values

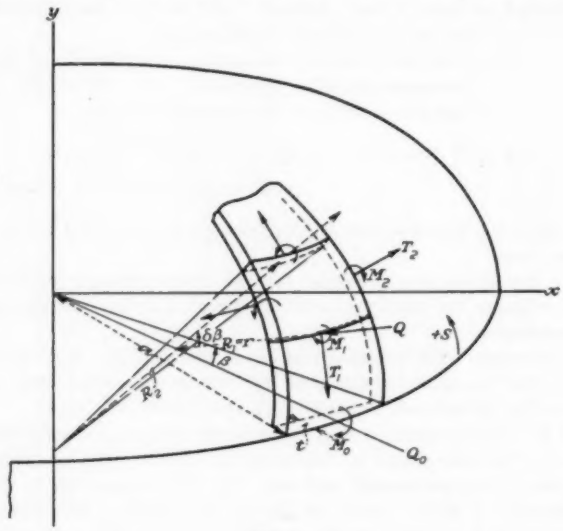


Fig. 6

of each occur at an interval of π in the values of $\lambda_0 s$. The same interval separates the zero values of the derivative of each function, hence the critical ordinates occur at such intervals. Also, it may be noted that zeros and critical ordinates for any of these functions occur with a phase difference of $\pi/4$. For example, the function $e^{-\lambda_0 s} \cos \lambda_0 s$ has a zero value at $\lambda_0 s = \pi/2$, a minimum at $\pi/2 + \pi/4$, another zero at $\pi/2 + \pi$, a maximum at $\pi/2 + \pi + \pi/4$, etc. Successive critical ordinates of each function are in the ratio $e^{-\pi} = \frac{1}{23.1} = 0.043$. For example,

$$\left[\frac{e^{-(\lambda_0 s + \pi)} \cos(\lambda_0 s + \pi)}{e^{-\lambda_0 s} \cos \lambda_0 s}\right] = e^{-\pi}$$

The width of the zone between critical ordinates is

$$s = \frac{\pi}{\lambda_0} = \frac{\pi\sqrt{at}}{1.285}$$

or $2.44\sqrt{at}$. This is also the distance between successive zeros. As the thickness t is diminished for the same radius

[14] Such functions are characteristic of the problem of a beam on an elastic foundation, and their importance in this connection, e.g., for computing the stresses in a rail, led to tabulation. See Zimmermann, "Die Berechnung des Eisenbahn-Oberbaus," Berlin, 1888. Also Timoshenko and Lessells, "Applied Elasticity," p. 138.

a, this distance is also diminished; consequently the thinner the shell, the narrower the zone of influence of the forces and moments set up by the loads Q_0 and M_0. Equations [7] give the final results for these if Q_0 and M_0 are known; however, when the cylinder is joined to a head to form a pressure vessel, these edge loads have to be determined (see Appendix No. 1). This determination must be preceded by the analysis of the head.

The magnitudes of the stresses may be calculated from Equations [7]. To avoid a partial repetition of tabulation, the final statement of the stresses will be postponed until Q_0 and M_0 have been found.

The Head With Edge Loads Only

The head is to have the following character: Its middle surface is generated by revolving about the y-axis (Fig. 6) a meridian curve which approximates an ellipse, of longer semi-axis a and knuckle radius $R_1 = r$ constant for a certain zone, and not too small a part of a. The thickness t is small in comparison with a. The edge is in the equatorial plane, which contains also the centers of curvature of the knuckle surface. If the head has openings such as manholes, etc., they are to be sufficiently remote that the region of their local effects shall not extend into that in which we are interested near the junction of head and cylinder.[10]

The strip shown in Fig. 6 is isolated as a free body. It is in equilibrium under the action of the applied loads Q_0 and M_0 at the edge, and the restraining forces and couples which they will set up in the strip. These are shown acting on an element of the strip in Fig. 6.

The following notation, in part already presented, will be used:

T_1 = normal meridional force
T_2 = normal hoop force
Q = shearing force in the cross-section perpendicular to meridian, and in the direction perpendicular to the middle surface
M_1 = meridional bending moment
M_2 = hoop bending moment.

These are positive in the directions indicated in Fig. 6, and the magnitudes are for an area, in each case, unity $\times$ t.

The stresses due to the normal and shearing forces are assumed to be uniformly distributed over the thickness of the shell.

The stresses due to the bending couples are assumed to vary linearly over the thickness of the shell, with zero value at the middle surface and numerical maxima at the outer and inner surfaces.

$$\left.\begin{aligned} \sigma_1 &= \frac{T_1}{t} = \text{meridional normal stress} \\ \sigma_2 &= \frac{T_2}{t} = \text{hoop normal stress} \\ \tau &= \frac{Q}{t} = \text{meridional shearing stress} \end{aligned}\right\} \quad \ldots\ldots\ldots [8]$$

$$\left.\begin{aligned} \sigma_{1e} &= \frac{M_1\left(\frac{t}{2}\right)}{\frac{1 \times t^3}{12}} = \frac{6M_1}{t^2} = \text{maximum meridional stress due to bending; } e \text{ subscript for external surface, } i \text{ for internal} \\ \sigma_{2e} &= \frac{6M_2}{t^2} = \text{maximum hoop stress due to bending.} \end{aligned}\right\} \quad \ldots [9]$$

s = length of arc measured along meridian, positive away from edge
s_2 = length of arc on hoop circle (rarely used)
r = meridional radius of curvature at the edge.

The solution obtained for the cylinder is exact for thin shells. In the case of the head, the inclusion of the expressions for certain forces which are known to be slight has undesirable mathematical consequences. By omitting these terms we come to an approximate solution, which should be, however, physically adequate.

The following assumption is made as a basis:

For a value of "s" small as compared with "a," and for a sufficiently large value of "r/a," the displacement of each point of the strip is essentially normal to the middle surface; its component in the direction of the meridian is negligible.

In the zone in which this assumption holds, the meridional component of Q_0 is negligible, hence, even assuming it is all transmitted along the meridian, T_1 would be negligible. The stress σ_1 is therefore taken to be zero.

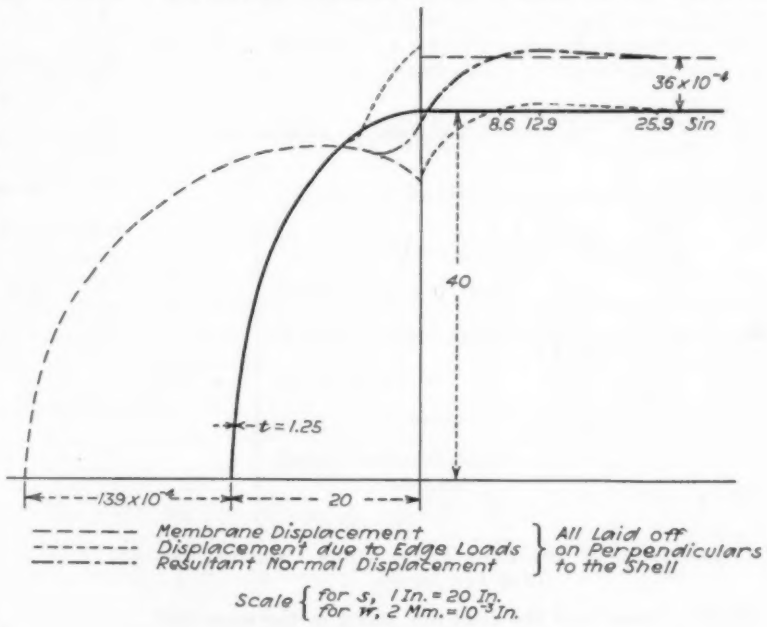


FIG. 7 THE NORMAL DISPLACEMENT w

In this case the force which resists the radial displacement of the element varies directly as that displacement and inversely as the square of the second radius of curvature of the shell.

Since R_2 increases with β, the factor $1/R_2^2$ decreases. Continuing the analogy used in the case of the cylinder, it may be said here that the strip behaves as a beam on an elastic foundation with a decreasing "modulus of foundation" (see Appendix No. 1) the loads being a shearing force and a bending couple at the end.

The normal displacement of an element at distance s from the edge is given by:

$$\frac{d^4w}{ds^4} + 4\lambda^4 w = 0 \quad \ldots\ldots\ldots [10]$$

in which

$$\lambda = \sqrt[4]{\frac{3(1-\mu^2)}{R_2^2 t^2}} \quad \ldots\ldots\ldots [11]$$

It is to be noted that λ here involves R_2, which is variable for most surfaces. Equation [3] is a special case of Equation

[10] for the cylinder, in which $R_2 = a$ = constant. Another case in which R_2 is constant is that of a hemispherical head. Then the integration of Equation [10] takes the same form as that of Equation [3], except for the determination of the constants of integration.

In case R_2 is variable, as for all types of ellipsoidal or quasi-ellipsoidal heads, the equation may be integrated numerically. Starting with $s = 0$ a meridional arc is taken, so short that by regarding R_2 as constant for this arc, we make no appreciable error in the computation (see cols. 4 and 7, Table 2). Then we have the same case for the first zone as for a hemisphere, and also as for the cylinder, with the following differences:

The sign of Q_0 is changed since the edge of the head is pulled outward radially.

λ varies as $1/\sqrt{R_2}$ and is equal to λ_0 only at the edge. Its value is approximately the edge constant for the first zone, but will be different for the next zones. To keep this clear, we distinguish the edge constant from the variable by the subscript. The derivatives of λ are taken to be negligible.

The solution to Equation [10] for a hemisphere, or for the first zone of an ellipsoid or quasi-ellipsoid, is

$$w = \frac{1}{2\lambda_0^3 D} e^{-\lambda s} [Q_0 \cos \lambda s - \lambda_0 M_0 (\cos \lambda s - \sin \lambda s)] \quad \ldots\ldots [12]$$

The forces and the bending couples are then

$$\left.\begin{aligned} T_2 &= \frac{Et}{R_2} w = 2\lambda_0 a \frac{a}{R_2} e^{-\lambda s} [Q_0 \cos \lambda s \\ &\qquad - \lambda_0 M_0 (\cos \lambda s - \sin \lambda s)] \\ Q &= D \frac{d^3 w}{ds^3} = \frac{a}{R_2} \sqrt{\frac{a}{R_2}} e^{-\lambda s} [Q_0 (\cos \lambda s - \sin \lambda s) \\ &\qquad + 2\lambda_0 M_0 \sin \lambda s] \\ M_1 &= D \frac{d^2 w}{ds^2} = \frac{a}{R_2} \frac{1}{\lambda_0} e^{-\lambda s} [-Q_0 \sin \lambda s \\ &\qquad + \lambda_0 M_0 (\cos \lambda s + \sin \lambda s)] \\ M_2 &= \mu M_1 - D (1 - \mu^2) \frac{\tan \beta}{R_2} \frac{dw}{ds} \end{aligned}\right\} \quad \ldots\ldots [13]$$

Determination of the Edge Loads When Cylinder and Head Are Joined

If the cylinder and the head are joined in the equatorial hoop of the latter to form a vessel which is then submitted to a uniformly distributed inner pressure, the following conditions must be fulfilled:

1 The total radial displacement, which is the sum of the radial displacement of the membrane and that due to the edge reactions, must be numerically the same for the two edges.

2 The tangent to the total-radial-displacement curve for the head and the tangent to the similar curve for the cylinder are common at the junction.

In the case at hand the shells have the same thickness t, and the first condition gives (Appendix No. 1) the value of the edge shearing force

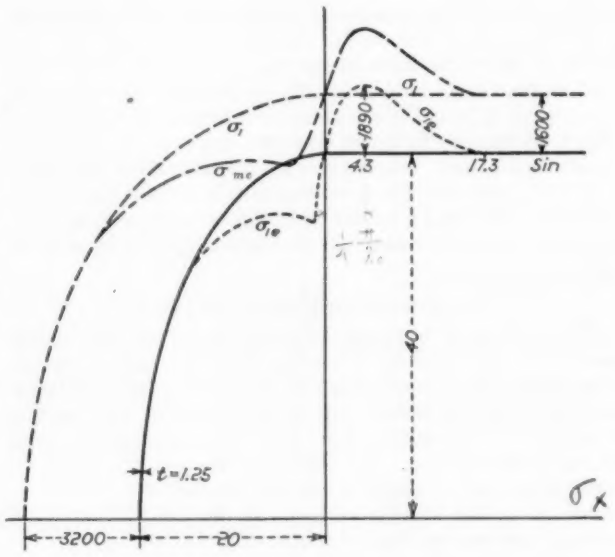
$$Q_0 = \frac{pk^2}{8\lambda_0} \quad \ldots\ldots [14]$$

The second condition gives the value of the edge bending moment

$$M_0 = 0 \quad \ldots\ldots [15]$$

Equations for the Local Effects Near the Junction of Head and Cylinder

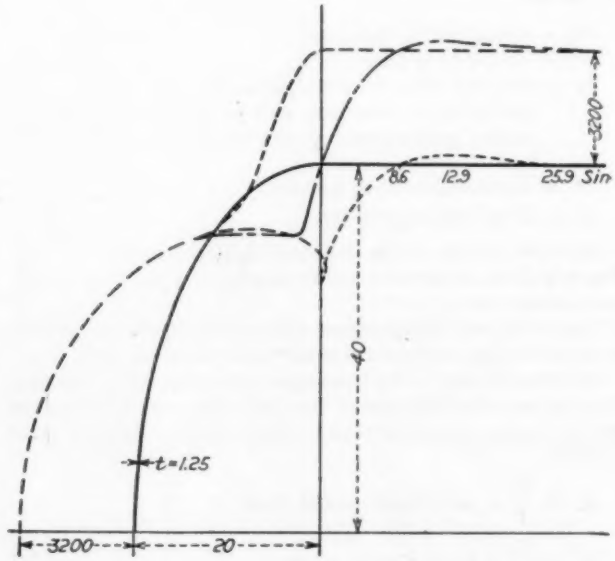
The values of the forces and couples set up by the mutual reactions at the junction of head and cylinder are found by substituting the values of Q_0 and M_0, given by Equations [14] and [15], in Equations [5] and [7], and [12] and [13]. The corresponding stresses are then obtained by use of Equations [8].



Membrane Stress
Discontinuity Stress (bending)
Resultant Meridional Stress
} All Laid off on Perpendiculars to the Shell

Scale { For s 1 In.=20 In.
For σ 1 Mm.=200 Lb. per Sq. In.

Fig. 8 The Meridional Stresses σ_1 and σ_{1e}



Membrane Stress
Discontinuity Stress
Resultant Hoop Stress
} All Laid off on Perpendiculars to the Shell

Scale { For s 1 In.=20 In.
For σ 1 Mm.=200 Lb. per Sq. In.

Fig. 9 The Hoop Stresses σ_2

$$
\left.\begin{aligned}
w &= -\frac{pk^2a^2}{4E\,t}\,e^{-\lambda_0 s}\cos\lambda_0 s;\quad \sigma_{1e} = \frac{3pk^2}{4\lambda_0^2t^2}\,e^{-\lambda_0 s}\sin\lambda_0 s\\
\sigma_2 &= -\frac{pk^2a}{4t}\,e^{-\lambda_0 s}\cos\lambda_0 s;\quad \sigma_{2e} = \mu\sigma_{1e}\ *\\
\tau &= -\frac{pk^2}{8\lambda_0 t}\,e^{-\lambda_0 s}(\cos\lambda_0 s - \sin\lambda_0 s);\\
&\qquad\qquad \sigma_{1i} = -\sigma_{1e}\\
&\qquad\qquad \sigma_{2i} = -\sigma_{2e}
\end{aligned}\right\}\quad ..[16]
$$

For the cylinder, these expressions hold for any point at distance s from the junction.

For the ellipsoid, or the quasi-ellipsoid, all signs are reversed and the expressions hold only for an edge zone whose radius of curvature R_2 may be regarded as constant within the accuracy desired.

The variation of these stresses is that of the functions $e^{-\lambda_0 s}\times\cos\lambda_0 s$, etc. discussed above and displayed in Fig. 11. The ordinates there have only to be multiplied by the constant factors which enter Equations [16]. The most important of these stresses are σ_2 and σ_{1e} (see illustrative example, local effects in cylinder), and their magnitudes are such as to make the use of the membrane formulas alone quite meaningless for the computation of stresses near the junction. The stresses σ_{2e} and τ are comparatively small. All of these local stresses are so rapidly damped out that their values outside an edge zone of width $s = 2.44\sqrt{at}$ are of no concern.

In superposing the membrane stresses and the local stresses it is to be noted that the bending stresses have opposite signs for the outer and inner surfaces, whereas each direct stress has the same sign throughout the thickness. Consequently, if we are concerned with numerically maximum values of the resultant stress, we shall consider both surfaces.

For the cylinder, the numerically greatest value of σ_2 is $\frac{pk^2a}{4t}$ and it occurs at the edge; the first maximum of σ_{1e} is $\frac{3pk^2}{4\lambda_0^2t^2}$ (0.32) and it occurs at $\lambda_0 s = \pi/4$, that is, $s = 0.61\sqrt{at}$. Since the values of $e^{-\lambda_0 s}\cos\lambda_0 s$ and $e^{-\lambda_0 s}\sin\lambda_0 s$ are equal for $\lambda_0 s = \pi/4$, the value of σ_2 when σ_{1e} is maximum is numerically $\frac{pk^2a}{4t}$ (0.32).

In dealing with the head it may happen that the edge zone within which R_2 may be regarded as constant is too small a part of the edge zone within which the local stresses are damped out. In that case, starting from the edge we take a sufficient number of zones, within each of which R_2 may be regarded as constant, to cover the region in which the local stresses are significant. After the edge zone, Equations [16] no longer hold for the head. They are replaced by (see footnote 20, p. 127):

$$
\left.\begin{aligned}
w &= \frac{pk^2a^2}{4Et}\left(e^{-\int_0^s\lambda ds}\right)\cos\left(\int_0^s\lambda ds\right)\\
\sigma_2 &= \frac{pk^2a^2}{4tR_2}\left(e^{-\int_0^s\lambda ds}\right)\cos\left(\int_0^s\lambda ds\right)\\
\tau &= \frac{pk^2a^2\sqrt{2}}{8\lambda tR_2^2}\left(e^{-\int_0^s\lambda ds}\right)\cos\left(\frac{\pi}{4}+\int_0^s\lambda ds\right)\\
\sigma_{1e} &= \frac{-3pk^2a^2}{4\sqrt{3\,(1-\mu^2)}\,tR_2}\left(e^{-\int_0^s\lambda ds}\right)\sin\left(\int_0^s\lambda ds\right)\\
\sigma_{2e} &= \mu\sigma_{1e}
\end{aligned}\right\}\quad ..[17]
$$

The values of the expressions $\int_0^s\lambda ds$ may be conveniently obtained by numerical integration, as illustrated in the following paragraphs.

Membrane Analysis

Note: Figures in brackets refer to equations in paper.

Cylinder

$$
\left.\begin{aligned}
w &= 36\times10^{-4}\quad\ldots\ldots[2b]\\
\sigma_1 &= 1600\\
\sigma_2 &= 3200
\end{aligned}\right\}\ [1b]
\qquad
\left.\begin{aligned}
k &= 2;\\
v &= \frac{1}{\sqrt{1+3\cos^2\beta}}\\
\sigma_1 &= 3200v\\
\sigma_2 &= \sigma_1\left(2-\frac{1}{v^2}\right)
\end{aligned}\right\}\ [1c]
$$

Head (see Table 1)

$$
\left.\begin{aligned}
\mathsf{L} &= [7.6]\log(2+\sqrt{3}\sin\beta)v - \left[\frac{8\sqrt{3}}{10}\right]v^2\sin\beta\\
&= \frac{8\sqrt{3}}{10}\left[\frac{9.5}{\sqrt{3}}\log(2+\sqrt{3}\sin\beta)v - v^2\sin\beta\right] = 0.8\sqrt{3}\,\mathsf{L}'\\
u &= \frac{pa^2}{2Et}\,\frac{\sqrt{3}}{4}\,\mathsf{L}\cos\beta = \frac{pa^2}{2Et}\,(0.6\,\mathsf{L}')\cos\beta\\
w &= \frac{pa^2}{2Et}\left\{\frac{\sqrt{3}}{4}\,\mathsf{L}\sin\beta + 4[1.7v^2-1]\right\}\\
&= \frac{pa^2}{2Et}\left\{0.6\,\mathsf{L}'\sin\beta + 6.8v^2 - 4\right\}
\end{aligned}\right\}\quad [2c]
$$

Analysis of Local Effects

Cylinder

$$
\lambda_0 = 0.182\ldots\ldots[4]
$$

$$
\left.\begin{aligned}
w &= -0.0043\,e^{-\lambda_0 s}\cos\lambda_0 s\\
\sigma_2 &= -3200\,e^{-\lambda_0 s}\cos\lambda_0 s\\
\tau &= -220\,e^{-\lambda_0 s}(\cos\lambda_0 s - \sin\lambda_0 s)\\
\sigma_{1e} &= +5820\,e^{-\lambda_0 s}\sin\lambda_0 s\\
\sigma_{2e} &= \mu\sigma_{1e} = 1740\,e^{-\lambda_0 s}\sin\lambda_0 s
\end{aligned}\right\}\ [16]\ \text{(Fig. 11)}
$$

Head (see Table 2)

$$
\int^s\lambda ds = 1.15\int_0^s\frac{ds}{\sqrt{R_2}}\quad\ldots\ldots[11]
$$

$$
\left.\begin{aligned}
w &= 0.0043\left(e^{-1.15\int_0^s\frac{ds}{\sqrt{R_2}}}\right)\cos\left(1.15\int_0^s\frac{ds}{\sqrt{R_2}}\right)\\
\sigma_2 &= \frac{128{,}000}{R_2}\left(e^{-1.15\int_0^s\frac{ds}{\sqrt{R_2}}}\right)\cos\left(1.15\int_0^s\frac{ds}{\sqrt{R_2}}\right)\\
\sigma_{1e} &= \frac{232{,}700}{R_2}\left(e^{-1.15\int_0^s\frac{ds}{\sqrt{R_2}}}\right)\sin\left(1.15\int_0^s\frac{ds}{\sqrt{R_2}}\right)
\end{aligned}\right\}\ [17]\ \text{(Fig. 11)}
$$

* For the head there will be another term here—see Equations [13].

TABLE 1 MEMBRANE DISPLACEMENTS AND STRESSES

1	2	3	4	0	6	7	8	9	10	11	12	13
β deg.	v	Log $[(2+\sqrt{3}\sin\beta)v]$	$\frac{9.5}{\sqrt{3}}$ (col. 3)	$v^2\sin\beta$	L′	0.6 L′ sin β	$6.8v^2-4$	$\frac{w}{pa^2/2Et}$	$\frac{u}{pa^2/2Et}$	σ_1	$\frac{1}{v^2}$	σ_2
0	0.500	0.000	0.000	0.000	0.000	0.000	−2.3	−2.3	0.00	1600	4.00	−3200
5	0.501	0.074	0.406	0.022	0.384	0.020	−2.29	−2.27	0.24	1605	3.99	−3190
10	0.506	0.152	0.834	0.044	0.790	0.082	−2.26	−2.18	0.47	1620	3.91	−3090
15	0.513	0.229	1.257	0.068	1.189	0.185	−2.21	−2.02	0.69	1645	3.80	−2960
20	0.523	0.303	1.663	0.094	1.57	0.32	−2.14	−1.82	0.89	1675	3.65	−2760
25	0.538	0.386	2.12	0.122	2.00	0.51	−2.03	−1.52	1.09	1725	3.46	−2520
30	0.555	0.465	2.55	0.154	2.40	0.72	−1.91	−1.19	1.25	1775	3.25	−2220
35	0.576	0.545	2.99	0.191	2.80	0.96	−1.74	−0.78	1.38	1845	3.01	−1860
40	0.602	0.627	3.44	0.233	3.21	1.24	−1.54	−0.30	1.48	1930	2.76	−1470
45	0.632	0.713	3.91	0.283	3.62	1.54	−1.28	+0.26	1.54	2030	2.50	−1015
50	0.668	0.800	4.38	0.342	4.04	1.86	−0.96	0.90	1.56	2140	2.24	− 514
55	0.709	0.885	4.86	0.412	4.45	2.19	−0.58	1.61	1.53	2270	1.99	+ 23
60	0.756	0.972	5.33	0.495	4.83	2.51	−0.12	2.39	1.45	2420	1.75	605
65	0.807	1.058	5.81	0.590	5.22	2.84	+0.43	3.27	1.33	2590	1.53	1220
70	0.860	1.138	6.24	0.695	5.54	3.07	1.03	4.10	1.13	2750	1.35	1790
75	0.912	1.209	6.63	0.804	5.83	3.38	1.66	5.04	0.94	2920	1.20	2340
80	0.958	1.267	6.95	0.903	6.05	3.58	2.24	5.82	0.63	3070	1.09	2800
85	0.988	1.303	7.15	0.974	6.18	3.69	2.65	6.34	0.32	3160	1.02	3100
90	1.000	1.317	7.22	1.000	6.22	3.74	2.80	6.54	0.00	3200	1.00	3200

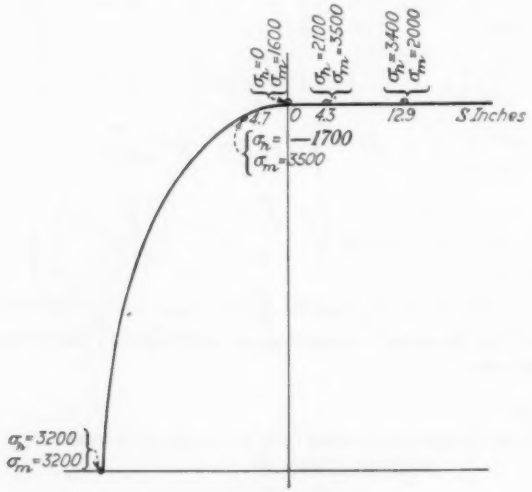


FIG. 10 KEY STRESSES

σ_h = resultant hoop stress — Tensile if written outside contour
σ_m = resultant meridional stress — Compressive if written inside contour

a = radius of cylinder or semi-major axis of ellipse = 40 in.
b = depth of head or semi-minor axis of ellipse = 20 in.
k = a/b = 2
t = thickness of shell for both head and cylinder = 1.25 in.
μ = ratio of lateral to longitudinal strain (in simple tension) = 0.3
E = Young's modulus = 3×10^7 lb. per sq. in.
p = 100 lb. per sq. in.

The length of the cylinder is, say, greater than the diameter.[15]

The investigation of the state of stress in the vessel involves, as has been seen, the membrane analysis and the analysis of local effects. (See page 123.) We are now in position to make each of these. The entire computation is shown in two charts, Tables 1 and 2, and was made with a log-log slide rule. In Table 1 it is not necessary to take so many points, nor need the interval $\delta\beta$ be always the same. The 5-deg. interval was taken for convenience in the v-column of Table 2. The order of filling the columns in Table 1 was 1, 12, 2, 3, ...In the local stress analysis of the cylinder, the values of the functions $e^{-\lambda_0 s}\cos\lambda_0 s$, etc., were found by assigning values to $\lambda_0 s = X$ in Fig. 11 and picking off the corresponding ordinates.

TABLE 2 LOCAL STRESSES AND DISPLACEMENTS

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
β deg.	v	$R_1 = ka^3$	R_1 (mean)	$\delta s = R_1\delta\beta$	$R_2 = ka^3$	R_2 (mean)	$\frac{\delta s}{\sqrt{R_2}}$	$\sum\frac{\delta s}{\sqrt{R_2}}$	$X = 1.15\sum\frac{\delta s}{\sqrt{R_2}}$	$e^{-X}\cos X$	$e^{-X}\sin X$	σ_2	$\sigma_{1\phi}$	$w \times 10^3$
0	0.500	10.0		0.000	40.0		0.000	0.000	0.000	1.00	0.00	3200	0	430
			10.03			40.1								
5	0.501	10.1		0.875	40.1		0.138	0.138	0.159	0.843	0.135	2690	− 684	363
			10.22			40.3								
10	0.506	10.38		0.894	40.5		0.140	0.278	0.320	0.688	0.229	2190	−1320	296
			10.59			40.8								
15	0.513	10.80		0.924	41.1		0.144	0.422	0.485	0.543	0.287	1700	−1635	234
			11.13			41.5								
20	0.523	11.45		0.972	41.9		0.150	0.572	0.657	0.411	0.317	1270	−1773	177
			11.95			42.5								
25	0.538	12.45		1.043	43.1		0.160	0.732	0.842	0.287	0.322	860	−1763	123
			13.06			43.8								
30	0.555	13.66		1.14	44.4		0.172	0.904	1.039	0.180	0.306	530	−1630	78
			14.47			45.3								
35	0.576	15.28		1.26	46.1		0.187	1.091	1.255	0.088	0.271	250	−1400	38
			16.37			47.2								
40	0.602	17.45		1.43	48.2		0.207	1.298	1.491	0.018	0.224	50	−1100	8
			18.80			49.4								
45	0.632	20.15		1.64	50.6		0.233	1.531	1.761	−0.033	0.169	−80	− 795	−14
			22.00			52.1								
50	0.668	23.85		1.92	53.5		0.265	1.796	2.070	−0.060	0.110	−150	− 490	−26

ILLUSTRATIVE EXAMPLE

A vessel with ellipsoidal heads is taken and the following values are assigned to the constants connected with its design:

[15] From the preceding discussion it is seen that it might be taken still shorter, as long as the local effects at the ends do not interfere mutually.

Explanation of Table 2. The value of $\int_0^s \frac{ds}{\sqrt{R_2}}$ for any value of s may be found in terms of β (see Fig. 1) from the relations

$$\delta s = R_1 \delta\beta; \quad R_2 = kav = \frac{ka}{\sqrt{1 + (k^2 - 1)\cos^2\beta}}$$

Neither δs nor R_2 is constant, but both may be so regarded if the interval $\delta\beta$ be taken so short that the variation of R_1 and R_2 within it is negligible. Then we may compute $\frac{\delta s}{\sqrt{R_2}}$ for each such interval, and at any β, such as β_n, we may write

$$\int_0^s \frac{ds}{\sqrt{R_2}} = \sum_{\beta=0}^{\beta_n} \frac{\delta s}{\sqrt{R_2}}$$

Thus we fill out the table as follows: Column 2 from that of the preceding table; column 3 represents the values of the meridional radius at the hoops $\beta = 0°, 5°, \ldots$; column 4 represents the average radius for the zone between the hoops; in column 5 it is recalled that $\delta\beta$ is in radians; columns 6 and 7 are similar to 3 and 4; column 8 is self-explanatory; each entry in column 9 is the sum of the entries in column 8 down to and including its level β_n. From the entries in column 10 as abscissas,

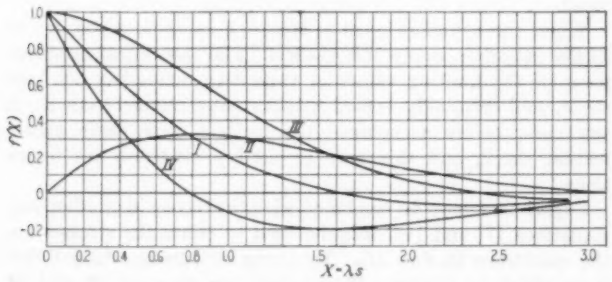


FIG. 11 THE CHARACTERISTIC FUNCTIONS $f_1(X) = e^{-X}\cos X$, $f_2(X) = e^{-X}\sin X$, $f_3(X) = e^{-X}(\cos X + \sin X)$, AND $f_4(X) = e^{-X}(\cos X - \sin X)$

the values of the functions tabulated in columns 11 and 12 are found as ordinates on curves I and II of Fig. 11.

With regard to the choice of intervals $\delta\beta$, each was taken as 5 deg., although nothing about the method requires that it be constant, to bring out the point that it may be taken longer near $\beta = 0$ and shorter as we progress. This is seen from the values of R_1 and R_2 in columns 4 and 7, or from column 8. However, we see from columns 13 and 14 that the stresses decrease numerically as β increases, and we expect to stop the computation as soon as they become negligible, or even as soon as the sum of the numerical values of the meridional membrane and local stresses passes its maximum. Experience with the method, applied to usual designs, will bring out a choice of intervals which will give desired accuracy with the least computation.

If the other quantities τ and σ_{2s} are desired, they may be readily obtained once column 10 is filled. Then the values of the four characteristic functions for any value of X may be picked off from Fig. 11. For example, the computation of τ would mean only the addition of a column which tabulates its values after a single series of slide-rule computations.

The results of the analyses are shown graphically in Figs. 7, 8, and 9. In Fig. 7 the scale for the displacements must be greatly exaggerated, and since the contour of the vessel is drawn to another scale, this leads to an unpleasant distortion. For example, the position of the ellipsoidal membrane after strain is such that the meridional tangent at the junction is horizontal. In Fig. 8 the bending stresses σ_{1s} are those at the outer surface. For the inner surface, the signs are reversed. The curves which display σ_{1i} could thus be obtained by reflecting those shown in the contour of the vessel. The maximum resultant meridional stress is consequently at the inner surface of the knuckle.

Fig. 9 displays the hoop stresses. If it is desired to include the effect of σ_{2s} (or σ_{2i}) it may be done by noting that the values are essentially three-tenths the corresponding values of σ_{1s} (or σ_{1i}). Thus, the ordinates of σ_{1s} in Fig. 8 could be shrunk by this factor and added to those shown.

In general, the significant stresses are those at the crown, the inner surface at the knuckle, and the outer surface of the cylinder near the junction. These are shown in Fig. 10. After one criterion, the favorable head, from the point of view of stress distribution, is one in which these key stresses are about equal. In this case they are all in the neighborhood of 3500, hence this is a favorable design. It should be noted that there is a tensile meridional stress and a compressive hoop stress at the inner surface of the knuckle, consequently a high stress difference there, with the attendant possibility of failure due to shear as the load p is increased. This is especially to be kept in mind in the usual design, for which we may expect a much higher value of the meridional stress at this point.

In his work cited in footnote 9, Huggenberger obtained similar results by use of a method of another form, from the point of view of derivation and application. He had also the opportunity to measure the strains and the angular displacements on a full-scale vessel of about the same dimensions in centimeters as are here given in inches.[16] He found a satisfactory check of computed and experimental results.[17] Using the above method and exactly his data we have the curves given in Fig. 12 as a check of his computations with ours.

GENERAL DISCUSSION OF THE METHOD

The work has been carried through here completely for one ellipsoidal head, but with slight consideration the results will apply to all such heads with $k = 2$, as may be seen from the computation charts. In Table 1, all the variation is taken care of by v in which the only constant is k. The values in columns 9 and 10 for $w/(pa^2/2Et)$ and $u/(pa^2/2Et)$ are for all ellipsoidal heads with $k = 2$. It is seen at once that the values in 11 and 13 could be similarly arranged for $\sigma_1/(pka/2t) = \sigma_1/(pa/t)$ and $\sigma_2/(pa/t)$. In Table 2, the situation is not quite so favorable, but still not unpleasant. A change in the value of a will have the following effect on the variable parts of the local stresses given in columns 11 and 12. If, for example, a becomes $3a$, all the values in column 8 will be multiplied by $\sqrt{3}$, which will therefore be a factor of the sum given in column 9. Consequently X in columns 10, 11, and 12 becomes $\sqrt{3}\,X$, and the values can be found from Fig. 11. Graphically this means that the results could be picked off a new series of curves which are gotten from those in Fig. 11 by shrinking along the axis of abscissas by the amount $\sqrt{3}$.

By considering the general Equations ([16] or [17]) instead of Tables 1 and 2, it is possible to make further conclusions. The factors there may be arranged to involve only the ratios a/t and a/R_2; consequently if a, t, and R_2 be changed in such a way as to keep these ratios constant, the stresses will be the same. From this we have the similarity theorem stated in the previous footnote. The stresses are in general con-

[16] It may be shown that, for one material and the same uniformly distributed inner pressure, geometrically similar vessels develop equal local stresses in similarly located points.

[17] See Fig. 38, p. 169, of the work cited in footnote 9.

trolled by the ratios k, a/t, and a/R_2, and the assignment of actual magnitudes to a, b, t, and R_2 is left free for satisfying other conditions attached to the design.

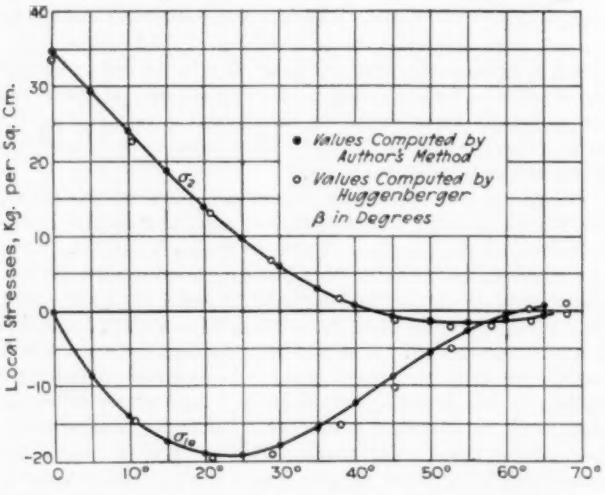


FIG. 12 COMPARISON WITH COMPUTATION OF HUGGENBERGER
(See Table VI, p. 163, "Ueber die Festigkeit der gewölbten Böden und der Zylinderschale.")

The assumption that the head and cylinder are of the same thickness is not a necessary one. A difference in thickness will give other values of Q_0 and M_0 which may be determined by the same method. Thus, for example, it is possible to compute the thickness which should be given to a hemispherical head to have it carry as high a stress as occurs in the cylinder.

In the case of the quasi-ellipsoidal head, the method given takes care of the local effects at the junction of head and cylinder. Then $R_1 = r$ is constant and R_2, of course, is computed from another law than that for the ellipse. This may be written, in one form, as

$$R_2 = r + (a - r) \sec \beta$$

The set of local effects at the shoulder where R_1 is changed will be of less consequence than those at the joint, and deformation of the head tends to reduce them further.

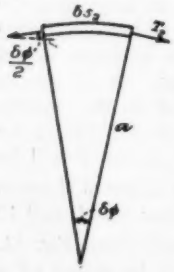


FIG. 13

The display of the displacements and stresses given in Figs. 7, 8, and 9 was designed for pictorial rather than practical purposes. Actually it will be more desirable and easier to develop the meridian of the head as an extension of the element of the cylinder, and lay off all quantities as ordinates with this line as axis of abscissas. The abscissas will then be found from Table 2, column 5, recalling that here we have only the increments in s, and that s at any point will be $\sum_0^{3p} \delta s$. The stress curves will then be roughly symmetrical with respect to the origin (at the joint) and therefore a rough estimate of the numerical maxima of the local stresses in the head will be those of the cylinder:

$$\sigma_2 = \frac{pak^2}{4t} \text{ occurs at edge } s = 0$$

$$\sigma_{1\phi} = \frac{3pk^2}{4\lambda_0^2 t^2} (0.32) \text{ occurs at } s = 0.61\sqrt{at}$$

In the case at hand, which, it will be recalled, is nearly the most favorable from the point of view of stress distribution, these values are

$$\sigma_2 = 3200$$

$$\sigma_{1\phi} = 1890 \text{ at } s = 4.3 \text{ in.}$$

It is seen, therefore, that the computation of these stresses is not an unnecessary refinement. Even here they are of the same order as the membrane stresses, and in less favorable shapes they will of course be higher. From the more exact equations ([17]), we find that the position of these maximum stresses, and the length of the zone in which they damp out, is controlled by the factor $\sqrt{\frac{R_2}{a}\frac{t}{a}}$, which is independent of p. This is of interest in the problem of the location of a welded joint. It is seen also that an increase of t lengthens this zone.

The applicability of the method is limited by the hypotheses contained in the discussion of the form of the head. These involve in spirit the two ratios mentioned, R_2/a and t/a. For the cylinder, $R_2/a = 1$ and the solution of the equations is exact. As the head approaches a flat plate in form, R_2/a becomes rapidly larger, and the assumptions which were made in order to get the solution are not fulfilled. For values of the ratio which characterize the heads shown desirable by experiment, the assumptions should be satisfied, always subject to the restriction that t/a is small.

When the cylinders must be regarded as thick pipes, we no longer expect to apply this theory. The theoretical establishment of the limiting values of these two ratios has not been made, and seemingly logical attacks have not succeeded. In any case, before acceptance the method must have experimental check, and this may be used to establish its limitations.

Experiment may be combined with the theory in another way. Equations [7] and [13] may be used to relate the stresses and the quantities Q_0 and M_0. By using instruments to get data from which to compute the stresses, we may get Q_0 and M_0 from these equations. This will check the theoretically determined values with those actually occurring in the vessel, and we can go on with the computation, using the corrected values.

This is important in the case of a riveted construction, since we do not expect the hypothetical boundary conditions to be exactly realized there. Actually also in welded vessels there is a restraint due to the material of the weld.[18] As has been mentioned, one complete experimental check of this method has been made on a full-size vessel with $t/a \cong 0.03$. Another on a model was made by Geckeler[19] with $t/a \cong 1/240$. Another was made on a vessel with hemispherical heads.[18] These checks may not, however, be regarded as sufficient. In the first place, there are not enough of them to properly test a theory which contains many rather bold assumptions; and in the second place, as far as is known, no tests have been made to compare well-designed quasi-ellipsoidal heads with ellipsoidal heads, axis ratio about 2, by determining the stresses in each while they are still below the elastic limit. Since it is claimed by many good authorities that it is possible to get a favorable distribution without going to the depth of the above ellipsoidal head, it would seem that a settlement of this question would logically precede any discussion of standardization of forms or changing of code.

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[18] Huggenberger, "Festigkeit zylindrischer Kessel gleichen Dicke mit halbkugelförmigen Böden." Festschrift zum 70. Geburtstag von Prof. Stodola. Füssli, Zürich, 1929.

[19] Geckeler, "Über die Festigkeitachsensymmetrischer Schalen." Forschungsarbeiten, V.D.I., Heft 276, 1926.

Appendix No. 1

The Displacements of the Ellipsoidal Membrane

THE derivation of the formulas for the membrane stresses, and those relating displacement and strain in a shell, are found in various books on the subject.[20] The expressions for the displacements of the ellipsoidal membrane given by Equation [2c] are believed to be new. They are derived as follows:

Consider the elliptical meridian in the xy-plane of Fig. 1. Its equation in parametric form is

$$x = a\cos\psi$$
$$y = b\sin\psi$$

Between the parametric angle ψ and the angle β we find the relation

$$b\cot\psi = a\cot\beta \quad \ldots\ldots\ldots\ldots\ldots [18]$$

From the well-known formula

$$R_1 = \frac{\left[1+\left(\frac{dy}{dx}\right)^2\right]^{3/2}}{\frac{d^2y}{dx^2}}$$

for the above values of x and y,

$$R_1 = -\frac{a^2}{b}\csc^3\beta\sin^3\psi$$

Upon the elimination of ψ by use of Equation [18]

$$|R_1| = \frac{a^2}{b}\frac{1}{\left(1+\frac{a^2-b^2}{b^2}\cos^2\beta\right)^{3/2}}$$

From Fig. 1,

$$R_2 = x\sec\beta = a\cos\psi\sec\beta = \frac{a^2}{b}\frac{1}{\left(1+\frac{a^2-b^2}{b^2}\cos^2\beta\right)^{1/2}}$$

Therefore

$$\left.\begin{aligned} R_1 &= kav^3 \\ R_2 &= kav \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots\ldots [19]$$

in which[21]

$$k = \frac{a}{b} \text{ and } v = \frac{1}{\sqrt{1+(k^2-1)\cos^2\beta}}$$

These formulas are more convenient than those in terms of x and y, especially in the following derivation. From the theory of thin shells we have the equations

$$\left.\begin{aligned} u &= \cos\beta\int_0^\beta \frac{\epsilon_1 R_1 - \epsilon_2 R_2}{\cos\beta}\,d\beta \\ w &= u\tan\beta + \epsilon_2 R_2 \end{aligned}\right\} \quad \ldots\ldots\ldots [20]$$

in which

ϵ_1 = meridional strain
ϵ_2 = hoop strain,

and the relations between these and the stresses are

$$\left.\begin{aligned} \epsilon_1 &= \frac{1}{E}(\sigma_1 - \mu\sigma_2) \\ \epsilon_2 &= \frac{1}{E}(\sigma_2 - \mu\sigma_1) \end{aligned}\right\} \quad \ldots\ldots\ldots\ldots [21]$$

For σ_1 and σ_2, the values given by Equations [1] are used with the values of R_1 and R_2 given by Equation [19]. The introduction of all these quantities into the integrand of Equation [20] gives

$$u = \frac{p\cos\beta}{2Et}\int_0^\beta \frac{1}{\cos\beta}\left\{(v-v^3)\left[v-v^3(1-2\mu)\right]\right\}\frac{d\beta}{v^2}$$
$$= \frac{pk^2a^2(k^2-1)}{2Et}\cos\beta\left[\int_0^\beta v^3\cos\beta d\beta - (1-2\mu)\int_0^\beta v^4\cos\beta d\beta\right]$$

The integrals may be evaluated exactly; then u takes the form given by Equation [2c]. From this the value of w is found in the form given in Equation [2c], by use of the values of ϵ_2 and R_2 determined from Equations [21], [1], and [19].

The Cylindrical Shell With Edge Loads

The differential equation for the normal displacement w is derived from the equilibrium of the element shown in Fig. 5 under the forces and couples:

T_2, the normal hoop force
Q, the meridional shearing force
M_1, the meridional bending couple
M_2, the hoop bending couple.

We have assumed the normal stresses to be uniformly distributed over the thickness of the shell; therefore $\sigma_2 = T_2/t$. The meridional stress σ_1 is here zero. From Equation [21] the hoop strain is

$$\epsilon_2 = \frac{W}{a} = \frac{1}{E}\frac{T_2}{t}$$

Then

$$T_2 = \frac{Etw}{a} \quad \ldots\ldots\ldots\ldots\ldots [22]$$

From Fig. 13, the radial component of $T_2\,\delta s$ is

$$T_2\,\delta s\sin\frac{\delta\varphi}{2} = T_2\,\delta s\frac{\delta\varphi}{2} = T_2\,\delta s\frac{\delta s_2}{2a}$$

Then, from [22], the radial component of both hoop forces on the element is

$$T_2\,\delta s\frac{\delta s_2}{a} = \frac{Etw}{a^2}\,\delta s\,\delta s_2$$

For unit arcs in each direction s and s_2 this is Etw/a^2; in other words, this component, which represents the resistance to radial displacement of the element, varies directly as the amount of that displacement. This is the condition that the strip behave as a beam on an elastic foundation. The elastic constant of this foundation, the so-called *modulus of foundation*, is here Et/a^2.

Since we have a two-dimensional state of stress with lateral deformation in bending prevented, E is replaced by $E/(1-\mu^2)$, and the usual beam formula becomes

$$\frac{d^2w}{ds^2} = -\frac{M_1}{D} \quad \ldots\ldots\ldots\ldots\ldots [23]$$

[20] For example, A. and L. Föppl, "Drang und Zwang," vol. II, Oldenbourg, Munich, 1928. For convenience, they are given, along with a fuller treatment of this whole question, in "The State of Stress in Thin-Walled Pressure Vessels," by W. M. Coates, Engineering Research Bulletin, University of Michigan.

[21] See "Handbuch der Physik," vol. VI, "Mechanik der elastischen Körper," in article by Geckeler.

in which $D = \dfrac{Et^3}{12(1-\mu^2)}$

From Equation [23],

$$\frac{d^2w}{ds^2} = -\frac{1}{D}\frac{dM_1}{ds} = -\frac{1}{D}Q \quad \ldots\ldots\ldots\ldots [24]$$

From the equilibrium of the element of Fig. 5 with respect to forces normal to the middle surface,

$$\frac{Etw}{a^2}\,\delta s\,\delta s_2 + Q\,\delta s_2 - \left(Q + \frac{dQ}{ds}\,\delta s\right)\delta s_2 = 0$$

$$\therefore \frac{dQ}{ds} = \frac{Etw}{a^2}$$

Equation [24] is differentiated again:

$$\frac{d^4w}{ds^4} = -\frac{1}{D}\frac{dQ}{ds} = -\frac{Etw}{Da^2} = \frac{-12(1-\mu^2)}{a^2t^2}\,w$$

after D is replaced by its value. Then

$$\frac{d^4w}{ds^4} + 4\lambda_0^4\,w = 0 \quad \ldots\ldots\ldots\ldots\ldots [3]$$

in which

$$\lambda_0 = \sqrt[4]{\frac{3(1-\mu^2)}{a^2t^2}} \quad \ldots\ldots\ldots\ldots\ldots [4]$$

This equation has the complete solution

$$w = e^{-\lambda_0 s}(a_1 \cos \lambda_0 s + a_2 \sin \lambda_0 s) + e^{+\lambda_0 s}(a_3 \cos \lambda_0 s + a_4 \sin \lambda_0 s)$$

in which the constants of integration are a_1, a_2, a_3, a_4. Since the deflections w decrease numerically with an increase in the distance s from the edge, the terms with the factor $e^{+\lambda s}$ are physically impossible for the long cylinder here taken. Then $a_3 = a_4 = 0$.

At $s = 0$, $Q = Q_0$ and $M = M_0$; therefore

$$Q_0 = -2D\lambda_0^3[(a_1 + a_2)]$$

$$M_0 = 2D\lambda_0^2[a_2]$$

and

$$a_1 + a_2 = -\frac{Q_0}{2\lambda_0^3 D},\quad a_2 = \frac{M_0}{2\lambda_0^2 D},\quad a_1 = -\frac{1}{2\lambda_0^3 D}(Q_0 + \lambda_0 M_0)$$

The equation of the radial-displacement curve of the cylinder becomes

$$w = -\frac{1}{2\lambda_0^3 D}\,e^{-\lambda_0 s}[Q_0 \cos \lambda_0 s + \lambda_0 M_0(\cos \lambda_0 s - \sin \lambda_0 s)] \ldots [5]$$

and from this we obtain the expressions for the forces and the bending couples given by Equations [7].

The Head With Edge Loads

The analysis follows the lines laid down in the previous paragraph, except that approximations are introduced.

The bending moments in a thin shell are

$$M_1 = D\left[\frac{1}{R_1}\frac{d\Theta}{d\beta} + \mu\frac{\tan\beta}{R_2}\Theta\right]$$

$$M_2 = D\left[\frac{\tan\beta}{R_2}\Theta + \frac{\mu}{R_1}\frac{d\Theta}{d\beta}\right] = \mu M_1 + D(1-\mu^2)\frac{\tan\beta}{R_2}\Theta$$

in which

$$\Theta = \frac{u}{R_1} - \frac{1}{R_1}\frac{dw}{d\beta}$$

represents the angular deflection of the normal to the middle surface of the shell, and is taken positive in the same sense as the angle β. The terms in the brackets represent the changes in the first and second curvatures of the middle surface. Thus, $\dfrac{1}{R_1}\dfrac{d\Theta}{d\beta}$ is the change in $\dfrac{1}{R_1}$, and $\dfrac{\tan\beta}{R_2}\Theta$ is the change in $\dfrac{1}{R_2}$. For example, the cylinder has a constant zero meridional curvature, and the second term drops out in the bracket of M_1 since $\beta = 0$; correspondingly the first term drops out of M_2; also the u-term drops out of Θ since the curvature $\dfrac{1}{R_1} = 0$. The equations for the cylinder then become $M_1 = -D\dfrac{d^2w}{ds^2}$ and $M_2 = \mu M_1$ as we have them.

Introducing the hypothesis that $u = 0$, we find $\Theta = -dw/ds$, and the expressions for the moments become

$$M_1 = -D\left[\frac{d^2w}{ds^2} + \frac{\mu\tan\beta}{R_2}\frac{dw}{ds}\right]$$

$$M_2 = \mu M_1 - D(1-\mu^2)\frac{\tan\beta}{R_2}\frac{dw}{ds}$$

Guided by the knowledge of the behavior of a cylinder, we assume that for heads of the type hypothesized, the term in dw/ds may be neglected in comparison with that in d^2w/ds^2. After the solution is obtained, we may return with the value found for w and investigate this question.[22] With this assumption

$$M_1 = -D\frac{d^2w}{ds^2}$$

The discussion of the hoop strain is as with the cylinder, with R_2 in place of a. The radial component of both hoop forces together is Etw/R_2 for each element $1 \times 1 \times t$.

The equilibrium of the element of Fig. 6 with respect to radial forces gives

$$\left(Q + \frac{dQ}{ds}\delta s\right)\delta s_2 - Q\delta s_2 - \frac{Etw}{R_2^2}\,\delta s\,\delta s_2 = 0 \quad \therefore \frac{dQ}{ds} = \frac{Etw}{R_2^2}$$

The equilibrium of the element with respect to meridional moments gives

$$\left(M_1 + \frac{dM_1}{ds}\,\delta s\right)\delta s_2 - M_1\delta s_2 - Q\delta s\delta s_2 = 0 \quad \therefore Q = \frac{dM}{ds}$$

Then

$$M_1 = -D\frac{d^2w}{ds^2},\quad Q = \frac{dM}{ds} = -D\frac{d^3w}{ds^3},\quad \frac{Etw}{R_2^2} = \frac{dQ}{ds} = -D\frac{d^4w}{ds^4},$$

$$\frac{d^4w}{ds^4} + \frac{Etw}{DR_2^2} = 0, \quad \text{and} \quad \frac{d^4w}{ds^4} + 4\lambda^4 w = 0$$

in which λ is as defined in Equation [11].

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[22] From Equation [17], we see that except at $\beta = 0$ the value found for w bears out this assumption for a sufficiently small t/a. At the edge we may expect some effect of the neglected term, and experiments show a slight moment here, as against the predicted value $M_0 = 0$.

Determination of the Edge Loads When Cylinder and Head Are Joined

Let w_m denote the radial displacement of the membrane and w_l that due to the edge loads.
At the edge hoop circle,

$$w_m = \frac{pa^2}{2Et}(2 - \mu) \text{ for cylinder, from Equation [2}b\text{]}$$

$$w_m = \frac{pa^2}{2Et}(2 - \mu - k^2) \text{ for head, from Equation [2}c\text{]}$$

$$w_l = -\frac{1}{2\lambda_0^3 D}(Q_0 + \lambda_0 M_0) \text{ for cylinder, from Equation [2}c\text{] for } s = 0$$

$$w_l = +\frac{1}{2\lambda_0^3 D}(Q_0 + \lambda_0 M_0) \text{ for head, from Equation [2}c\text{] for } s = 0$$

The first condition stated under this heading in the paper itself becomes

$$w_{m]\text{cyl.}} + w_{l]\text{cyl.}} = w_{m]\text{head}} + w_{l]\text{head}}$$

$$\therefore Q_0 = \frac{\lambda_0^3 D p a^2 k^2}{2Et} = \frac{pk^2}{8\lambda_0}$$

As for the second condition, the membrane displacement gives no angular deflection at the edge because of symmetry, and the edge load gives that found by substituting the value of w in the expression for Θ.
At the edge of the cylinder, $s = 0$, $\lambda = \lambda_0$,

$$Q_{0]\text{cyl.}} = -Q_{0]\text{head}} \qquad \Theta_{0]\text{cyl.}} = \frac{1}{2\lambda_0^3 D}\left(-Q_0 - 2\lambda_0 M_0\right)$$

At the edge of the head, $s = 0$, $\lambda = \lambda_0$,

$$\Theta_{0]\text{head}} = \frac{1}{2\lambda_0^3 D}\left(Q_0 - 2\lambda_0 M_0\right)$$

Since ds for the head equals $-ds$ for the cylinder

$$\Theta_{0]\text{head}} = -\Theta_{0]\text{cyl.}}$$

$$\therefore M_0 = 0$$

Acknowledgment

In addition to the works referred to in the various footnotes, the author wishes to acknowledge the approximate method due to Meissner (*Schweizerische Bauzeitung*, July 4, 1925), and to mention once more that of Geckeler (footnote 19).

Discussion

D. B. Wesstrom.[23] The writer confesses that he is interested mainly in the practical application of the theory presented in this paper. The general attitude of the pressure-vessel industry on the subject of accurate calculation of strains and stresses in heads is a passive one. The designing engineer today is satisfied with the knowledge, supported by both experiment and theory, that the ellipsoidal and quasi-ellipsoidal heads with an axis ratio of 2 to 1 are the best type of head from the combined standpoint of cost, manufacturing convenience, and maximum utilization of the strength of the material without exceeding the working stress. Regardless of this, the accurate calculation and determination of strains and stresses in heads of pressure vessels will become increasingly important in the next few years and will be of paramount importance in the case of vessels designed to operate at very high temperatures, where the effect of "creep" in the steel is introduced. This is readily seen when it is noted that an excess of the actual working stress over the design working stress of only 10 or 15 per cent may double or triple the amount of "creep" or cumulative plastic deformation, thus shortening the life of the vessel.

It is gratifying to know, then, that theory is ahead of practice on the subject of stresses in heads, and the author is to be congratulated for his admirable investigation of this subject, and also for his thorough treatment of it and the excellent manner in which he presents it.

Attention is called to the fact that the A.S.M.E. Boiler Code Committee has recently considered tolerances for quasi-ellipsoidal heads in which a deviation of the head section from the true ellipse of 1.25 per cent of the head diameter is allowed, which tolerance, it is believed, was determined by manufacturing limitations. It would be interesting if the author were to apply his equations to this problem and determine the ratio of maximum stress in such a head to that in a truly ellipsoidal one for various plate thicknesses.

The mathematical curves derived by the author are remarkably similar to corresponding curves derived from experimental data presented by E. Höhn in footnote 5, and likewise to curves determined independently by T. W. Greene and the writer. The similarity is evident in the shape of the curves and in the general location and magnitude of the maximum and minimum points.

Although the author's curves show the point of normal meridional stress, or point of zero bending, at the edge of the head, where $s = 0$, the experimental data referred to place this point a certain distance beyond, somewhere in the cylinder. This disagreement may be due partly to inaccuracies in locating the edge point in an actual experiment and partly to the neglect of minor effects, as in footnote 22. It seems to the writer that this is the most favorable location of a welded joint, and he would like to ask the author to amplify his remarks on this matter. Would the reinforcing effect of the weld modify the location of this optimum point?

This paper furnishes an excellent example of the application of mathematics to refined mechanical design.

A. B. Kinzel.[24] The author is to be complimented on a very ingenious piece of work. The basic assumptions made are the same as those which have been used by many others in the past, including Huggenberger's work on the same subject, although the mathematical treatment is very much more simple than that used by Huggenberger. The author has given a flexible and reasonably simple method of calculating stresses in full heads, and by means of this calculation has checked the values obtained by the more complicated method of computation, showing the local stress concentration at the junction of ellipsoidal heads with cylindrical shells.

In the past few years considerable progress has been made in the design of heads for pressure vessels, and the elliptical head with the 2 to 1 major-minor axis ratio has been largely adopted. Huggenberger has further calculated the local stresses, with the result that a seven-arc head in which the stresses are further reduced in the general vicinity of the point of tangency of the head and cylindrical shell is recommended. It seems that the author has an opportunity of giving an even better solution of this problem, due to his greatly simplified mathematical attack. It

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occurs that in a large pressure vessel the sections near the ends of the cylindrical section could be formed so as to be conical or even portions of ellipsoids or paraboloids, and that by so forming these sections the stress concentration at the continuity of the elliptical section with the cylindrical section could be greatly reduced.

The introduction of the elliptical head to replace the short knuckle radius head has reduced the stress concentration from approximately three to five times that of the calculated stress, to approximately 25 per cent local stress increase, and a design of the shape of the head to reduce this 25 per cent to approximately 5 per cent is available in a rather complicated form by Huggenberger's work, and should be available in a more simple form by the extension of the work of the author. This leads to the hope that in the not too distant future the design of pressure vessels with a factor of safety less than $5^1/_2$ will be produced as it has been clearly demonstrated in the past that the factor of safety was really about 2, with the factor of ignorance about 3.

S. Timoshenko.[25] The author must be congratulated for his very interesting paper which is of practical importance to designers of pressure vessels. He has reduced the problem of stress analysis in pressure vessels to that of bending of a beam on an elastic foundation. In this manner the problem is considerably simplified as compared with existing solutions of the same problem given by Huggenberger and by Geckeler. The numerical example illustrates that all calculations can be made with sufficient accuracy by using the slide rule, hence the method can be adapted for practical use.

The only criticism the writer has refers to the derivation of Equation [10]. Although this equation is satisfactory at the joint and gives satisfactory values of local stresses at this place, the use of the same equation at points far away from the joint can be justified only on the basis of a more thorough investigation of this equation. The derivation given on the twelfth page, in which δS_2 is considered as a constant, is satisfactory at the joint, but at the points distant from the joint the variation of the width δS_2 of the strip should be taken into account.

F. L. Everett.[26] Several of those who have discussed the paper have inquired whether physical tests have been made which check the analytical expressions obtained by the author. The writer is aware that tests have been performed on full-size riveted vessels which showed good qualitative agreement with the analysis which has been presented. The design of heads was the usual dished type having a radius of curvature equal to the diameter of the drum and a short knuckle radius of several inches.

The drums were subjected to hydrostatic pressure. Simultaneously sufficient measurements of strain were made to determine the stresses in meridional and hoop directions on and in longitudinal and circumferential directions on the cylindrical shell. Marked local effects, particularly at the knuckle, were observed, and even an area under compression was noted here.

The pressure during a later part of the test was increased until failure of the head at the knuckle resulted. A head of "basket-handle" form, however, on test showed superior characteristics and was adopted as the design for future practice.

Author's Closure

The author wishes to thank those who have discussed this paper for the appreciation which they have expressed. Although the presentation of a method for computing the stresses in shells requires considerable mathematical notation, the mere statement of the results can be made in a very simple form, which permits rapid calculation, for such cases as here discussed. It is hoped that this has been made clear, in spite of errors which have occurred in the preprint, especially in the placing of the parts of the analysis in the illustrative example.

As regards the discussion by Mr. Wesstrom, the author agrees that the interest of the engineer is in the application of the theory, not in its presentation. This means that he need be concerned only with formulas [1], [2] and [16] or [17]. The general passive attitude on the computation of stresses in heads may be explained by the difficulty attached to such computations. As soon as results may be obtained rapidly on paper, it is to be expected that the designs will follow and check them. It is shown here how easily this may be done for a certain class of heads, also that some experimental check has been made. It is hoped that more will follow. It cannot, however, be said that "theory is ahead of practice" in this field since a simple fundamental method for the computation in manheads does not exist, nor even one for full heads beyond a certain thickness. As already mentioned in the paper good authorities, e.g., Höhn on the empirical side and Biezeno on the theoretical, do not agree that the 2 to 1 ellipsoid is the most desirable form. The author knows of no publications, either of theory or of experimental results, which demonstrate that such a ratio of diameter to depth is the best for full quasi-ellipsoidal heads. Höhn, for example, would not agree with this, but maintains that a shallower one is fully as satisfactory from the viewpoint of stress distribution, and more satisfactory from that of manufacture. As for manheads, there has been no demonstration that this ratio is the best. Experiment shows (Abhandlung 70 in the series referred to in footnote 8) that near the manhole flange there should be a zone which is concave outward. A theoretical discussion of the comparative merits of quasi-ellipsoidal and true ellipsoidal heads requires the investigation of the second set of discontinuity effects which are introduced at the junction of the knuckle and the spherical cap in the former, but are absent in the latter. These are of the same character as the local effects at the junction of head and cylinder, although smaller in good designs. A design could possibly be arranged in which these sets of local effects were opposed. Without a demonstration that the mathematical hypotheses are satisfied in the physical problem, such computation could not be conclusive. The question is an interesting one, of fundamental importance in settling the shape of the head, and also one for which theory will at least provide the design for a series of experiments. Concerning the location of the weld, it is again necessary to introduce experiment, as indicated in the paragraph just preceding the Appendix, p. 10, to determine if the value of M_0 is significantly different from the value zero, which this method gives. If such is the case, the position of zero moment is determined from Equation [7], with M_0 and Q_0 at their experimentally determined values (see the corrected curves in footnote 17). The reinforcing effect of the weld would affect this position for any value of M_0 other than zero. The amount of this reinforcing could be determined only by experiment. Some measurements are given in the work cited in footnotes 9 and 18.

With regard to the discussion by Mr. Kinzel it is perhaps in place to call attention again to the two pieces of work mentioned in the acknowledgment at the end of the Appendix. Both preceded the mentioned work of Huggenberger, although neither contains the extended experimental discussion found there. This paper seeks to give the theoretical side another presentation, and makes simplifications in the computation. The application of this method to a seven-arc head is, as suggested, not difficult; in the computation presented the elliptical meridian may be

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thought of as a 14-arc one. As the number of arcs decreases, however, the question of the discontinuity effects at their junctions enters, as mentioned before for the meridian made up of two arcs.

The comparison of multi-arc heads and ellipsoidal ones is probably only a mathematical question, and one in which the manufacturer is hardly in position to distinguish one form from the other. The practical problem, as suggested by Mr. Westrom, is the comparison of the two-arc head (or maybe three-arc) with the ellipsoidal one. An important reason for not including a "most favorable form" computation in this paper is that the use of the method is not justified as the head becomes flat, and the limitations have not been established to show just how far up it may be used. Until this is done, it is questionable practice to use this method to compare the behavior of heads so alike in character as the ellipsoidal and the head built up on a multi-arc meridian, since the differences may very well be of the same order as the effects neglected in the method. It may be said that the discontinuity effects cannot be entirely removed at the junction of a cylinder with any full-dished head, entirely concave inward; however, they approach zero as the depth of the head becomes infinite. Even when not zero, they may be negligible in good designs. Thus, for the hemispherical head they are so small that the greatest resultant stress in the vessel occurs in the cylinder. As the head is made shallower, to approach the ellipsoidal shape of diameter twice the depth, the greatest resultant stress in cylinder and head approach each other. The rigorous establishing of a mathematically "most favorable form" in which they are equal is still outstanding, and the question is open whether it may be accomplished in a shallower head than that given by the 2 to 1 ellipsoid. Mathematically there is no advantage in thinking of the meridian as made up of parabolic arcs, and practically, for such short lengths as those taken, the distinction between arcs of a circle, ellipse, parabola, or torus is not realizable. As for a conical head, the method permits extension to give the discontinuity effects set up at the junction with the cylinder, provided the cone is not too flat. Another point to consider is that there is a meridional force to account for at the junction of a cone and a cylinder. The hope for a fundamental solution to the problem of the design of the head, after the form desired, will be realized most speedily by the use of existing theory and experimental methods in combination. The cases for which this has been done by Huggenberger, as mentioned by Mr. Kinzel and also in the paragraph preceding the Appendix, are for shells of the type $\frac{\text{thickness}}{\text{diameter}} \leqq 0.03$. Thicker shells, such as occur in high-pressure vessels, are still to be investigated; and there is small reason to expect that the approximate method (as by Geckeler, Huggenberger, or here) will give accuracy to within five per cent.

Professor Timoshenko's criticism of the derivation of Equation [10] is justified, and the presentation in the Appendix at the place mentioned is decidedly inadequate. The entrance of additional variables beyond those which enter in the case of the cylinder has been taken into account, however, although the derivation does not show this. After their order was found to be negligible, they were dropped off. The explanation for the omission of this discussion is that the author was too drastic in cutting out mathematics which he felt would be out of place in this paper. Instead of adding it here now, a more complete presentation will be given later, in which the question of the range of applicability of this method will also be discussed. This will give a more satisfactory presentation and avoid duplication in publishing.

There was some oral discussion as to experimental check of this method. It will be noted that the paragraph before the Appendix lists some such checks. The work mentioned by Mr. Everett represents another, and one which has not been published, as far as the author knows. It is of interest to get more than a qualitative check, since the question is practically settled from that viewpoint. As has been pointed out, quantitative results can be obtained with no injury to the vessels, and the design of heads can be worked into a settled question as soon as a sufficient number of these results are published.

APM-52-13

Torsional-Vibration Dampers

By J. P. DEN HARTOG,[1] and J. ORMONDROYD,[2] EAST PITTSBURGH, PA.

In this paper the authors present a theory of the action of friction dampers on systems in torsional resonance, with special application to gas- and Diesel-engine installations. Model tests carried out to check this theory are described. The effect of introducing springs in the Lanchester damper is also discussed.

THE frictional-vibration damper—commonly known as the Lanchester damper—is used extensively on gas and Diesel engines to limit the amplitudes of critical torsional vibrations. The damper (Fig. 1) consists of two flywheels *a*, free to rotate about one end of the cranskhaft on bushings *b*, and driven by the crankshaft through friction rings *c*. The flywheels are pressed against the friction rings by means of loading springs and adjustable nuts *d*.

The purpose of the damper is to dissipate energy when the amplitude of vibration on the damper hub *e* tends to become large, as in the case of torsional resonance of the engine. When violent oscillations occur on the damper hub, the inertia of the flywheel prevents it from following the motion; the resultant relative motion between the hub and the flywheels gives rise to rubbing on the friction surfaces and a consequent loss of energy. When the oscillations on the hub are small, the flywheels stick to the friction rings, follow the motion of the hub, and no energy is dissipated. Thus the friction-driven flywheel possesses the valuable property of dissipating energy only when energy should be dissipated—that is, when the vibrations of the crankshaft threaten to become large.

It is clear that the most advantageous location of the damper on the engine is the place where the torsional-vibration amplitudes are largest. Since a gas or Diesel engine always carries a rather large flywheel of which the torsional-vibration amplitudes are small, the logical place for the damper is the end of the crankshaft away from the main flywheel.

Lanchester's patent considers only *viscous* friction as a means of driving the flywheel and dissipating energy. This limitation was probably suggested by the apparent ease of mathematical treatment, since viscous friction gives rise to forces simply proportional to the rubbing velocity. Although the mathematical analysis of the viscous-friction damper is simple, the practical difficulties inherent in its construction have always been so great that this type of friction is never used. All practical dampers known to the authors employ ordinary dry friction as the means of energy dissipation. So far as they know, no rational theory of the action of a dry-friction damper has yet been published. Dampers are just made—not designed.

The Energy Method of Calculating Effects of Friction Dampers

It is proposed to discuss the action of dry-friction dampers by means of the energy method. This method of calculating the effects of the damper depends on the fact that the energy put into the vibration by the variable gas pressures in the cylinders equals the energy dissipated by all the damping actions of the system. Wydler[3] was the first to discuss torsional resonance in Diesel engines by means of this method. He considered all the damping actions lumped into one single damping force which was proportional to the velocity. The equivalent constants of this assumed law he obtained empirically by measuring the amplitude of torsional vibration at resonance in several Diesel engines. Lewis[4] used the same methods, but assumed that the damping arose chiefly from the internal hysteresis of the crankshaft metal. He used Rowett's[5] experimentally de-

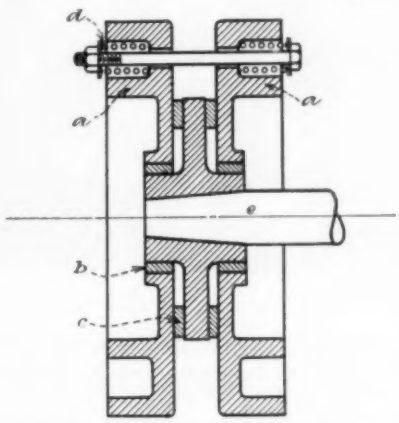


Fig. 1 Torsional Friction Damper

termined law of energy loss in mild steel under torsional stresses. These experiments show that the energy loss per unit volume per cycle is independent of the frequency and proportional to some power of the maximum shear stress existing during the cycle. Below the fatigue limit Lewis showed that the energy loss per unit volume per cycle W followed the simple law:

$$W = H\alpha_0^{2.3} \quad [1]$$

where H is a constant calculated from the test results and α_0 is the amplitude of twist in the test piece. Above the fatigue limit, α_0 must be raised to a higher power to express the results found in the experiment.

Assumptions Made in Applying Energy Method

In applying the energy method to the system of an engine with a friction damper the following assumptions are made:

1 The existence of sinusoidal vibrations in the crankshaft and 90-deg. phase relationship between impressed moments and resultant motions at resonance
2 The dissipation of all vibrational energy in the friction damper and in the metallic hysteresis of the crankshaft
3 The existence of Coulomb friction in the damper.

The first assumption of approximately sinusoidal motion is justified by numerous records of vibration at resonance which always show almost pure sinusoidal motion. The assumption

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] H. Wydler, "Drehschwingungen in Kolbenmaschinenanlagen," Berlin, 1922.

[4] F. M. Lewis, Trans. Soc. Nav. Arch. & Marine Eng., vol. 33 (1925), p. 109.

[5] Rowett, Proc. Roy. Soc. London, vol. 89.

of a 90-deg. phase relationship between the oscillating moments and the motions they cause is used in calculating the input energy. It can be shown that this leads to accurate values of damping losses in the case of damping forces which are proportional to velocity, even when the maximum amplitudes occur with the phase angle far from 90 deg. For the case of a damping force which leads to the law that the energy loss per unit volume per cycle is proportional to the square of the maxi-

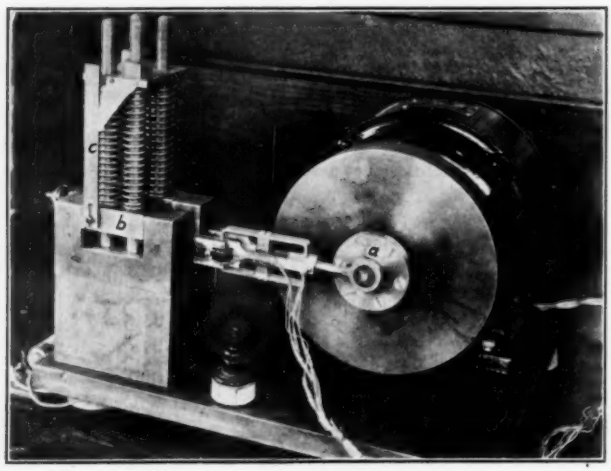


FIG. 2 FRICTION TESTING DEVICE

mum stress, the phase angle *is* always 90 deg. at resonance.[6] For the authors' case the experimental results will have to justify its use.

The second assumption indicates that the presence of all friction, windage, and impact dampings which may be effective in limiting torsional vibration are being neglected. The laws of these dampings are not known, and it appears that experimental facts can be explained without assuming their existence.

The third assumption has been found to be true with a high degree of accuracy by means of a series of tests, which will now be described.

TESTS SHOWING EXISTENCE OF COULOMB DAMPING

A variable-speed motor (Fig. 2) carrying a fairly heavy flywheel was made to drive a small, light table (2 sq. in. in area) back and forth by means of an $^1/_8$-in. eccentric *a*. The table was lined with the kind of brake lining commonly used in dampers, and any desired pressure up to 100 lb. per sq. in. could be exerted on it by means of a pressure plate *b* and four calibrated springs *c*. The table was coupled to the eccentric by means of a connecting rod on which two magnetic extensometers were mounted. These extensometers were fed with 500-cycle current and were connected in a Wheatstone-bridge circuit with an oscillograph. The amplitude of the 500-cycle wave recorded in the oscillograph varies linearly with the widths of the air gaps in the extensometers, so that a record is obtained of the changes in length of the connecting rod. In order to translate this into stress, a calibration was carried out as shown in Fig. 3. The table *a* was taken from between the pressure plates, swung aside, and loaded with a dead weight *b* by means of a pulley *c*. The reading thus obtained in the oscillograph made it possible to read the records directly in terms of the force in the connecting rod.

Fig. 4 shows a typical record thus taken. It is seen to have a practically flat top and bottom, so that Coulomb's law of a friction independent of the velocity actually holds for the conditions in the damper. The curvature of the vertical sides of the diagram can be explained by elasticity in the connecting rod and in the shaft and bearings of the driving motor. In case this flexibility were zero the vertical sides should theoretically be straight lines, while with the elasticity they should be parabolic arcs. The theoretical parabolas have been completed on the record by dotted lines.

Viscous friction would have given a sine curve instead of one with flat tops and bottoms. The presence of oil, even of a bath of oil, with the ordinary pressures of about 30 lb. per sq. in. did not alter the shape of the curve, but merely decreased its height, i.e., decreased the "coefficient of friction." Only with an oil bath and a pressure as low as 5 lb. per sq. in. was it possible

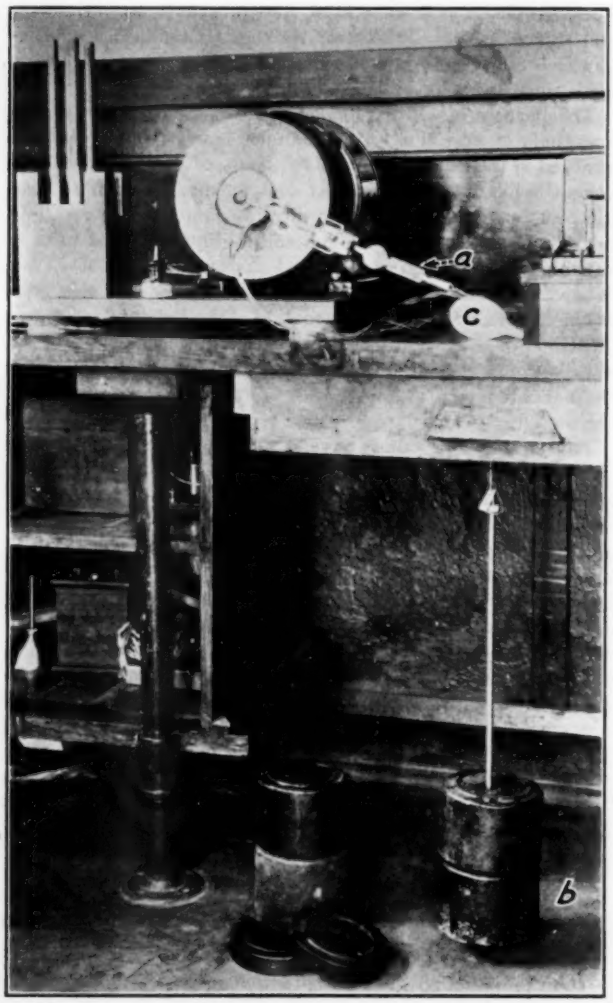


FIG. 3 METHOD OF CALIBRATING

to obtain an approximately sinusoidal diagram. The fact that it was actually possible to get this diagram with such low stresses in the connecting rod shows conclusively that the flat tops of the Coulomb damping diagram are not caused by any mechanical hysteresis or binding in the connecting rod or extensometers. It may also be mentioned that the stresses in the rod were of the order of 100 times the stresses caused by the dynamical effect of the table, which fact was also checked experimentally.

[6] See Kimball and Lovell, *Mechanical Engineering*, May, 1927, p. 440.

The data obtained for the coefficients of friction in the various tests show that for each particular surface condition Coulomb's law holds fairly well, but that there is an enormous variation in the coefficient for various surface conditions. In other words, once the machine was running, changes in pressure hardly affected the coefficient of friction (there was a tendency for a decrease in f for increasing pressure); neither did changes in the running speed of the motor have any appreciable effect (the tendency being to increase f with the speed). The influence of the condition of the surface is best illustrated by the following history of a series of tests:

1 Freshly cleaned and sandpapered surface.... f = 50 per cent
2 The same surface after being exposed to the atmosphere (Pittsburgh) for two months... f = 40 per cent
3 Oil just put on surface..... f = 18 per cent
4 Oil soaked in overnight..... f = 15 per cent
5 Soaked in gasoline for one hour..... f = 21 per cent
6 Sandpapered..... f = 27 per cent

The temperature also has an appreciable effect as shown by the following test. At room temperature f was found for a clean dry surface to be 42 per cent. The pressure plates and table were then heated to about 60 deg. cent. by a gas flame, which decreased f to 30 per cent. Immediately thereupon a fan was turned on the machine, cooling it down to room temperature in about five minutes, when f became 39.5 per cent.

On actual engine dampers the surfaces are always found to be more or less greasy and hot, so that for design purposes a coefficient of friction of not higher than 15 per cent should be taken.

In general, these tests have shown that the assumption of Coulomb friction in the damper is well justified.

The System to Be Analyzed

Since in nearly all practical cases of torsional resonance in gas or Diesel engines it is only the lowest natural frequency of the system which is dangerous, the system can be represented in the simplified form shown in Fig. 5.

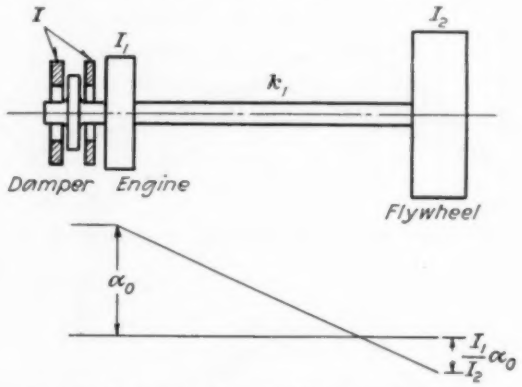


Fig. 5 Simplified System Used for Analysis

the total moment of inertia of all the slipping parts of the friction damper.

One Harmonic Torque Acting on Engine

At resonance the effects of all the harmonic gas-pressure torques except the one in synchronism with the natural frequency are negligible. Therefore, we can consider a single simple harmonic torque as acting on the engine.

Let this torque be

$$M = M_0 \sin \omega t \quad [2]$$

where $\omega = 2\pi$ times the natural frequency of the system. For methods of calculating the value of M_0, see Lewis' paper.[4] This moment gives rise to an oscillation in the system in which I_1 and I_2 have the relative amplitudes at resonance shown in the diagram of Fig. 5. Let the motion of I_1 be

$$\alpha = \alpha_0 \sin (\omega t - \varphi) \quad [3]$$

This is also the motion of the damper hub in the simplified scheme of Fig. 5. When α_0 is known, the motions and stresses in the whole system are known.

M is superimposed on the constant average working torque of the engine, and α is superimposed on the steady rotation of the engine.

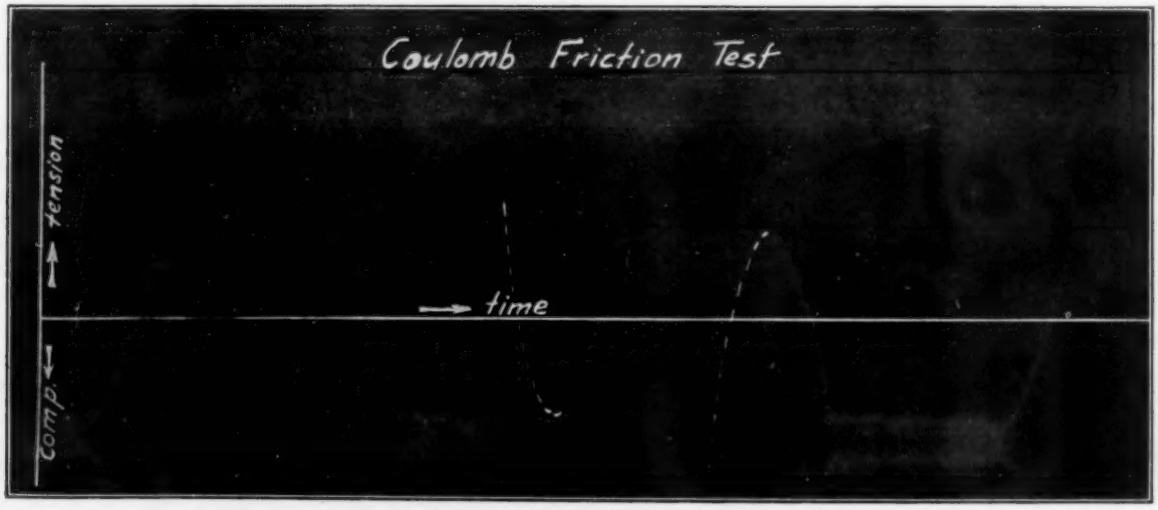


Fig. 4 Oscillogram Showing Coulomb Friction

Here I_1 represents the moment of inertia of the rotating and reciprocating parts of the gas or Diesel engine; I_2 the moment of inertia of a flywheel or electrical generator driven by the engine; k_1 the torsional stiffness of the shaft (torque units per radian twist) connecting the engine and the flywheel; and I

Energy Put Into Vibration per Cycle

Let W = work per cycle put into the vibration by M. Then

$$W = \int_0^{\frac{2\pi}{\omega}} M d\alpha = \pi M_0 \alpha_0 \sin \varphi \quad [4]$$

M and α are given as functions of the time in [2] and [3] and $2\pi/\omega$ is the period of one complete cycle. At resonance φ = 90 deg. according to the first assumption made, and the work per cycle is

$$W = \pi M_0 \alpha_0 \quad [5]$$

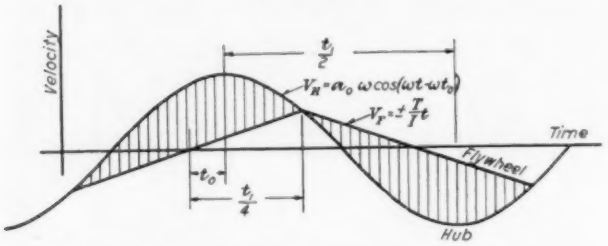


Fig. 6(*a*) Continuous Slipping of Flywheel

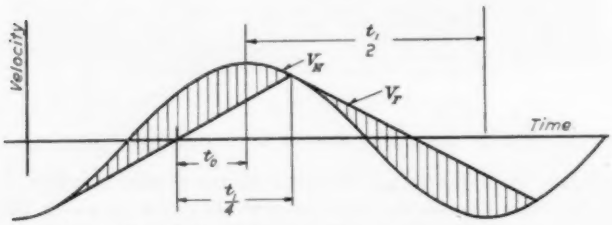


Fig. 6(*b*) Limit of Continuous Slip

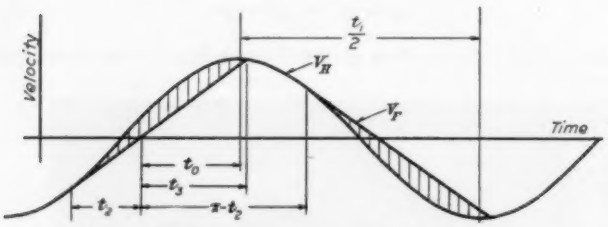


Fig. 6(*c*) Partial Slip

Friction Torque in Damper

In calculating the energy lost in the damper it is necessary to know the torque T required to slip the damper flywheels on the friction rings. This torque is

$$T = 2fF \frac{2}{3} \frac{R_2^3 - R_1^3}{R_2^2 - R_1^2} = 2fFR \quad [6]$$

where f = coefficient of friction on friction rings
$F = nkd$ = total load on friction surfaces
n = number of loading springs
k = stiffness of loading springs (lb. per in. deflection)
d = deflection or set-up of loading springs
R_1 = inner radius of friction ring
R_2 = outer radius of friction ring
R = radius at which the total friction force can be considered as concentrated.

Equation [6] is based on the assumption of a uniform distribution of pressure over the surfaces of the friction rings. Because of variations in the pressure distribution which commonly occur in some practical designs, R may vary considerably from the value as found in Equation [6].

First Approximation to Damper Action

If we assume that the damper flywheels are so heavy that they rotate uniformly at the average angular velocity of the crankshaft, the angular oscillation of the damper hub is the relative oscillation between the hub and the flywheel. The work done against friction for every cycle is

$$W = 4T\alpha_0 \quad [7]$$

If this work is equated to the input work given in Equation [5], it can be verified that the value of α_0 becomes infinite for $T/M_0 < \pi/4$ and zero for $T/M_0 > \pi/4$, which obviously is physically impossible.

To overcome this difficulty, J. Dickson, of the William Beardmore Co., Glasgow, Scotland, proposed to consider the friction damper in conjunction with the shaft-hysteresis damping as used by Lewis[4] and given in Equation [1]. This leads to the energy equation

$$\pi M_0 \alpha_0 = 4T\alpha_0 + H\alpha_0^{2.3} \quad [8]$$

Here H must be considered as a constant obtained by using Rowett's experimental results and integrating over the whole crankshaft.

Solving [8] leads to

$$\alpha_0 = \sqrt[1.3]{\frac{\pi M_0 - 4T}{H}} \quad [9]$$

This approximate solution is not satisfactory in many respects. It gives no method of proportioning the damper in advance. This is natural since it is based on the assumption that the damper flywheel is "heavy." Next it leads to the result that $\alpha_0 = 0$ when $T = \frac{\pi}{4} M_0$, and furthermore no real solution for α_0 exists for $T > \frac{\pi}{4} M_0$. This does not coincide with observations on engines equipped with dampers, which show that α_0 never becomes zero, no matter how large T is made.

These difficulties led to a closer study of the action of the damper.

Actual Motions of Flywheel in Friction Damper

Under steady conditions the damper flywheel will be rotating with an average angular velocity equal to the average angular velocity of the crankshaft. Superimposed on this motion it will have a motion relative to the oscillating hub. This motion will be periodic in nature and its frequency will be the same as the hub oscillation frequency. There are three possible types of motion for the damper flywheel:

1 If the acceleration of the damper hub is always less in value than T/I the flywheel never slips but follows the motion of the hub completely
2 The damper flywheel may slip continuously
3 The flywheel may follow the motion of the hub during part of the cycle and slip during the rest of it.

These three possibilities can best be understood by an inspection of the velocity diagrams of Fig. 6. During the slipping phase of the flywheel motion it is acted upon by a constant friction torque. Therefore, from Newton's law of motion the angular velocity of the flywheel must be a straight-line function of time. Due to the fact that the hub motion follows a symmetrical law, the damper-flywheel motion must also be sym-

metrical. These conditions can be met only in the manner shown in Fig. 6(*a*). The velocity of the flywheel increases when the hub velocity is greater and decreases when the hub velocity is less than the flywheel velocity. The slope of the flywheel velocity curve is equal to T/I. As the damper loading springs are tightened up the flywheel velocity curve becomes steeper. If the damper hub has a fixed amplitude independent of the damper torque the velocity curves of the hub and the flywheel finally cross at points where their slopes are equal as shown in Fig. 6(*b*). This represents the limit of damper torque below which slipping occurs continuously. If the damper torque is increased further the flywheel clings to the hub until the acceleration of the hub is large enough to shake it loose as shown in Fig. 6(*c*).

Work per Cycle Done in Friction Damper

In Appendix No. 1 it is shown that the work per cycle done against friction in the damper is equal to the friction torque T multiplied by the total shaded area of Fig. 6. For a given constant motion of the damper hub, the work per cycle W thus calculated is shown as a function of the friction torque T in the curve of Fig. 7. The ordinates of this curve are not W and T themselves, but values proportional to those quantities; in fact, they are W and T multiplied by combinations of the characteristic constants of the damper flywheel and the hub motion. This has the advantage of making the relationship in Fig. 7 dimensionless and therefore applicable to any system where I, ω, and α_0 are known. The curve in Fig. 7 indicates that the work per

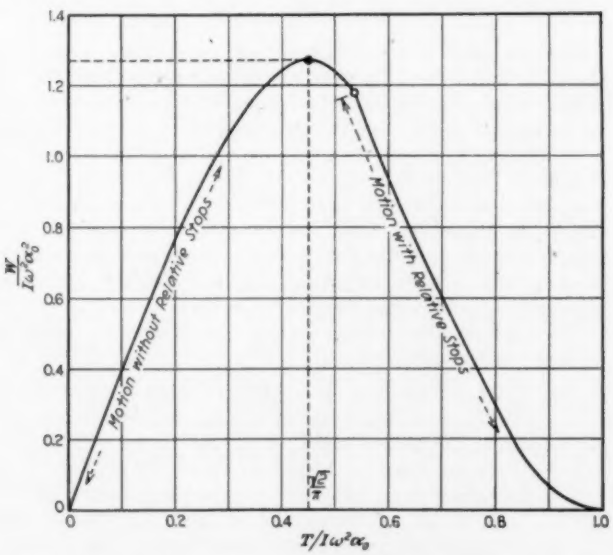


Fig. 7 Work Absorbed in Damper Vs. Friction Torque

cycle absorbed in the damper is zero when the damper torque is zero, and again when the flywheel is following the hub motion completely—that is, when the damper is a solid part of the engine. At some intermediate value of T the work absorbed reaches a maximum. Designers, of course, are interested in obtaining the maximum amount of energy dissipation out of any damper they may build. From Fig. 7 it can be seen that

$$W_{max} = \frac{4}{\pi} I\omega^2\alpha_0^2 \quad [10]$$

when

$$T = \frac{\sqrt{2}}{\pi} I\omega^2\alpha_0 \quad [11]$$

Having an expression for the work absorption in the damper, we can calculate α_0 by equating the energies [7] and [10]:

$$\pi M_0\alpha_0 = \frac{4}{\pi} I\omega^2\alpha_0^2 \quad [12]$$

Solving for α_0,

$$\alpha_{0_{min}} = \left(\frac{\pi}{2}\right)^2 \frac{M_0}{I\omega^2} \quad [13]$$

and

$$T = \sqrt{2}\,\frac{\pi}{4} M_0 \quad [14]$$

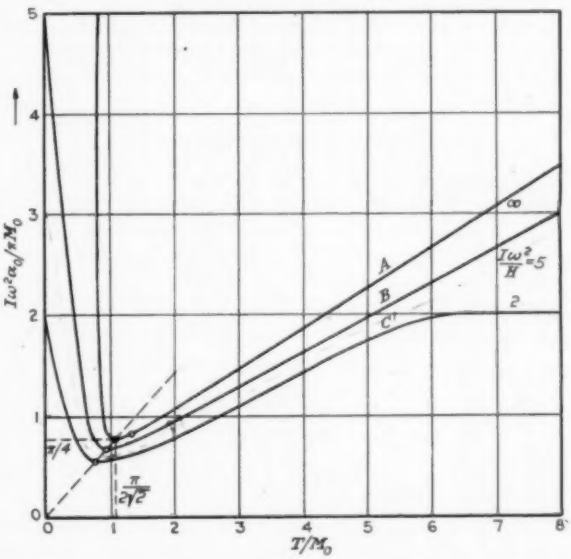


Fig. 8 Resonant Amplitude of Vibration Vs. Friction Torque on Damper

Equations [13] and [14] give the chief design data for any damper when M_0 is known. The amplitude α_0 at resonance will be chosen such that the stresses in the shaft between I_1 and I_2 (Fig. 5) will be below the fatigue limit. The value of α_0 desired determines the magnitude of I when M_0 and ω are the given data. A simple design rule to follow is to limit α_0 to such a value that the maximum oscillating stress is about 10 or 12 per cent of the ultimate tensile strength of the crankshaft steel. This follows from the facts that in most steels the ultimate strength in torsion is about half the ultimate strength in tension; that the fatigue limit in torsion is again about half the ultimate strength in torsion; and that oil holes and abrupt changes of section at the crank cheeks may give rise to stress concentration which may easily double the nominal calculated stress. This chain of facts leads to the simple rule quoted above.

Influence of Shaft Hysteresis Damping

All existing test data on hysteresis losses in steels can be described with fair accuracy for *low* stresses by the law[4]

$$W = H\alpha_0^2 \quad [15]$$

This law is a good approximation where the maximum stresses are below the fatigue limit. Therefore we can consider it to hold when a properly designed and adjusted damper is applied to an engine. The value of H is determined from the experimental curve on the crankshaft steel and an integration over the whole volume of the shaft. Considering hysteresis, Equation [12] becomes

$$\pi M_0 \alpha_0 = \frac{4}{\pi} I\omega^2 \alpha_0^2 + H\alpha_0^2 \quad [16]$$

and

$$\alpha_{0_{\min}} = \left(\frac{\pi}{2}\right)^2 \frac{M_0}{I\omega^2}\left(\frac{1}{1+\frac{\pi H}{4I\omega^2}}\right) \quad [17]$$

$$T = \sqrt{2}\,\frac{\pi}{4} M_0 \left(\frac{1}{1+\frac{\pi H}{4I\omega^2}}\right) \quad [18]$$

In most gas or Diesel engines the hysteresis will give very little assistance to a properly designed damper, set at its optimum torque—it will decrease the amplitude of motion at resonance to about 85 per cent of the value at which the friction damper alone would hold it.

Effect of Variation of Damper Torque

The amplitude of motion at resonance α_0 depends on the damper torque T. This is discussed in Appendix No. 1, sections C and D, while the results of the analysis are shown in Fig. 8. As in Fig. 7, the ordinates of Fig. 8 are dimensionless quantities, making the diagram applicable to any system.

The action of the damper alone, neglecting shaft hysteresis, is shown in curve A, indicating that the damper is completely ineffective until $\frac{T}{M_0} > \frac{\pi}{4}$. At $T = \sqrt{2}\,\frac{\pi}{4} M_0$, the amplitude of motion has its minimum value

$$\alpha_{0_{\min}} = \left(\frac{\pi}{2}\right)^2 \frac{M_0}{I\omega^2}$$

These results are the same as given in Equations [13] and [14] in the earlier part of the paper. As T approaches infinity, the values of $I\omega^2\alpha_0/\pi M_0$ approach infinity also. Physically this means that the amplitude of motion increases until the accelerations are large enough to shake the damper loose, no matter how large T is made.

Curve A, Fig. 8, indicates that if the damper torque is slightly less than the optimum value the damper becomes ineffective; but if the torque is made larger than the optimum value the resonant amplitude increases only slightly over the minimum possible value. In other words, the damper can easily be set too loosely; but it is rather insensitive to overtightening. This is the saving feature which has made it possible to build dampers without designing them. In the hands of a practical man, a damper is more likely to be set too tight than too loose.

Effect of Damper Torque Variation With Combined Damping

In any real system to which a damper may be attached, other types of damping always exist. This is shown by the fact that finite amplitudes of motion always occur in real systems at resonance, even when no damper is used. The largest natural damping existing in a system as shown in Fig. 5 is due to the metallic hysteresis of the shaft material under varying stresses.

For ease of discussion it was assumed in Appendix No. 1 that the hysteresis loss in the shaft followed the simple law $W = H\alpha_0^2$. Under this assumption curves B and C in Fig. 8 were drawn for values of $I\omega^2/H = 2$ and 5. The meaning of this ratio $I\omega^2/H$ is $\pi/4$ times the ratio of the damper work absorption to the shaft-hysteresis work absorption, as can be seen from Equations [10] and [15].

The simple law of hysteresis loss actually only holds for stresses below the fatigue limit. However, the hysteresis loss increases very rapidly for stresses above the fatigue limit, so that there the simple law assumed does not hold. For this reason the curves B and C of Fig. 8 only represent the action of combined damping qualitatively for amplitudes where high stresses exist. The actual curve for any given system can be constructed graphically by using the real hysteresis-loss curve. In real systems where hysteresis damping acts by itself, the amplitude of motion at resonance is usually so large that the maximum stresses in the system are far above the fatigue limit. For this reason a simple law of hysteresis loss as assumed above (or the one used by Lewis) cannot be used in predicting amplitudes at resonance. For instance, Porter,[7] using the law of hysteresis deduced by Lewis from Rowett's tests for *low stresses* in *mild* steel, calculates resonant amplitudes which are much larger than the actual values observed in the engines tested by him. Much more energy was lost than the simple law could predict; and Porter's results would have been off even further if the law he used had been deduced from tests on actual crankshaft steel, since Föppl[8] has shown that the hysteresis loss in crankshaft steels is much less for a given stress than in ordinary mild steel.

From this it can be seen that curves B and C of Fig. 8 are quantitatively accurate only near the minimum values of $I\omega^2\alpha_0/\pi M_0$; for higher values the curves give only a qualitative description of the effect of combined damping. They show that the resonant amplitudes always remain finite and that the amplitude increases much faster for damper torques less than the optimum value than for damper torques larger than this value.

Field Experience

Field experience with dampers indicates that they "run away" or fail to function when the damper torque becomes too small. What is more, it has been found that the damper torque varies up and down even at one setting of the loading springs. If the damper is supposedly set at the optimum torque, a variation below that torque may let the damper "run away," while a variation above it will have little effect. Curve B, Fig. 8, represents about the conditions in most real engines. The ratio of the slopes of the two sides of the curve is about 14:1, showing the relative danger of too little damper torque as against too much damper torque.

There are several practical reasons why damper torques may become smaller than expected:

1 If the optimum torque is set by calculation, the assumed value of the coefficient of friction may be taken larger than the actual value

2 If the coefficient of friction is known for ordinary temperatures it may become small as the damper heats up (see results of friction tests, ante). If the damper torque is set by experiment with the damper at ordinary temperatures, it becomes smaller as the damper heats up

3 If the surfaces of the friction rings are not perfectly true, the rings may "ride" near the inner radius or near the outer radius. In some designs this might make a difference of 50 per cent between the maximum torque and the minimum torque. A variation this large could easily account for the failure of a damper to operate properly, and variations of this magnitude have been found in practice

4 If a damper is designed by means of the approximate formula [8], the damper torque is always made at least

[7] F. P. Porter, Trans. A.S.M.E., vol. 50 (1928), no. 17, paper no. APM-50-8.

[8] O. Föppl, *Z.V.D.I.*, Sept. 15, 1928.

30 per cent less than the optimum value and the system depends on hysteresis or other accidental damping for success

5 Oil frequently gets into the damper friction surfaces

6 Any accidental obstruction which keeps the friction surfaces separated or which hinders the free rotation of the damper flywheels will tend to make the damper fail

7 If the loading springs are very stiff, wear in the friction rings drops the torque rapidly.

A safe operating rule is to set the damper torque cold to somewhat above the optimum value. From Fig. 8, it can be seen that this does not lead to much larger amplitudes than the minimum, and it safeguards the damper against any decrease in torque which may reasonably arise in practice.

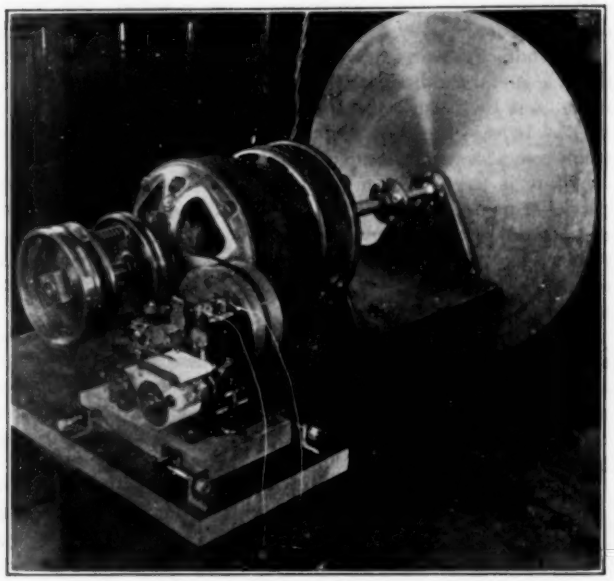


FIG. 9 MODEL USED FOR TESTS

A safe design rule is to assume the coefficients of friction to be 0.15 or less. The actual value of the coefficient of friction does not affect the damper-flywheel design, but it does determine the loading spring design. The smaller the coefficient of friction the larger the spring load required for a given torque.

How will a damper designed for a certain oscillating frequency operate at some other frequency? The question has no practical significance for lower frequencies than the design frequency, since the latter is usually the lowest natural frequency of the system to which the damper is attached.

Higher frequencies (second or third natural frequencies of the system) usually are not dangerous even without a damper, because the relative displacements in the shaft are so large that hysteresis alone is quite sufficient to keep the amplitudes within safe limits. However, if they ever become dangerous it is seen from the expression on the vertical ordinates of Fig. 8, that the damper is even more effective at high frequencies than at low ones.

MODEL TESTS

In order to test the accuracy of the theory given above, a model, Fig. 9, was built. It consisted of a single-phase induction motor, coupled on one side to a flywheel and having a damper mounted on the other side.

The instantaneous power input into a single-phase motor is:

$$[e \sin \omega t][i \sin (\omega t + \varphi)]$$

which expression can be transformed into

$$\frac{ei}{2} \cos \varphi - \frac{ei}{2} \cos (2\omega t + \varphi)$$

The power input consists of a constant part and of a variable part with a frequency equal to twice that of the line voltage. The amplitude of the power input variation is $ei/2$, or EI watts, that is, the product of the readings of a voltmeter and an ammeter. In case all this power is converted to torque, the amplitude of the torque variation can be found to be[9]

$$\frac{7.14EI}{\text{r.p.m.}} \text{ ft-lb.}$$

With the assumption that 10 per cent of this power input goes into losses, we finally find:

$$\pi M = \frac{240\ EI}{\text{r.p.m.}} \text{ in-lb} \qquad [19]$$

The input torque was determined by means of this formula.

The angular oscillations of the shaft near the damper hub were measured by means of a torsiograph.[10] Tests were carried out with three different damper flywheels with moments of inertia of 0.12, 0.27, and 0.42 lb-in-sec.^{2}, respectively.

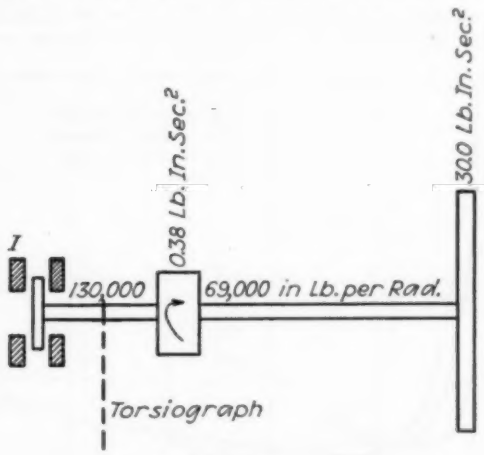


FIG. 10 MODEL CHARACTERISTICS

The test procedure was as follows: The loading springs were set to a certain pressure and the friction torque on the damper was measured by means of a spring scale, while turning the apparatus over very slowly by hand. Then the machine was run, varying the frequency of the input and consequently the speed until the resonance point was found, where a record and the various meter readings were taken. The machine was then stopped in order to reset the loading springs for the next run.

It was found that in case the damper was either entirely loose or fully locked, the critical speed was distinct and could be determined with good accuracy. When the damper was working near its optimum point, however, there was really no "critical speed," the amplitude being almost constant over quite a broad speed range.

It is evident that the critical speed of the machine without

[9] For a more fundamental treatment, see the paper by Kimball and Alger, Trans. A.I.E.E., 1924, p. 730.

[10] Ormondroyd, Trans. A.I.E.E., 1926, p. 443.

damper is higher than the one with the damper locked on. (See Fig. 10.) In fact, in the authors' set-up the variations were rather large; the experiment gave the following values:

From 500 r.p.m. to 435 r.p.m. with the light damper
From 500 r.p.m. to 340 r.p.m. with the medium damper
From 500 r.p.m. to 318 r.p.m. with the heavy damper.

The change from the high critical speed to the lower one as a function of the tightening of the loading springs took place in a highly interesting manner. For any pressure below the optimum the critical speed was equal to the *upper* one, that is, to the one without any damper. In the neighborhood of the optimum point the critical speed became indeterminate as described above, while for pressures distinctly higher than the optimum the critical speed was equal to the *lower* one practically independent of the degree of tightening. The behavior below the optimum pressure may be regarded as an experimental verification of the theoretical result that Coulomb damping in a vibrating system does not affect the critical speed, provided the damping be not so large that partial stops occur in the motion. (See Appendix No. 2.)

On account of the flexibility between the rotor and the damper the angles measured at the torsiograph are not exactly equal to the ones on the damper hub; neither is the work input per cycle equal to α_0 times the expression [19]. To both quantities a suitable correction was applied. The observed data, reduced to a dimensionless form, have been plotted in Fig. 11.

A comparison with Fig. 8 shows that the general shape of the curves obtained corresponds to the requirements of the theory. However, it appears that the experimental curves are considerably lower than the theory predicts. This may be due to any of the following causes.

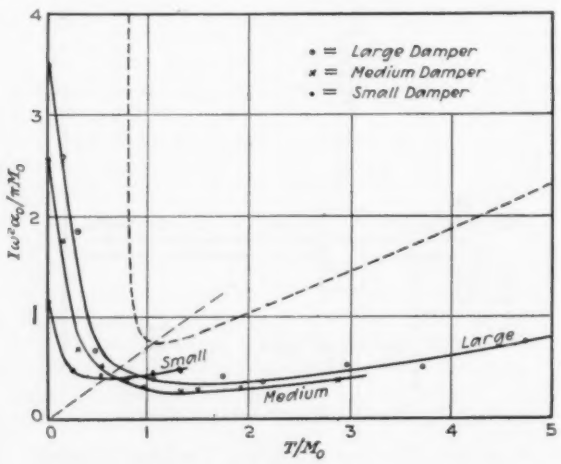


FIG. 11 TEST RESULTS

a Uncertainty in the determination of the electrical input torque
b The existence of additional damping in the system, apart from the friction damper
c Inherent inaccuracy of the three assumptions on which the theory is based.

At the present it cannot be decided yet which of these, or possibly other, factors is the cause of the numerical discrepancy. Additional tests on a yet simpler model are now in course of progress.

It must be remarked, however, that these tests indicate that the damper is approximately twice as effective as the theory predicts, and that a design by means of the formulas [13] and [14] should lead to entirely satisfactory results.

THE DAMPER WITH SPRINGS

It may be possible in certain cases to improve on the design of the damper by coupling its flywheel to the shaft not only by friction but also by some torsional spring. This adds an extra degree of freedom to the system, and consequently adds one more natural frequency to it. In cases where the damper inertia is rather small with respect to the rest of the system, the original critical speeds will not be appreciably altered by the addition of the damper, while the additional critical speed will be nearly equal to that of the damper itself.

With the same three assumptions made before, the work absorbed per cycle by the damper friction has been calculated in Appendix No. 2 and the results are shown in Fig. 12. In this figure ω is still the frequency of the motion of the damper hub, while ω_n is the natural frequency of the damper considered by itself. In case $\omega_n = 0$; that is, for a springless damper, the result coincides with the one of Fig. 7. For the general case that $\omega_n \neq 0$, it has not been possible to obtain a solution where partial gripping occurs [see Fig. 6(*c*)]. However, this limitation is not serious, for we know that all the curves of Fig. 12 have to pass through the point $T/I\alpha_0\omega^2 = 1$, since there the damper is entirely locked. This determines the general trend of the curves also in the region of partial stops.

The graph indicates that by the application of springs a considerably greater amount of energy can be dissipated by a damper of a given size. This increase is obtained on account of the greater relative motion between the damper flywheel and the hub. The limitations of this design, therefore, are the stresses in the springs.

By equating the dissipated and the input energies, an equation is found in Appendix No. 2 from which the angle α_0 can be solved in the same manner as before. This leads to the results shown in Fig. 13, which again coincides with Fig. 8 for the case that $\omega_n = 0$. It is seen that the optimum angle for a given size of damper can be considerably reduced by means of the springs. In fact, theoretically the optimum angle α_0 becomes zero for $\omega = \omega_{n1}$, that is, for a damper in resonance with the forced frequency of the system (dynamic vibration absorber).[11] However, this means a great relative displacement between the damper flywheel and its hub, which would break the damper springs. The stress in the damper springs is determined by the amplitude θ_0 of the relative motion between the damper flywheel and hub. This amplitude is calculated in Appendix No. 2, the results being shown in Fig. 14. It is seen that the stress becomes larger the nearer to its natural frequency the damper is operated. Since the efficacy of the springs increases in the same manner, the design of the damper becomes a matter of compromise. If possible, the natural frequency of the damper should be kept outside the running range. If this is not practicable, a good expedient is the application of motion-limiting stops. For instance, if a gas engine has to be designed for a speed range of from 500 to 1200 r.p.m. and it appears that the main critical speed is around 800 r.p.m., the first thing to attempt is to shift this critical out of the running range by altering the flexibility of the shafts. Quite often this is not possible. Then a damper might be designed with a resonance corresponding to an engine speed of 640 r.p.m. This damper becomes quite light, as can be seen from the relations in the Figs. 12 and 13. If no precautions are taken, however, the damper will be destroyed when the engine is operated near 640 r.p.m. Therefore stops should be provided,

[11] See paper by the authors, Trans. A.S.M.E., 1928, paper no. APM-50-7.

restricting the play of the damper flywheel to an amount insuring the safety of its springs. These stops will be idle in the entire range, except near 640 r.p.m. At that speed, however, the engine is safe by itself and does not need any damper.

It has to be remarked that the theory of the damper with springs as given here has not yet had an experimental verification as far as we know.

Summary and Conclusions

1 A theory has been given of the Lanchester friction damper as applied particularly to gas- or Diesel-engine installations.

2 Tests have been conducted on a model, which as a whole backed up the theory. The damper works somewhat better than the theory states.

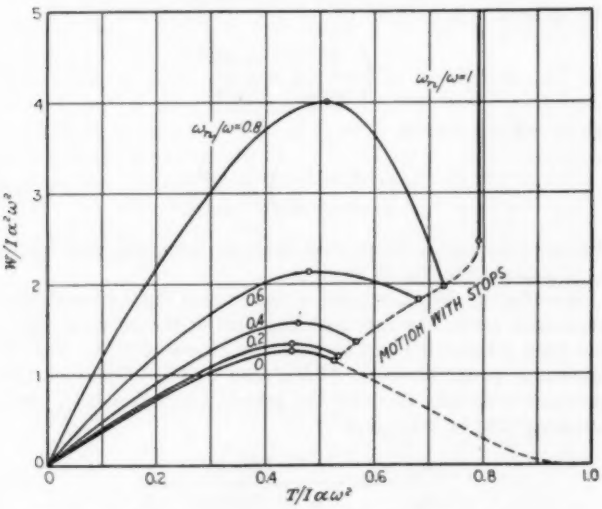


Fig. 12 Work Dissipation in Damper With Springs

3 A possible improvement of the damper, due to the insertion of springs, is discussed, showing that the same effect can be obtained with a damper of much smaller size as when no springs were present.

Appendix No. 1

THEORY OF THE DAMPER WITHOUT SPRINGS

A—Energy Dissipation in Damper

Let W = work done against friction during one cycle of hub motion
v_f = instantaneous "superimposed" angular velocity of damper flywheel
v_h = instantaneous "superimposed" angular velocity of damper hub
$\alpha = \alpha_0 \sin \omega t$ = angular motion of damper hub
ω = 2π times the frequency of damper-hub motion
t = time
$t_1 = 2\pi/\omega$ = period of damper-hub motion
T = torque necessary to slip damper flywheel on friction rings
I = total moment of inertia of the damper flywheels, and
θ = relative angular displacement between damper hub and flywheel.

Then the work done against the friction of the damper during one cycle of the hub motion is:

$$W = \int T d\theta = \int_0^{2\pi/\omega} T \frac{d\theta}{dt} dt = \int_0^{2\pi/\omega} T(v_h - v_f)dt$$

$$= 2T \int_{-\pi/2\omega}^{\pi/2\omega} (v_h - v_f)dt \quad [20]$$

or W is $2T$ times the shaded area in Fig. 6(*a*), 6(*b*), or 6(*c*).

This area will now be calculated. Considering Fig. 6(*a*), 6(*b*), or 6(*c*), the time is taken as zero at the instant the flywheel superimposed velocity is zero and about to become positive, or, more accurately, at the instant the flywheel velocity is equal to its average velocity and about to become greater.

We have

$$v_f = \pm \frac{T}{I} t \quad [21]$$

in which the signs depend on the position in the cycle.

$$v_h = \alpha_0 \omega \cos(\omega t - \omega t_0) \quad [22]$$

These formulas are perfectly general. The further analysis, however, becomes different for the cases of Figs. 6(*a*) and 6(*c*). First calculate the case of continuous slipping [Fig. 6(*a*)]. The time t_0 can be determined from the consideration that $v_f = v_h$ when $t = \pm t_1/4$, or

$$\frac{T}{I}\frac{2\pi}{4\omega} = \alpha_0 \omega \cos\left(\frac{\pi}{2} - \omega t_0\right)$$

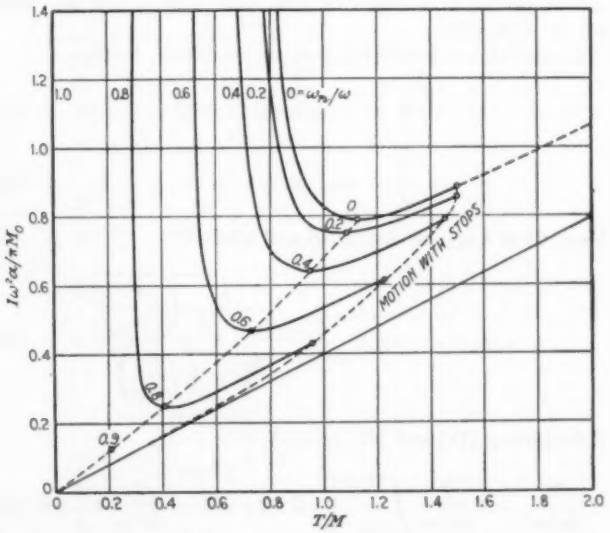


Fig. 13 Amplitudes Vs. Friction Torque for Spring Damper

Solving this for t_0,

$$\sin \omega t_0 = \frac{\pi}{2}\frac{T}{I\omega^2\alpha_0} \quad [23]$$

Substituting [21] and [22] in [20], performing the integration and using the result [23],

$$W = 4T\alpha_0 \sqrt{1 - \left(\frac{\pi}{2}\right)^2 \left(\frac{T}{I\omega^2\alpha_0}\right)^2} \quad [24]$$

which can be written conveniently in the following dimensionless form:

$$\frac{W}{I\omega^2\alpha_0^2} = 4\frac{T}{I\omega^2\alpha_0} \sqrt{1 - \left(\frac{\pi}{2}\right)^2 \left(\frac{T}{I\omega^2\alpha_0}\right)^2} \quad [25]$$

Equations [23], [24], and [25] hold only when

$$\frac{dv_f}{dt} \leqslant \frac{dv_h}{dt} \text{ at } t = t_1/4$$

which is sufficiently clear from the Figs. 6(*a*) and 6(*b*). The limit to which the equations are true [Fig. 6(*b*)] is therefore given by the condition

$$\left(\frac{dv_f}{dt}\right)_{t=\frac{\pi}{2\omega}} = \left(\frac{dv_h}{dt}\right)_{t=\frac{\pi}{2\omega}}$$

Substituting [21] and [22] in this,

$$\frac{T}{I} = \alpha_0\omega^2 \sin\left[\frac{\pi}{2} - \sin^{-1}\frac{\pi}{2}\frac{T}{I\omega^2\alpha_0}\right]$$

or

$$\frac{T}{I\omega^2\alpha_0} = \sqrt{1-\left(\frac{\pi}{2}\right)^2\left(\frac{T}{I\omega^2\alpha_0}\right)^2} = \sqrt{\frac{1}{1+\left(\frac{\pi}{2}\right)^2}} = 0.536 \ldots [26]$$

At this value of $T/I\omega^2\alpha_0$ the damper flywheel begins to move with the damper hub part of the time.

B—Damper Flywheel Locked to Hub Part of the Time

The conditions of motion are shown graphically in Fig. 6(*c*). At the instant $t = -t_2$, it is seen that $v_h = v_f$ and also that $dv_h/dt = dv_f/dt$.

Expressed mathematically, these conditions become

$$\frac{T}{I}t_2 = -\alpha_0\omega\cos\omega(t_0 + t_2) \ldots\ldots [27]$$

$$\frac{T}{I} = \alpha_0\omega^2 \sin\omega(t_0 + t_2) \ldots\ldots [28]$$

From these t_2 can be eliminated and t_0 solved:

$$\omega t_0 = \pi - \sin^{-1}\frac{T}{I\omega^2\alpha_0} - \sqrt{\frac{1-\left(\frac{T}{I\omega^2\alpha_0}\right)^2}{\left(\frac{T}{I\omega^2\alpha_0}\right)^2}} \ldots\ldots [29]$$

Substituting [21] and [22] in [20],

$$\frac{W}{I\omega^2\alpha_0^2} = 2\frac{T}{I\omega^2\alpha_0}\int_{-\omega t_2}^{+\omega t_3}\left[\cos\omega(t-t_0) - \frac{T}{I\omega^2\alpha_0}\omega t\right]\omega dt. [30]$$

The lower limit of integration is determined by [27] and the upper limit by the similar equation

$$\frac{T}{I}t_3 = \alpha_0\omega\cos(t_3 - t_0) \ldots\ldots [31]$$

expressing the fact that at the upper limit the velocities of the hub and the flywheel are equal.

The numerical evaluation of [30] then runs as follows: For a certain value of $T/I\omega^2\alpha_0$, first t_0 is computed from [29]. With this result, t_2 and t_3 are found numerically from [27] and [31], and since the two equations are transcendental, this can only be done by cut and try. With the values t_0, t_2, and t_3 numerically established, the calculation of $W/I\omega^2\alpha_0^2$ from [30] offers no difficulty.

In this manner the right-hand part of the curve Fig. 7 has been determined point by point.

C—Effect of Friction Damper on System

In the main body of the paper it is shown that the work per cycle put into the system, Fig. 5, is:

$$W = \pi M_0\alpha_0 \ldots\ldots [5]$$

The work per cycle absorbed by the damper is given by Fig. 7. Since only for a part of this curve is an explicit mathematical expression available, it will be discussed in the following analysis as:

$$\frac{W}{I\omega^2\alpha_0^2} = F\left(\frac{T}{I\omega^2\alpha_0}\right) \ldots\ldots [32]$$

where F means "function of."

Equating the input and output energies [5] and [32] leads to the general relationship

$$F\left(\frac{T}{I\omega^2\alpha_0}\right) = \frac{\pi M_0}{I\omega^2\alpha_0} \ldots\ldots [33]$$

which can be written

$$F\left(\frac{T/\pi M_0}{I\omega^2\alpha_0/\pi M_0}\right) = \frac{\pi M_0}{I\omega^2\alpha_0} \ldots\ldots [34]$$

This equation gives the relation between $I\omega^2\alpha_0/\pi M_0$ and $T/\pi M_0$ as it is plotted in Fig. 8, curve *A*.

Since for the right-hand part of the curve of Fig. 7 no analytical expression exists, the corresponding part of the curve of Fig. 8 has been calculated point by point by means of [33]. For the left-hand part, however ($T/I\omega^2\alpha_0 < 0.536$), Equation [25] provides a definite form for the general expression [32]. Substituting [25] in [34] gives

$$\frac{I\omega^2\alpha_0}{\pi M_0} = 2\pi \cdot \frac{(T/\pi M_0)^2}{\sqrt{16\,(T/\pi M_0)^2 - 1}} \ldots\ldots [35]$$

representing the equation of the left-hand part of curve *A*, of Fig. 8. It is seen that this curve has a minimum for

$$\frac{T}{\pi M_0} = \frac{\sqrt{2}}{4}; \qquad \frac{I\omega^2\alpha_0}{\pi M_0} = \frac{\pi}{4}$$

The curve shows that the damper is completely ineffective where $T/\pi M_0 < \frac{1}{4}$, that is, for small values of the friction torque.

D—Combined Effect of Friction Damper and Metallic Hysteresis

The friction damper never constitutes the whole damping in any real system to which it may be attached. This is shown by the fact that the amplitude of motion in systems to which no friction damper is attached is always finite even at resonance. If the damper is attached to a system in which the metallic hysteresis in the shaft dissipates an amount of work per cycle according to the law

$$W = H\alpha_0^2 \ldots\ldots [15]$$

Equation [33] must be rewritten as follows:

$$F\left(\frac{T}{I\omega^2\alpha_0}\right) + \frac{H\alpha_0^2}{I\omega^2\alpha_0^2} = \frac{\pi M_0}{I\omega^2\alpha_0} \ldots\ldots [36]$$

Reducing this to the form corresponding to the form [34]

$$F\left(\frac{T/\pi M_0}{I\omega^2\alpha_0/\pi M_0}\right) + \frac{H}{I\omega^2} = \frac{\pi M_0}{I\omega^2\alpha_0} \ldots\ldots [37]$$

The quantity $H/I\omega^2$ appearing in this result indicates the ratio of effectiveness of the shaft hysteresis and of the damper itself. In Fig. 8, the curves B and C have been plotted from the relation [37].

Appendix No. 2

THEORY OF THE DAMPER WITH SPRINGS

IF THE damper flywheel of Fig. 1 is coupled to the shaft not only by plain friction but also by some torsional spring, a theory similar to the one developed in Appendix No. 1 can be given. The first step is to derive an expression for the work absorbed per cycle when the damper hub is executing a harmonic vibration with the amplitude α_0.

If the maximum relative angular displacement θ_0 between the damper flywheel and hub were known, the energy dissipation per cycle $4T\theta_0$ would be known also.

The problem therefore presents itself as the general case of forced vibration in a single-degree-of-freedom system affected by Coulomb damping.[12] Fig. 15 shows a scheme of the system under consideration, presented in linear rather than torsional form merely on account of greater simplicity of the drawing. Let the hub perform the angular motion $\alpha = \alpha_0 \cos(\omega t + \varphi)$, and furthermore let the damper flywheel I be coupled to the hub by the torsional flexibility k (lb-in. per radian) and by the friction torque $\pm T$. Let the angular motion of the flywheel I be β. Then the differential equation of the motion of I becomes

$$I\ddot{\beta} + k[\beta - \alpha_0 \cos(\omega t + \varphi)] \pm T = 0$$

or, using the relative angular motion,

$$\theta = \beta - \alpha_0 \cos(\omega t + \varphi) \quad \ldots\ldots [38]$$

as a new variable:

$$I\ddot{\theta} + k\theta \pm T = I\omega^2\alpha_0 \cos(\omega t + \varphi) \quad \ldots\ldots [39]$$

The + sign of T holds during the half-cycle that $\dot{\theta}$ is positive, while the equation with negative T has to be applied during the time that $\dot{\theta}$ is negative.

At the time $t = 0$, let θ have its maximum positive value θ_0; and let $\dot{\theta} = 0$. Then, assuming that motion starts immediately, $\dot{\theta}$ will be negative, and Equation [39] with the lower sign holds. Its general solution is:

$$\theta = C_1 \sin \omega_n t + C_2 \cos \omega_n t + \frac{I\omega^2\alpha_0}{k} \cdot \frac{\omega_n^2}{\omega_n^2 - \omega^2} \cos(\omega t + \varphi) + \frac{T}{k} \quad \ldots\ldots [40]$$

where $\omega_n = \sqrt{k/I}$ is the natural frequency of the damper.

Substituting the two initial conditions into this expression, C_1 and C_2 can be calculated so that

$$\theta = \theta_0 \cos \omega_n t + \frac{T}{k}(1 - \cos \omega_n t) + \frac{I\omega^2\alpha_0}{k} \cdot \frac{\omega_n^2}{\omega_n^2 - \omega^2} \left[\cos\varphi(\cos \omega t - \cos \omega_n t) + \sin\varphi\left(\frac{\omega}{\omega_n} \sin \omega_n t - \sin \omega t\right)\right] \quad \ldots\ldots [41]$$

Now a "forced vibration" is a periodic motion with the forced frequency ω. Since there is no inherent difference between forward and backward motion in this case, the vibration also has to be symmetrical. In other words, after half a cycle the flywheel I must have reached its other extreme position, or

$$\omega t = \pi \quad \theta = -\theta_0 \quad \dot{\theta} = 0$$

For satisfying these two conditions, Equation [41] contains the parameters θ_0 and φ. Their calculation leads to

$$\sin\varphi = \frac{k\theta_0}{I\omega^2\alpha_0} \cdot \frac{\omega_n^2 - \omega^2}{\omega_n^2} \quad \ldots\ldots [42]$$

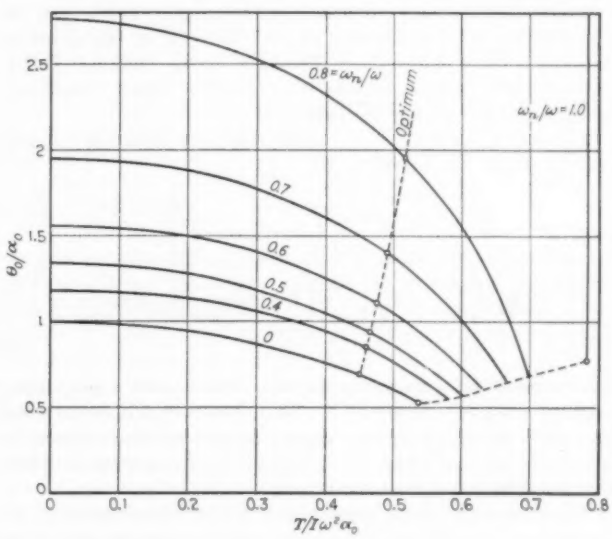


FIG. 14 RELATIVE AMPLITUDE BETWEEN FLYWHEEL AND HUB

$$\theta_0 = \sqrt{\left(\frac{I\omega^2\alpha_0}{k} \cdot \frac{\omega_n^2}{\omega_n^2 - \omega^2}\right)^2 - \left(\frac{T}{k}\right)^2 \left(\frac{\omega_n \sin \frac{\pi\omega_n}{\omega}}{\omega + \omega \cos \frac{\pi\omega_n}{\omega}}\right)^2} \quad \ldots\ldots [43]$$

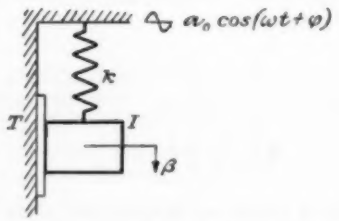


FIG. 15 FORCED VIBRATION WITH COULOMB DAMPING

The latter expression can be rewritten in the dimensionless form

$$\frac{\theta_0}{\alpha_0} = \sqrt{\left[\frac{1}{\left(\frac{\omega_n}{\omega}\right)^2 - 1}\right]^2 - \left(\frac{T}{I\omega^2\alpha_0}\right)^2 \left(\frac{\omega}{\omega_n} \cdot \frac{\sin \frac{\pi\omega_n}{\omega}}{1 + \cos \pi \frac{\omega_n}{\omega}}\right)^2} \quad [44]$$

giving the ratio between the relative amplitude θ_0 in the damper and the absolute amplitude α_0 of the hub. Since this ratio is of importance for the calculation of the stresses in the damper springs, the numerical results of [44] have been plotted in Fig. 14.

As was remarked before, the work absorption per cycle in the damper is $W = 4T\theta_0$, so that by means of [44] it is found that

[12] This problem has been treated in detail in a doctor's thesis presented in June, 1929, at the University of Pittsburgh under the title, "Forced Vibrations With Coulomb Damping."

$$\frac{W}{I\alpha^2\omega^2} = 4\left(\frac{T}{I\alpha_0\omega^2}\right)\sqrt{\left[\frac{1}{\left(\frac{\omega_n}{\omega}\right)^2-1}\right]^2-\left(\frac{T}{I\alpha_0\omega^2}\right)^2\left(\frac{\omega}{\omega_n}\cdot\frac{\sin\pi\omega_n/\omega}{1+\cos\pi\omega_n/\omega}\right)^2} \quad \ldots[45]$$

giving $W/I\alpha_0^2\omega^2$ as a function of $T/I\alpha_0\omega^2$ and of the parameter ω_n/ω. For the case of the damper without springs, $\omega_n = 0$, and it is immediately seen that [45] reduces to [25].

Fig. 12 shows the results of a numerical calculation of [45]. It is seen in this figure, as well as in Fig. 13, that the expressions [44] and [45] are only valid in a certain region, which calls for an explanation. Substitution of [42] and [43] in [40] gives an expression for θ determining the shape of the motion. Differentiating this expression gives the velocity $\dot{\theta}$, which should not become positive during the half-cycle $0 < \omega t < \pi$.

For values $\omega_n/\omega < 1$, i.e., below resonance with the damper, this condition is equivalent to

$$\frac{T}{I\omega^2\alpha_0} \leq \sqrt{\frac{\frac{\omega_n^2}{\omega^2}\left(1+\cos\pi\frac{\omega_n}{\omega}\right)^2}{\left(\frac{\omega_n^2}{\omega^2}-1\right)^2\left[\frac{\omega_n^2}{\omega^2}\left(1+\cos\pi\frac{\omega_n}{\omega}\right)^2+\sin^2\pi\frac{\omega_n}{\omega}\right]}} \quad \ldots\ldots[46]$$

This result is by no means evident, and it takes a good many pages of analysis to arrive at it. Since, however, it does not affect the main argument of the damper theory, reference should be made to the University of Pittsburgh thesis mentioned, which contains the derivations in detail.

For a damping torque greater than the one determined by [46] the damper will be gripped solid by the hub during part of its motion. No solution of the problem has been obtained for his case, except when there are no springs (see Appendix No. 1). However, such a solution would not be of great importance, since the maxima of the work curves (Fig. 12) occur in the region for which [45] is valid.

Applying these results to a complete Diesel-engine installation, we have to equate again the output energy [45] to the input energy [5]. Solving for α_0 and writing the result in a dimensionless form, we obtain finally:

$$\frac{I\omega^2\alpha_0}{\pi M_0} = \frac{\left(\frac{T}{\pi M_0}\right)^2\frac{\omega}{\omega_n}\cdot\frac{\sin\pi\omega_n/\omega}{1+\cos\pi\omega_n/\omega}}{\sqrt{\left(\frac{T}{\pi M_0}\right)^2\left[\frac{1}{(\omega_n/\omega)^2-1}\right]^2-\frac{1}{16}}} \quad \ldots\ldots[47]$$

giving the angle α_0 at the hub as a function of the friction torque T. It can be easily verified again that for the case of the unsprung damper ($\omega_n = 0$) this result reduces to [35].

Numerical calculations of [47] lead to the graph shown in Fig. 13.

Discussion

F. M. Lewis.[13] Methods of avoiding torsional criticals in reciprocating engines may be divided into three general classes:

(1) Those methods which depend upon adjusting the mass and elasticity of the system in such a way that the criticals will be outside the desired operating range.

(2) The vector cancellation methods; for instance, if equal flywheels are placed at each end of an engine, the major criticals will cancel out, since the impulses imparted to the system by the various cylinders considered as vectors will cancel. Then by the choice of a suitable firing order certain minors also may be cancelled. There are also other methods of accomplishing the same result.

(3) There are the damping methods, and these may be subdivided into those which utilize the inherent damping of the system and those which depend upon special devices, such as the Lanchester damper discussed in the paper.

The friction damper has found considerable favor among European engineers, but in this country it has been seldom used except in certain automobile engines. It has been the policy of the writer to avoid using the damper wherever it was possible to apply one of the other methods just mentioned. The objections to the device are based purely on practical considerations. The damper adds another complication to an already complicated machine. As yet no satisfactory method of design has been proposed. As the authors of this paper very aptly state the case, "dampers are not designed, but just built." Further, the damper is somewhat unreliable in action. If of the proper size and properly adjusted, it will suppress criticals, although it apparently does this by removing the sharp peak of the critical and spreading the vibration over a longer range. This spreading is in itself an undesirable feature. With time the adjustment of the damper will alter due to wear of the friction surfaces and the changing of the coefficient of friction of these. It is extremely difficult to determine whether a damper is functioning properly without tests which are beyond the skill of the ordinary mechanic or operator, and this makes its maintenance a difficult problem.

It is significant that although the engines of the *Graf Zeppelin* were fitted with friction dampers of this type, crankshaft failures occurred from torsional vibration. Although the operators of these engines were undoubtedly as skilled as could be reasonably expected, the existence of a dangerous condition was unsuspected until failures occurred. The writer does not wish to infer that the dampers were entirely at fault in this case, however, as other causes also operated.

There will remain cases, however, where it is impossible to suppress criticals in any other way than by a damper, and it would be extremely useful to have a correct theory of damper design which could be applied to these cases. The theory of the damper where viscous rather than Coulomb friction is the damping medium has been very ably presented by Gümbel in Z.V.D.I. and by Carter in the reports of the British Aeronautical Research Committee. A theory dependent on viscous friction, however, cannot be entirely satisfactory, as the action of Coulomb friction must be considerably different. The authors have attempted to supply this need. The paper they present is extremely interesting and thought-provoking.

With regard to the experiments made to demonstrate the action of Coulomb friction, the writer would inquire on one point—that is, whether there was any indication of static friction being greater than sliding friction, or, in other words, was there any indication of initial sticking with the surfaces tested?

The problem which the authors have attacked is one of great difficulty. Where non-linear relations are involved, the theory of vibration, like most physical theories, becomes considerably involved. In the present case one must deal with a damping force which is a discontinuous function of the velocity.

The writer must confess frankly that he does not find the theory as presented by the authors entirely satisfactory. Much of the theory is correct, but the results obtained are rather misleading, and it is difficult to see how these results can be applied to design without much more information than is here presented.

One of the points which gave much difficulty was to determine what the authors meant by the expression "resonant frequency" and its symbol ω. After some correspondence with them on

[13] Professor of Engineering, Webb Institute of Naval Architecture New York, N. Y. Mem. A.S.M.E.

this point, the writer was advised that they mean "frequency of maximum amplitude," but it is thought that a more correct statement might be "frequency at which the exciting moment is 90 deg. in phase to the engine vibration." It should be pointed out that neither one of these definitions is the usual one. This is a point of considerable importance, for when one comes to the authors' results he finds that they are expressed in terms of the dimensionless variable $I\omega^2\alpha/\pi M_o$. Using the authors' definition of ω, this variable contains two unknowns, α and ω, and it is impossible to solve for α without knowing ω. Further, ω will vary over a very wide range with different sizes and adjustments of dampers.

In correspondence the authors state that in design work ω is known from experimental results, but the paper gives no hint of this. It should be pointed out that, in any complete solution of the problem, ω must be expressed as a function of four dimensionless variables. These variables would be:

The ratio of the forcing frequency to the original natural frequency of the system;

The ratio of the damper mass to the engine mass;

A function of the engine friction; and

The ratio of the damper friction to the exciting forces, or T/M.

It is in this manner that the results of Carter for the viscous-friction damper are expressed. One great difficulty of the problem is that of representing α as a function of four variables.

In discussing the infinite damper the authors show that the solution is indeterminate provided that nothing but Coulomb damping is present. It may be shown likewise for the finite damper that this is true where the frequency is equal to the original natural frequency of the system. At some frequency lower than this the relation stated by the authors will be approximately correct, but the authors do not tell us what that frequency is.

The same difficulties are present in the theory of the damper with springs to an even greater extent, and one cannot consider the theory as presented as being at all satisfactory. The problem is one of great difficulty. For a complete solution, five instead of four variables would be needed, an additional one being added for the effect of the damper springs. The effect of adding the spring-connected flywheel is to profoundly alter the original elastic system. Instead of a single original critical speed, there now would be two critical speeds—one above and one below the original critical speed. It is at these new critical points where one is interested in the amplitude of vibration. Provided only Coulomb damping is present, an indeterminate solution is obtained; for the energy absorbed, being a linear function of the amplitude, and the energy put into the system being likewise a linear function, the two cannot, in general, be made equal.

One assumption made in the theory of the spring damper seems highly questionable. The relative motion of the damper and engine masses is determined, assuming a forced periodic motion of the engine masses. In doing this the damper friction T is taken into account. It would seem essential that if the effect of T upon the damper mass is to be taken into account, its effect upon the engine masses should be likewise considered. If this friction T is considered, the harmonic moment M will not produce a harmonic motion in the engine masses, and the energy absorption will therefore not be $\pi M_o\alpha_o$. This may help to explain the fact that the authors obtain a determinate solution for a problem which is essentially indeterminate.

The same difficulty as regards ω applies here as to the damper without springs; that is, results are expressed in the form $I\omega^2\alpha/\pi M_o$, and α cannot be solved for until ω is known, and no hint is given as to how ω may be obtained. This value of ω will vary even more widely than it did with the simple friction damper.

The same objection might be applied to the theory of the simple friction damper; that is, the effect of T upon the damper mass is considered, but its effect upon the engine mass is neglected. The error in this is involved in the assumption of sinusoidal engine vibration and is contained in Assumption 1 on the first page.

J. J. WYDLER.[14] This paper is a fine contribution to the art, and will be welcomed by engineers as a valuable help to a rational solution of the problems to which it relates.

As to the usefulness of dampers in general, the writer would say that attempts have been made in many instances to develop damping apparatus which would make a power plant workable over the whole operating range. Only in particular cases have heavy low-speed engine sets been fitted with special dampers. In general, such dampers have been most widely applied in the form of hydraulic or oil couplings, particularly in marine plants in which oil engines have been used to drive propeller shafts through the medium of reduction gears. More recently such couplings were installed in the power plants of the newest giant Junkers land airplane for passengers.

The greatest use to which such dampers have been put is in the case of high-speed multi-cylinder internal-combustion engines, especially where these are made a part of some of the higher-priced automobile assemblies. One of the most interesting attempts to make use of the damping principle is in connection with the propelling plants of dirigibles.

There is, in fact, a special need for dampers where engines are used to supply power for locomotion, because, primarily, such engines, if they cover the best range of usefulness, need to operate at all speeds unhampered by any critical speed range. However, at best, but small use has been made of damping equipment.

Probably the principal reason why dampers have not been more generally utilized is that they introduce an additional mechanism and a complication to the plant, and one requiring more or less careful attention. A damper absorbs energy by transforming it into the heat of friction which has to be dissipated. In smaller engine sets air cooling may suffice, but in the case of heavier installations water cooling may become necessary. It would be interesting to have the authors tell how long they have found the materials making up the friction surfaces to last in dampers they have used under normal operating conditions.

Another reason why dampers have not been more generally used may be (as indicated in the paper) that a damper cannot be entirely relied upon to protect the power plant if its engine has more than one critical speed range or if the critical speed range for some reason shifts from that of the original installation. A most serious failure of the latter sort resulted in the breakdown of four propeller shafts of the *Graf Zeppelin*. In this case, before starting the trip, a little too much tension was put upon the springs of the couplings between the engines and the propellers. As a result of this, the critical speed ranges were shifted enough to render the dampers ineffective.[15]

In the opinion of the writer the specialists who have to deal with vibration troubles in multi-cylinder plants should not lose sight of opportunities to balance the inciting impulses through choosing proper crank angles. Of course, in the case of six-cylinder sets, little can be accomplished along this line. However, a ten-cylinder set could be built up by a combination of a six- and a four-cylinder set, each one in itself satisfactorily balanced as to the reciprocating mass forces, with the combination properly balanced as to tangential harmonic impulses.

When once torsional vibrations have been suppressed or eliminated in this way, it may be said that this problem has been solved as satisfactorily as the balancing of reciprocating inertia forces.

[14] Combustion Utilities Corporation, New York, N. Y.

[15] Thoma, *V.D.I.*, September 28, 1929, p. 1387.

Until we have learned to render critical speeds harmless by balancing the inciting impulses, we should welcome every means, such as dampers, which aid in the suppression of heavy vibrations.

J. Dickson.[16] The authors are due considerable praise. This paper is probably the first of its kind, judging from the lack of printed matter on this subject. This lack in publications, however, is no argument that "dampers are just made, not designed."

On the fourth page the writer is credited with attempting to overcome a difficulty which later on has increased to difficulties. Equation [8] provides no means of proportioning the damper flywheels in advance. True, but the same in a sense could be said of Equation [13]. The designer must allow for inconsistencies in the coefficient of friction in a practical damper, under service conditions. Suppose the springs are designed on a 15 per cent base as recommended and 30 per cent is obtained, the value of T/I is doubled, and the diagram, Fig. 6 (a), of the continuously slipping flywheel cannot be constructed. Then, again, it is doubtful whether the conditions as represented in Fig. 6(a), on which the theory is based, do exist. Energy is taken from the vibrating shaft by dissipation, as agreed on in the second assumption, also by giving kinetic energy to the damper flywheel (this energy the designer in using Equation [8] corrects for and thus eliminates all the other difficulties mentioned). The greater the ratio T/I, the greater is this latter proportion of the total energy.

At time $t_1/4$ [see Fig. 6(a)] the kinetic energy of the flywheel reaches a maximum and commences to dissipate its energy by exerting on the shaft a torque T, causing a change in velocity either by superimposing on the periodic vibration an increase in speed to the shafting system as a whole, or, as is more probable, by setting up in the shaft another vibration due to its position on the elastic curve, its magnitude, and periodic nature.

Summing up, the writer does not consider that the shaded portion of Fig. 6(a) on which the formulas are based is a true representation of the energy dissipated, and designers should be wary, provided they accept Fig. 6(a), of proportioning the damper flywheels on the optimum values given.

R. G. Gage.[17] Although the chief value of the paper undoubtedly will be for those particularly concerned in the design of dampers, there is definite information which is available for application by the operating engineer, and it is on this alone that this comment is made.

The authors' early statement that "dampers are just made, not designed," is at first frightening, and one wonders just what that may mean. Knowing that dampers of the type under discussion have been in successful operation on Beardmore six-cylinder engines for over two years, one wonders if this has been merely a matter of luck.

It is stated that the damper efficiency is modified by

(*a*) The condition of the surfaces of the friction plates (the coefficient of friction varying from 50 per cent to 15 per cent)
(*b*) The damper-spring tension
(*c*) The temperature of the damper.

In service, the feroda friction plates are found to be covered with oil, and it is best that this is so. The surface wear is thus reduced, and the coefficient of friction, although at its lowest value, is more constant. The spring tension does not vary appreciably between settings. The damper temperature is usually within moderate limits, but when under extreme conditions of cold, the engine is known to be rough at its critical speed.

The secret of the success of the present dampers is probably due to the fact, as stated in the paper, that spring tension greater than that necessary to give the optimum flywheel torque has little effect on the efficacy of the damper. Positive stops between the flywheel halves, if amply designed, guarantee sufficient spring loading. Recognizing the possibility of oil on the surfaces of the friction plates, the designer probably worked on the lowest value of the coefficient of friction, given as 15 per cent.

In what way would a damper which was "designed" rather than "made" be an improvement in operation?

George J. Dashefsky.[18] The New York Navy Yard some years ago had occasion to experiment with vibration dampers of the solid-friction type as applied to the Diesel-engine propulsion plants of submarines. The efficacy in damping a particular mode of vibration was soon observed to depend upon the adjustment of the pressure between the friction surfaces. The main difficulty experienced lay in the glazing of the friction surfaces, with consequent loss in effective damping.

In a recent problem with camshaft vibration of a 3000-hp. submarine Diesel engine, a vibration damper developed at the Navy Yard, utilizing viscous friction, was successfully applied as a solution. This type of damper made its appeal because of its mechanical simplicity.

An investigation made in connection with the development of this viscous-friction damper, in some respects, covered common ground with matter presented in the paper. A description of the experiments and findings may be of interest.

The viscous-friction damper comprises essentially a fluid-tight drum, built upon a hub, and a flywheel or inertia member residing within the casing and free to oscillate upon the hub. A gap of determined size is left between the casing and the sides and the periphery of the inertia member. The casing is filled with a viscous medium. The damper is secured to the vibrating member by means of the hub. Vibration imposed upon the casing of the damper results in relative motion between the casing and inertia member; distortion of the viscous medium takes place, with consequent absorption of vibration energy.

The characteristics of the viscous-friction damper were analyzed mathematically and its predicted damping characteristics checked with fair accuracy by an experimental determination. The "life-size" damper was mounted upon a steel rod and suspended by clamping the free end of the rod in an appropriate bracket so that the axis of the vibration system was in a vertical plane. The stimulating torque was obtained by mounting an adjustable unbalanced mass upon the upper side of the damper. The mass was rotated by an electric motor, the speed of which determined the frequency of the stimulating torque. Simultaneous records of the motion of both the casing (oscillating member) and the flywheel (inertia member) were recorded by an especially adapted torsiograph.

Theory indicates that if the motion of the casing be a simple harmonic, the motion impressed on the inertia member should also be a simple harmonic of the same frequency. This was substantiated by the experiments mentioned. A similar verification of the motion of the flywheel member of the solid-friction damper would be of interest and would possibly show experimentally if, and where, in the cycle of vibration, slipping between the flywheel hub ceases.

The authors have indicated that the solid-friction damper shows a critical point of maximum damping ability. This point depends upon a relation of the "slipping" torque of the damper to the harmonic disturbing torque. In the case of the viscous-friction damper, the friction torque is a function of the coefficient of

[16] William Beardmore Company, Ltd., Glasgow, Scotland.
[17] Office of Chief Electrical Engineer, Canadian National Railways, Montreal, Canada.
[18] Vibration Expert, Navy Yard, New York, N. Y.

viscosity of the working fluid, which coefficient is dependent upon temperature. The curves of damping energy absorption with respect to temperature, as indicated by theory and experiment, are similar to those indicated for the solid-friction damper.

In order to spread the range of effectiveness of the viscous friction damper, it has been proposed to divide the damper in two, utilizing two flywheels of different moments of inertia. By properly proportioning the flywheels, two different critical temperatures result, and a broader range of effective damping for the damper as a whole.

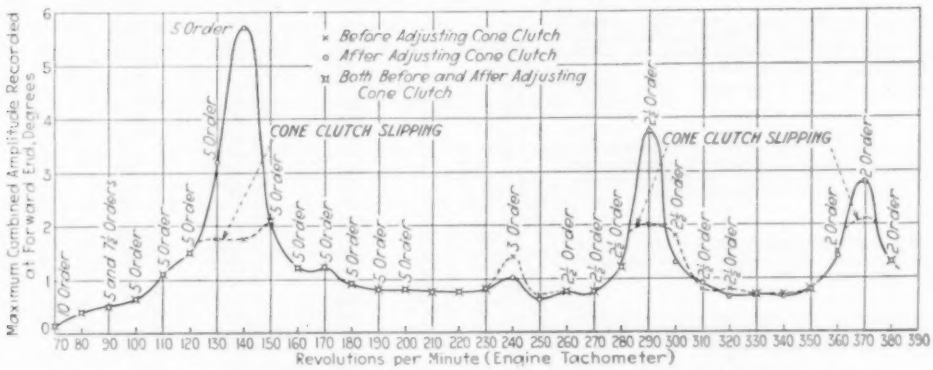


FIG. 15 RECORDED VIBRATIONS FROM RESULTS OF TORSIOGRAPHIC TESTS ON ENGINES OF U. S. SUBMARINES *V-5* AND *V-6*
(See Installation No. 6 in Table 1.)

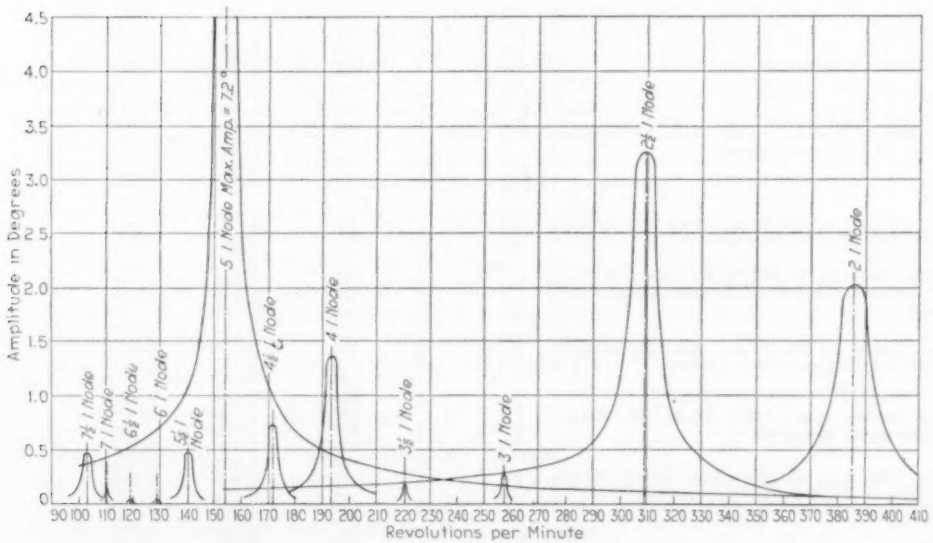


FIG. 16 CALCULATED RESONANCE CURVES, TESTS ON ENGINES OF U. S. SUBMARINES *V-5* AND *V-6*, FOR COMPONENT AMPLITUDES OF VIBRATION AT AIR COMPRESSOR AND FOR CHARGING CONDITION
(All orders given are due to the stimulation at the main cylinders except the fourth order, which is due to the stimulation at the air compressor, and the second order, which is due to the combined effect of main cylinders and alternating cylinders. See Installation No. 6 in Table 1.)

In the case of the solid-friction damper the effectiveness of the damper depends upon a relation between the "slipping" torque of the damper and the torque which stimulates the vibration. (See Equation [14] of the paper.) Where more than one synchronous speed occurs within the operating range of an installation, the adjustment of the solid-friction damper may be such as to be relatively ineffective for some of the critical speeds. In such a case the device suggested for broadening the effective range of the viscous friction damper by employing two flywheels may be utilized. The flywheels and "slipping" torques should be so proportioned as to give a compromise of combined damping, effective on all of the synchronous vibrations encountered.

FREDERIC P. PORTER.[19] The authors have referred in several instances to the law of hysteresis damping. They appear to recognize this law only in a qualitative sense. On the sixth page the authors state that "the largest natural damping existing in a system, as shown in Fig. 5, is due to the metallic hysteresis of the shaft material under varying stresses." Yet shortly after this they say that "in real systems where hysteresis damping acts by itself, the amplitude of motion at resonance is usually so large that the maximum stresses in the system are far above the fatigue limit."

Now, the resonant condition referred to here occurs each time the ratio of a natural frequency of the system to the frequency of a stimulating impulse is equal to unity. In four-cycle engines this will occur whenever the ratio of the natural frequency in v.p.m. to the shaft speed in r.p.m. is one of the series of numbers ½, 1, 1½, 2, 2½, 3, etc., and in two-cycle engines when this ratio is one of the series 1, 2, 3, 4, 5, etc. If a four-cycle installation designed to operate at 300 r.p.m. had a natural frequency of 2000 v.p.m., resonance would occur near 300 r.p.m., as follows:

Order	R.p.m.
6	333
6½	308
7	286
7½	267
8	250

Apparently the authors believe that at each of these speeds a vibration will occur that will exceed the fatigue limit of the shaft material.

Lewis, in a paper mentioned by the authors, proposed using a law for the hysteresis damping in engine installation that he determined from experimental results of Rowett. Following this proposal by Lewis, the writer modified this law by an empirical factor which he determined from the result of measuring the vibrations of actual installations with a torsiograph. This line of investigation led the writer to the formula for hysteresis loss or, which would be more correct, for the overall damping loss in engine installations whose synchronous vibrations are not limited by some attached source of damping such as a marine propeller or friction damper, etc., as follows:

$$K = 4.5 \times 10^{-10} S^{2.3}$$

[19] Fairbanks Morse Company, Beloit, Wis.

where K is the energy (lb-in.) lost per stress cycle per cubic unit (in.^{3}) of steel.

The authors object to this procedure by saying that it cannot be done, even though they admit that the largest natural damping existing in certain installations is due to metallic hysteresis. The writer has used his overall "hysteresis" damping formula for several years and has applied it to many installations that have been tested by him. In all cases he has found a fair agreement between the calculated and recorded amplitudes on actual installations where such conditions as slipping clutches, clearance between connected members of the shaft line-up, etc., have not partially or totally destroyed the resonant characteristics. The procedure of such calculations was described in his previous paper mentioned by the authors, and follows the general scheme outlined by Lewis.

TABLE 1 COMPARISON OF CALCULATED AND RECORDED VIBRATIONS

		Order of vibration	Frequency	R.p.m.	Recorded amplitudes, deg.	Calculated amplitudes, deg.
(1)	Four-cycle, six-cylinder, 1200-i.hp. at 150 r.p.m.; bore, 22; stroke, 32; firing order, 1–4–2–6–3–5; U. S. Army dredges, Dan C. King class, original installation. (Described in A.S.M.E., 1928, Paper No. APM-50-8.)	9	1	102	0.53	0.68
		6	1	156	1.80	1.86
(2)	Four-cycle, 8-cylinder, 600-b.hp. at 380 r.p.m.; bore, 13.5; stroke, 15; firing order, 1–5–7–3–8–4–2–6; Nelseco Diesel engine installation; U. S. submarines *S-18* to *S-41*, surface condition. (Described in A.S.M.E., 1928, Paper No. APM-50-8.)	8	2	241	0.99	1.000
		6½	2	295	0.366	0.580
		5½	2	350	1.002	0.885
(3)	Installation nearly similar to (2); U. S. submarine *S-44*; surface condition	8	2	238	1.0	1.000
		5½	2	342	0.9	0.885
	Charging condition	5½	1	338	1.00	0.885
(4)	Four-cycle, ten-cylinder, 2350-b.hp. at 345 r.p.m.; bore and stroke, 20⅞; port firing order, 1–7–3–9–5–10–4–8–2–6; bureau type engine; U. S. submarine *T-3*; port engine; surface condition. (Described in A.S.M.E., 1928, Paper No. APM-50-8.)	4	2	307	0.264	0.458
		4½	2	273	0.298	0.428
		5	2	246	1.372	2.060
		5½	3	334	0.274	0.563
		7½	3	246	0.175	0.591
(5)	Installation similar to (4); starboard firing order, 1–6–2–8–4–10–5–9–3–7; U. S. Submarine *T-3*; starboard engine; surface condition. (Described in A.S.M.E., 1928, Paper No. APM-50-8.)	4	2	311	0.311	0.440
		4½	2	268	0.218	0.428
		5	2	250	0.935[a]	2.060
		5½	2	230	0.167	0.271
		5½	3	340	0.274	0.563
		7½	3	250	0.248	0.591
(6)	Installation using engine similar to (4); shaft line-up entirely different; U. S. submarines *V-5* and *V-6*; charging condition; preliminary installation tested at New York Navy Yard.	5	1	140	5.70	7.200
		2½	1	290	3.50	3.250
		2	1	270	2.40	2.040
(7)	Four-cycle, six-cylinder, 1000-b.hp. at 390 r.p.m.; bore, 17¾; stroke, 16⅝; firing order, 1–5–3–6–2–4; bureau type engines; U. S. submarines *V-1* to *V-3*; forward auxiliary engines.	6	1	388	1.30	1.512
		7½	1	310	0.31	0.323
		9	1	258	0.67	0.549
(8)	Two-cycle, six-cylinder, 2250-b.hp. at 310 r.p.m.; bore, 21⅝; stroke, 21⅝; firing order, 1–5–3–4–2–6; Busch-Sulzer engines; U. S. submarines *V-1* to *V-3*; main engines; charging condition (propeller disconnected).	6	1	140	1.62	1.660
		9	2	233	0.43	0.570
(9)	Same installation as (8), with 7080 lb-ft.^{2} inertia removed from flywheel, charging condition.	6	1	145	1.97	1.950
		9	2	241	0.52	0.570
(10)	Installation of (9) in surface condition.	6	2	150	2.10	1.950
		9	3	242	0.33	0.570
(11)	Two-cycle, six-cylinder, 4000-b.hp. at 125 r.p.m.; bore, 29; stroke, 44; firing order 1–5–3–4–2–6; Nordberg engine.	6	1	137.5	2.50	2.560
		7	1	118.0	0.14	0.160
		8	1	103.0	0.20	0.280
		9	1	91.7	0.44	0.450
		10	1	82.5	0.20	0.180
		11	1	75.0	0.08	0.060
		12	1	68.8	0.33	0.590
(12)	Four-cycle, six-cylinder, 480-b.hp. at 257 r.p.m.; bore, 15; stroke, 20; firing order, 1–5–3–6–2–4; Rathbun-Jones engine.	6	1	317	1.45	1.405
		7½	1	252	0.25	0.360
		9	1	210	0.30	0.472
(13)	Four-cycle, eight-cylinder, 360-b.hp. at 275 r.p.m.; bore, 12^{15}⁄$_{16}$; stroke, 18; firing order, 1–5–2–7–4–8–3–6; U. S. Army dredge *Willets Point*; Winton engine.	3½	1	310	2.100	1.860
		4	1	272	4.100	5.920
		6	1	181	1.640	1.300
		8	1	136	1.310	1.173
(14)	Four-cycle, six-cylinder, 300-hp. at 600 r.p.m.; bore, 10; stroke, 12; firing order, 1–5–3–6–2–4; Ingersoll-Rand engine.	6	1	620	0.703	0.738
		7½	1	496	0.180	0.223
		9	1	413	0.190	0.278
		12	1	310	0.120	0.142
(15)	Four-cycle, six-cylinder, 700-hp.; bore, 14¾; stroke, 16; firing order, 1–5–3–6–2–4; Ingersoll-Rand engine.	4½	1	538	0.788	0.665
		6	1	403	1.000	0.801
		7½	1	324	0.150	0.198
		9	1	269	0.220	0.302
(16)	Four-cycle, six-cylinder, 1400-b.hp. at 345 r.p.m.; bore, 20⅞; stroke, 20⅞; firing order, 1–5–3–6–2–4; U. S. submarine *V-4*; bureau type engine; charging condition.	4½	1	296	0.260	0.287
		6	1	217	0.800	0.933
		9	1	146	0.270	0.325
		9	2	339	0.200	0.223
(17)	Installation of (16) in surface condition.	4½	2	295	0.240	0.278
		6	2	217	0.700	0.914
		9	2	145	0.260	0.376
		9	3	339	0.200	0.204
(18)	Four-cycle, six-cylinder, 550-hp. at 450 r.p.m.; bore, 13.78; stroke, 13.78; installation described by Wydler.	6	1	390	1.130	1.190
(19)	Four-cycle, six-cylinder 1200-hp. at 450 r.p.m.; bore, 17.75; stroke, 16.56; installation described by Wydler.	6	1	358	1.190	1.530

[a] Clutch found to slip at 0.9 deg. amp.

The authors have referred to Föppl's laboratory experiments in this regard. It would be interesting if they would make an actual numerical comparison.

Table 1 gives the results obtained on some of the installations investigated. Installations No. 4 and 5 had been in operation for several years before the torsiographic test was made upon them. It is reasonable to expect, therefore, that certain parts, such as the hub of the motor armature upon the shaft, the couplings, the hub under the clutch, etc., had loosened slightly during the years of service, thus destroying to some extent the synchronous vibrations which would otherwise result.

Figs. 15 and 16 show the recorded and calculated results obtained for installation No. 6. In Fig. 15 the maximum recorded vibrations are plotted upon the speeds recorded by the engine tachometer. This was done to speed up the work of the investigation. When this tachometer was later calibrated, it was found to be reading a little low, particularly at low speeds. In the first series of tests no amplitudes were found greater than about 2 deg. Upon investigation it was found that the cone friction clutch between the engine and motors was slipping. This clutch was adjusted tighter, and the test was repeated. This time the amplitudes increased to the values shown in Fig. 15. Even with this adjustment the clutch was found to slip in way of the large five-order vibration. Since this vibration was recorded to an amplitude of 5.7 deg., however, it can be concluded that the clutch did not slip for those vibrations whose amplitudes are less than this amount.

Installations (18) and (19) were described by Wydler in "Drehschwingungen in Kolbenmaschinenanlagen," Berlin, 1922. The horsepower values for these installations are those given by Wydler and are in the metric units. Wydler also gives results for a third installation which has a friction clutch between the engine and generator. On account of the uncertainty introduced by the possibility of this clutch slipping, a comparison of the calculated and recorded amplitudes for this installation is considered questionable.

The foregoing data, the writer believes, justify the use of his partly empirical formula for the determination of the maximum amplitudes of Diesel-engine installations, at least until some better method is proposed.

On the third page the authors state that "in nearly all practical cases of torsional resonance in gas or Diesel engines it is only the lowest natural frequency of the system which is dangerous." While it is true that in many installations the lowest natural frequency is the only one to cause trouble, the application of Diesel engines to the generation of power is so diversified that many cases also arise in practice where the second or third natural frequency is also objectionable. Table 1 will illustrate this fact.

The authors make a statement on the fifth page to the effect that a safe design factor for vibratory stresses is 10 or 12 per cent of the ultimate tensile strength, since "in most steels the ultimate strength in torsion is about half the ultimate strength in tension, and that the fatigue strength in torsion is again about half the ultimate strength in torsion." They apparently are a little mixed on the ideas of ultimate strength and endurance limit.[20]

Author's Closure

Professor Lewis agrees with the authors that "it would be extremely useful to have a correct theory of damper design," but finds that the theory as presented is not entirely satisfactory. The main points in his objections are the facts that (1) the distortion of the harmonic motion of the engine mass due to the damper torque is neglected, (2) the value of the resonant frequency ω is not known, and (3) the ratio of the damper mass to the engine mass is not considered in the solution.

The authors are well aware that the solution presented is an approximate one. They were and are of the opinion that the approximation is very close, but in the absence of an exact solution the degree of approximation naturally cannot be calculated. In order to reply satisfactorily to Professor Lewis' criticism, the authors have developed an exact solution of the problem since the presentation of the paper in December, which meets the three objections mentioned. The system considered is that of Fig. 5 with the exception that the inertia of the main flywheel I_2 has been made infinite (Fig. 17). In practical cases I_2 is often ten or more times larger than I_1 so that this is justified, especially since the exact solution to be given now only serves the purpose of giving a comparison with the approximate solution of the paper.

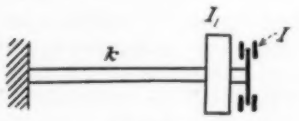


Fig. 17 Scheme For Exact Calculation

Let the angular deflections of I_1 and I be α and γ, respectively, and let the engine torque acting on I_1 be $M_0 \cos(\omega t + \varphi)$. We restrict ourselves to the case of continuous slippage [cf. Fig. 6(*a*)]. During the half-cycle that $0 < \omega t < \pi$, the following differential equations describe the motion:

$$\left.\begin{aligned} I_1\ddot{\alpha} + k\alpha - T &= M_0 \cos(\omega t + \varphi) \\ I\ddot{\gamma} + T &= 0 \end{aligned}\right\} \quad \ldots\ldots\ldots [48]$$

in which φ is a phase angle to be determined later. The general solutions of [48] are:

$$\left.\begin{aligned} \alpha &= C_1 \sin \omega_n t + C_2 \cos \omega_n t \\ &\quad + \frac{T}{k} + \frac{M_0}{k} \cdot \frac{\omega_n^2}{\omega_n^2 - \omega^2} \cdot \cos(\omega t + \varphi) \\ \gamma &= -\frac{T}{2I} t^2 + C_3 t + C_4 \end{aligned}\right\} \quad \ldots\ldots [49]$$

where

$$\omega_n = \sqrt{k/I_1}$$

In formulating the boundary conditions, the time $t = 0$ is taken at the instant that $\gamma - \alpha$ is a minimum. At $t = 0$ the relative motion just reverses, and between $t = 0$ and $t = \pi/\omega$ the relative motion has always the same direction, so that $\dot{\gamma} - \dot{\alpha} > 0$ and [48] holds. The eight boundary conditions are:

$$\left.\begin{aligned} t = 0 \quad \alpha &= \alpha_0 \quad \dot{\alpha} = \dot{\alpha}_0 \quad \gamma - \alpha = -\theta_0 \quad \dot{\gamma} - \dot{\alpha} = 0 \\ t = \pi/\omega \quad \alpha &= -\alpha_0 \quad \dot{\alpha} = -\dot{\alpha}_0 \quad \gamma - \alpha = +\theta_0 \quad \dot{\gamma} - \dot{\alpha} = 0 \end{aligned}\right\} [50]$$

expressing the fact that after half a cycle the deflections and velocities must be opposite to those at $t = 0$, which follows from the symmetry of the system under consideration.

It is noted that Θ_0 is the maximum value which $\gamma - \alpha$ reaches during a cycle, but that α_0 is not the maximum of α during the cycle. The eight conditions [50] will be used to determine the four integration constants and the four parameters:

$$C_1,\ C_2,\ C_3,\ C_4,\ \Theta_0,\ \alpha_0,\ \dot{\alpha}_0, \text{ and } \varphi$$

Substitution of [50] into [49] after some lengthy calculations leads to the solution:

$$\left.\begin{aligned} \cos\varphi &= \frac{k\theta_0}{M_0 V} \qquad \sin\varphi = -\frac{T}{M_0 V} \cdot \left\{ U + \frac{\pi}{2} \cdot \frac{I_1}{I} \cdot \frac{\omega_n^2}{\omega^2} \right\} \\ \theta_0 &= \frac{M_0}{k} \cdot \sqrt{V^2 - \left(\frac{T}{M_0}\right)^2 \cdot \left\{ U + \frac{\pi}{2} \cdot \frac{I_1}{I} \cdot \frac{\omega_n^2}{\omega^2} \right\}^2} \end{aligned}\right\} . [51]$$

where the abbreviations U and V have the following meaning:[21]

$$U = \frac{\dfrac{\omega_n}{\omega} \cdot \sin \pi \dfrac{\omega_n}{\omega}}{1 + \cos \pi \dfrac{\omega_n}{\omega}} \; ; \qquad V = \frac{\omega_n^2}{\omega_n^2 - \omega^2}$$

The motion of the engine also is calculated from [49] and [50]:

[20] H. E. Moore and J. B. Kommers, "Fatigue of Metals."

[21] *Phil. Mag.*, 1930, p. 801.

$$\alpha = -\frac{TU\omega}{k\omega_n}\cdot\sin\omega_n t + \frac{T}{M_0}(1-\cos\omega_n t) + \frac{M_0 V}{k}\cdot\cos(\omega t+\varphi)\ldots[52]$$

The maximum value of α during one cycle in which we are primarily interested happens at some time t_0 between $t = 0$ and $t = \pi/\omega$. A direct analytical calculation of this t_0 by differentiation of [52] is not possible, since [52] is transcendental because ω and ω_n are incommensurable. However, for the cases in hand ω is close to ω_n, so that a first approximation for t_0 can be obtained by putting $\omega = \omega_n$. Then α is calculated for this t_0 and for a neighboring value on each side of t_0; the maximum being determined by graphical interpolation.

The maximum thus obtained is yet a function of ω. As such it will reach a maximum or "come to resonance" for some ω smaller than ω_n. This ω cannot be calculated directly, but a first approximation is found by assuming that α becomes maximum at the same frequency that the *relative* motion θ_0 becomes a maximum. This latter frequency can be calculated from [51] by successive approximations.

The numerical calculations are extremely laborious, though several short cuts can be applied.

The solution thus obtained is valid only when no stops occur during the motion. It can be shown that this condition can be expressed as:

$$\frac{T}{M_0}\left(1+\frac{I_1}{I}\right) \leq \frac{V\omega^2}{\omega_n^2}\cdot\cos\varphi\ldots\ldots\ldots\ldots[53]$$

It can also be shown that the solution [51] reduces to the solution given in the paper when $I/I_1 = 0$. Therefore *the solution given in the paper is exact for an infinitely small damper mass.* The results of the numerical calculations are given in Table 2 below and also in Figs. 18 and 19.

TABLE 2

$I/I_1 = 0$			$I/I_1 = 0.1$		
$\frac{T}{M}$	$\frac{\omega_n}{\omega}$	$\frac{I\omega^2\alpha}{M_0}$	$\frac{T}{M}$	$\frac{\omega_n}{\omega}$	$\frac{I\omega^2\alpha}{M_0}$
0.8	1	6.60	0.8	1.0014	6.74
0.9	1	2.89	0.9	1.0096	2.95
1.0	1	2.54	1.0	1.0156	2.58
1.1	1	2.47	1.1	1.020	2.52
1.2	1	2.49	1.2	1.023	2.54
1.3	1	2.56	1.3	1.025	2.61
1.4	1	2.66	1.39	1.027	2.69
1.45	1	2.71			
$I/I_1 = 0.2$			$I/I_1 = 1.0$		
$\frac{T}{M}$	$\frac{\omega_n}{\omega}$	$\frac{I\omega^2\alpha}{M_0}$	$\frac{T}{M}$	$\frac{\omega_n}{\omega}$	$\frac{I\omega^2\alpha}{M_0}$
0.8	1.0028	6.87	0.8	1.0124	7.88
0.9	1.0190	3.00	0.9	1.089	3.34
1.0	1.031	2.63	1.0	1.145	2.90
1.1	1.039	2.56	1.03	.....	....
1.2	1.046	2.58	...	.....	....
1.3	1.050	2.66	...	.....	....
1.327	1.051	2.68	...	.....	....

These two figures give sufficient information to answer Professor Lewis' objections. It is noted that in Fig. 18, the minimum point for $I/I_1 = 0.2$ is $3\frac{1}{2}$ per cent in error with respect to the solution given in the paper. A damper with such a ratio is a rather heavy one, so that it can be stated that *for all practical dampers the solution given in the paper is less than 3.5 per cent in error.* The value of the friction torque at which the minimum occurs also agrees with the approximate value.

Fig. 19 shows the "resonant frequency" ω as a function of the friction torque. The ordinates are presented in a dimensionless form, ω_l being the natural frequency of the system with the damper flywheel completely loose, while ω_t is the natural frequency with the damper completely tight.

It is seen that the minimum amplitude occurs about at:

$$\frac{\omega-\omega_l}{\omega_t-\omega_l} = 0.6\ldots\ldots\ldots\ldots\ldots\ldots[54]$$

for all practical damper sizes. In a heavy damper, $I/I_1 = 0.2$, for instance, [54] gives:

$$\frac{\omega}{\omega_l} = 0.965$$

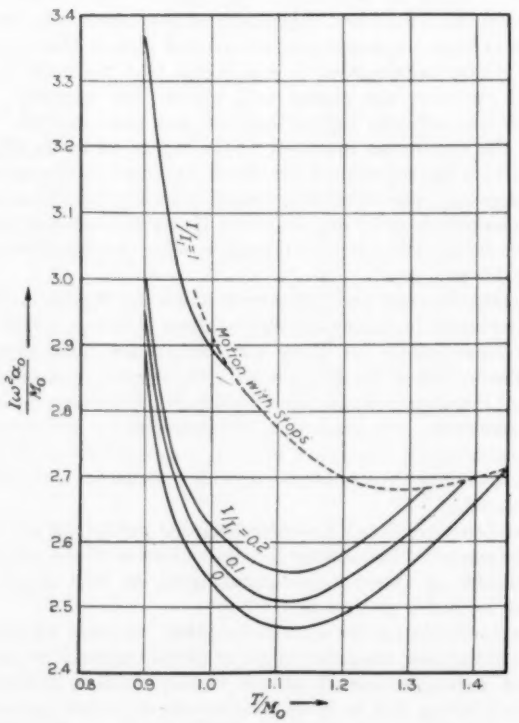


FIG. 18 AMPLITUDES BY EXACT CALCULATION

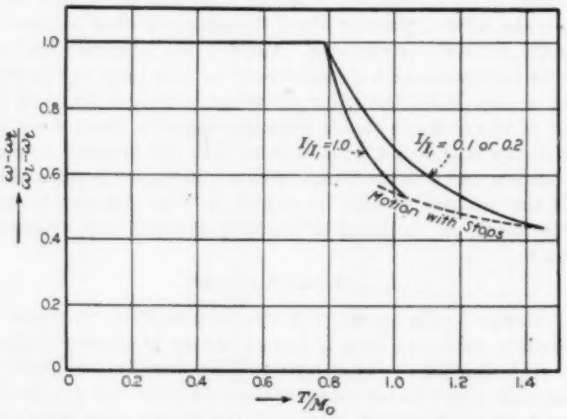


FIG. 19 RESONANT FREQUENCIES BY EXACT CALCULATION

If in formula [13] the original natural frequency ω_l were taken instead of the resonant frequency ω, the result $\alpha_{\min}$ comes out 7 per cent large. But the error due to the finite damper mass is 3.5 per cent in the other direction. The total error committed therefore is 3.5 per cent, which is very satisfactory for an amplitude calculation.

In addition to this exact theory a series of careful tests was carried out on a new model, which had no flexibility between I_1 and I. The torque input was accomplished mechanically by means of a coil spring (Fig. 20). The flywheels were *not* revolving; the driving motor merely twisting one end of the coil spring back and forth through an angle of 30 deg. by means of an eccentric on its shaft.

The recording was done by means of a mirror $^1/_2$ in. in diameter suspended in a universal suspension. A point source of light was reflected by this mirror and thrown on a photographic plate, while the entire room was in the dark. The mirror was moved back and forth about its vertical axis by means of a wire connected to that end of the driving spring which was

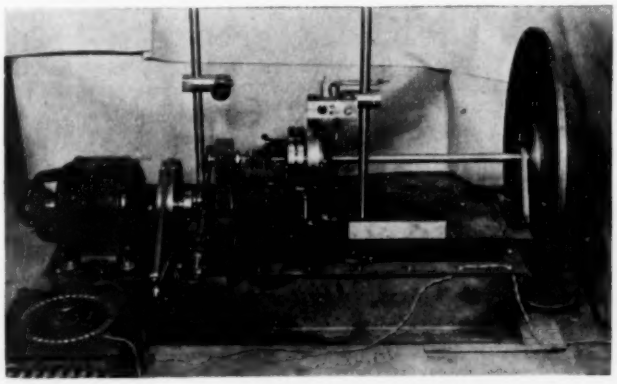


FIG. 20 MODEL WITH MECHANICAL TORQUE INPUT

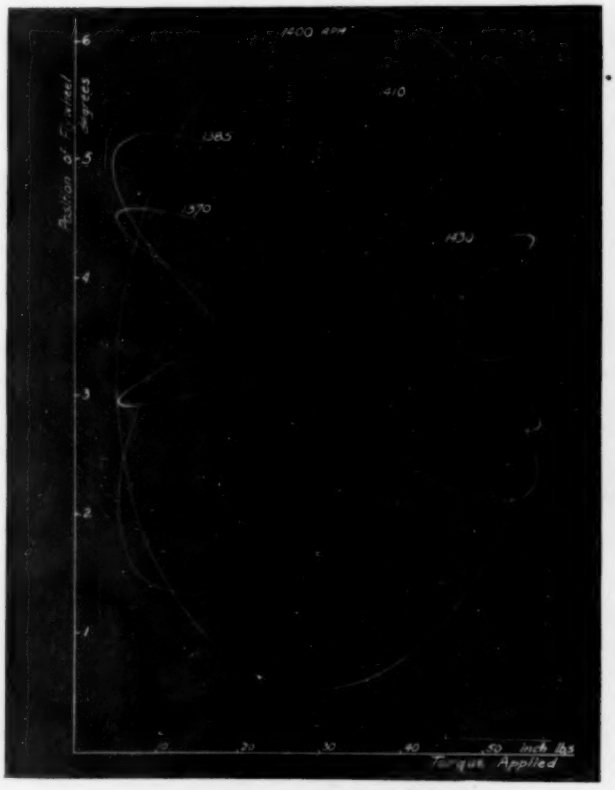


FIG. 21 RECORDS WITH DAMPER LOOSE

moving 30 deg. This gave a horizontal track of light on the plate. The vibration of the flywheel I_1 rocked the mirror about its horizontal axis giving a vertical track of light. The result was a figure resembling an ellipse. Fig. 21 shows a plate which was exposed at five speeds near resonance when the damper was loose, while Fig. 22 gives the same with the damper set in its optimum position.

With this apparatus tests were carried out on three damper sizes with ratios $I/I_1 = 0.104$, 0.241, and 0.963, respectively. The test results are given in Fig. 23, showing a complete agreement with the theory.

The objection raised against the theory of the damper with springs as presented in the paper are of the same order as those against the damper without springs. In general, the objections are quite valid when the damper mass is so large as to *materially* change the critical speeds and to have such a large non-harmonic reaction on the machine that its motion ceases to be harmonic. This is never the case with practical dampers. Though the original critical speed will be changed into two critical speeds by the application of a spring damper, these two criticals will be close to the original and the curves presented will hold for any of these speeds for practical damper sizes. A proof of this by an exact theory seems impossible on account of the difficulties involved.

Professor Lewis' views of the damper as a complicated piece of apparatus which needs expert maintenance and can easily get out of adjustment are somewhat pessimistic. The Diesel-electric engines built by the Westinghouse Company for loco-

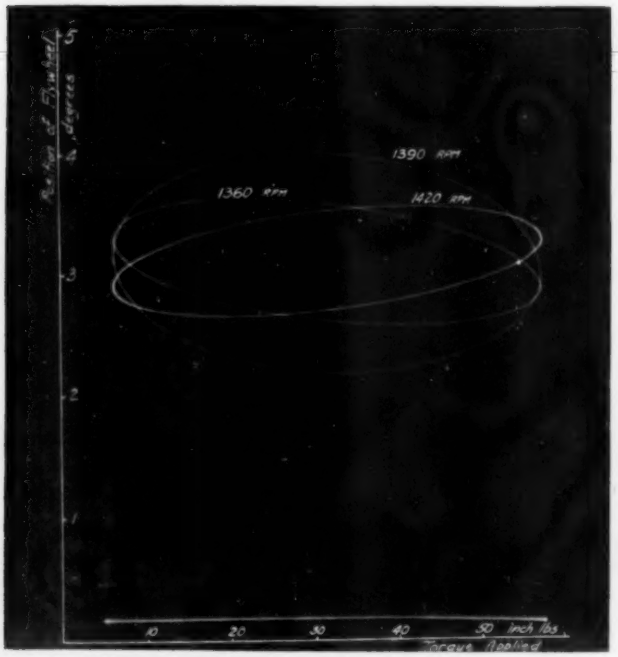


FIG. 22 RECORDS WITH DAMPER TIGHTENED RIGHT

motives and rail cars are in satisfactory operation without adjustments in service.

In reply to Dr. Wydler's questions regarding the cooling and the life of dampers, the following can be said:

No cooling has ever been necessary in the dampers on engines of 1200 hp. and less. In rail-car engines, the damper has lubricating-oil temperature when the engine runs at a speed away from resonance. For continuous running in resonance, the damper runs about 30 deg. fahr. hotter than the lubricating oil. Some dampers under our observation have run almost three years without signs of appreciable wear on the friction rings. It must be remembered that a damper usually operates at the critical speed only a short time. Even when it does operate there it does very little work—less than 1 per cent of the engine rated horsepower is dissipated by the damper in an ordinary case.

Mr. Dickson's design of the dampers on the Beardmore engines was very rational in that it was based on the idea of

energy dissipation. Without having seen Mr. Dickson's design methods, the authors would probably have never written this paper. The only essential difference between our method and Mr. Dickson's is that we consider the motion of the damper flywheels in some detail, while he assumes that it is rotating at the average shaft speed. Our method leads us to a definite damper inertia value for a desired amplitude, while Mr. Dickson was forced to determine his damper size by reasoning not related to energy considerations. Our method leads to a value of the optimum friction torque in the damper which is always larger than the torque which would be used if Equation [8] were followed. In an actual design we both should arrive at about the same damper flywheel inertia; the practical difference coming only in the loading used on the friction surfaces.

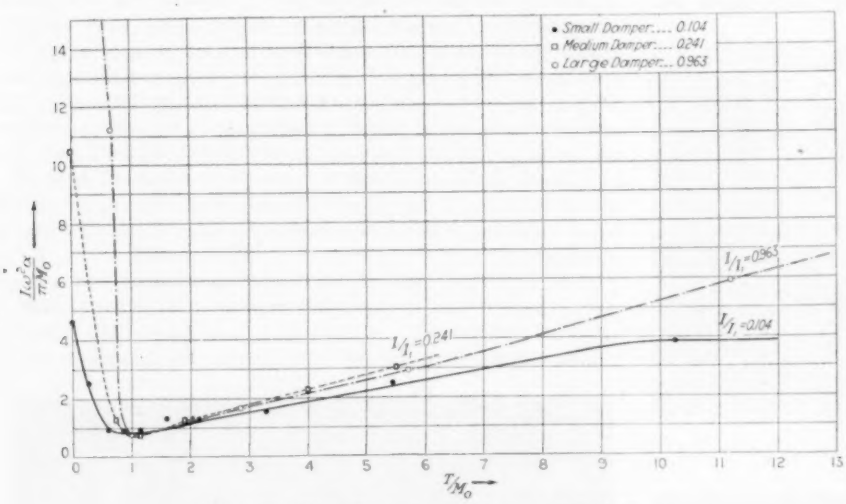


FIG. 23 TEST RESULTS ON MODEL OF FIG. 20

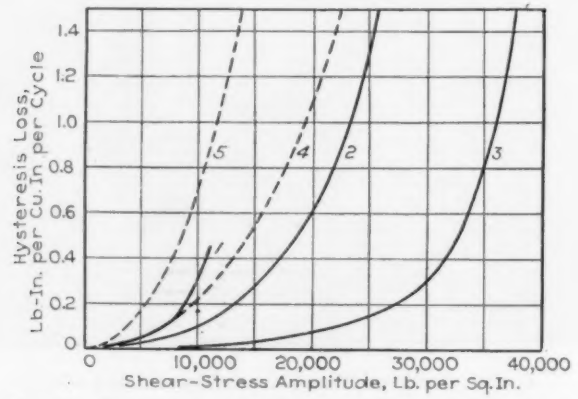


FIG. 24 EXPERIMENTAL HYSTERESIS CURVES COMPARED TO PORTER'S LAW

Mr. Gage's observations from the operating point of view are especially appreciated. There is no difference between a damper "made" right and one "designed" right. The "making" by pure "cut and try" takes much more time than the "designing," and the result in many cases will not be as good.

Mr. Dashefsky's tests on viscous dampers are interesting. The advantage of a purely viscous damper is that the optimum value for α_0 for a given damper flywheel is $\pi/4$ times as small as for the friction damper. This was stated by Carter in a report of the British Aeronautical Research Committee in 1927, and the authors' analysis leads to the same result.

Mr. Porter criticizes the paper on a number of minor points. The first statement made is that "the authors appear to recognize the law of hysteresis damping only in a qualitative sense." In this connection an inspection of Fig. 24 seems instructive. The figure shows several hysteresis damping curves; No. 1 was taken by Rowett on mild steel; No. 2 by Föppl[22] on oil-hardened carbon steel; and No. 3 by Föppl on "crankshaft steel."

Professor Lewis used Rowett's law for stresses below 7000 lb. per sq. in. and expressed it in the mathematical form:

$$K = 1.37 \times 10^{-10} S^{2.3}$$

Curve No. 4 in Fig. 24 shows this law. Mr. Porter found that this law did not give correct results, and replaced the factor 1.37 by 4.5. In looking over Mr. Porter's valuable list of calculated and measured amplitudes, we feel that the coefficient proposed by him is still too small if averages mean anything in discussing the results. While over two-thirds of all the cases given show remarkably good agreement between calculations and tests, three-fourths of them show calculated amplitudes larger than measured amplitudes. This indicates that the actual damping losses are larger than the calculated losses. Taking an average, the coefficient should be 5.85 instead of 4.5 as recommended by Mr. Porter.

The discrepancies shown in Fig. 24, granted that Föppl's results are correct, can be explained by the assumption that hysteresis really plays a small part in damping Diesel-engine vibrations or that stresses much higher than ever imagined exist in parts of the crankshaft.

If this is right, we are back to the position of Holzer and Wydler, who considered the actual law of damping to be unknown and proposed the assumption of an overall damping which followed the law of viscous friction. This, at least, has the advantage of leading to energy losses proportional to the square of the amplitude, which is more convenient for calculation purposes than the exponent 2.3 used by Lewis in his hysteresis law. It may be that the form of Professor Lewis' hysteresis law has been so successful only because its exponent is so near in value to the exponent which would hold in the case of a frankly empirical overall viscous damping law. If, in some manner, a damping force proportional to velocity actually exists, then the frequency of vibration should also be a factor in the energy lost. This may actually be the case. Amplitudes as big as 6 deg. have been reported on installation having low natural frequencies, whereas 3 deg. is a very large amplitude on rail-car Diesels which have one-noded natural frequencies. It is realized that the difference in physical layout of the systems compared might account for the difference in magnitude without calling in any frequency effect.

It would be interesting if all the available torsiograph records of internal-combustion engines along with the pertinent data of the engines from which the records were taken could be tabulated somewhat as Mr. Porter has presented his results. Actual hysteresis tests on crankshaft steels should be carried out. So far as we know, no American investigator has studied the hysteresis loss as a function of stress amplitude.

[22] Föppl, Becker, and Heydekampf: "Die Dauerprüfung der Werkstoffe," Berlin, 1929, p. 99.
Föppl, *Schweiz. Bauzeitung*, vol. 86, no. 23.

APM-52-14

Longitudinal Wave Transmission and Impact

By L. H. DONNELL,[1] ANN ARBOR, MICH.

When a force is applied to a body, waves of stress and velocity radiate through the body from the point of application of the force. When these waves strike the boundaries of the body they may be wholly or partially transmitted to surrounding bodies or reflected back into the body, where they may reverberate back and forth within its confines, like sound waves within the body of air in a closed room. By superposition of these radiated and reflected waves the stress and motion of all parts of the body are gradually, and in general discontinuously, brought up to the values ordinarily assumed.

In the case of a small body this whole process takes place so quickly that there is great difficulty in detecting it, and generally it is of little practical importance. But there are cases where it is desirable to study the process in detail, particularly when the dimensions to be dealt with are no longer very small compared to the velocities of such waves, and in consequence the time element is no longer negligible, or where the time of application or variation of applied forces is very small and hence of the same order of magnitude as the aforementioned time element, as in cases of impact or harmonic forces of considerable frequency. In the present paper an attempt is made to consider some of the more important phases of this subject in as simple a manner as possible.

THE wave transmission considered here is "mechanical" wave transmission or the propagation of mechanical states of matter under the action of the simple laws of mechanics. The propagation of sound, of ripples on the surface of liquids, and of earthquake shocks are familiar examples. So-called "standing" waves, such as observed in the strings and pipes of musical instruments, can be considered to be the result of superposition of systems of transmitted waves.

When a force is applied to a given body, even one which we ordinarily think of as rigid, the body does not really behave as a rigid one—that is, it does not at once assume, as a whole, the state of stress or motion which we predict from elementary mechanics. When the force is applied, waves of stress and velocity radiate through the body from the point of application of the force.

When these waves strike the boundaries of the body, they may be wholly or partially transmitted to surrounding bodies (as sound waves to surrounding air, for instance) or reflected back into the body, where they may reverberate back and forth within the confines of the body, like sound waves within the body of air in a closed room. By superposition of these radiated and reflected waves the stress and motion of all parts of the body are gradually, and in general discontinuously, brought up to, or nearly up to, the values ordinarily assumed. It is necessary to say "nearly," because there is almost sure to be some loss of energy in this process, either to the surroundings or through damping action within the body, although this loss is generally negligible.

In a small body this whole process takes place so quickly that we are unaware of it and would have great difficulty in detecting it, and in such cases it is generally of little practical importance.

But there are cases where it is desirable to study the process in detail, particularly when the dimensions to be dealt with are no longer very small compared to the velocities of such waves, and in consequence the time element is no longer negligible, or where the time of application or variation of applied forces is very small and hence of the same order of magnitude as the aforementioned time element, as in cases of impact or harmonic forces of considerable frequency. In the present paper an attempt is made to consider some of the more important phases of this subject in as simple a manner as possible.

Longitudinal Pressure Waves in Thin,[2] Prismatical Bars—General

Consider a long, thin uniform bar of elastic material, Fig. 1. If an impulse is applied to the end of this bar, say, by a uniformly distributed compressive force f acting on the end for a time t, then a "pressure wave" will be started at the end and will move along the rod. This "pressure wave" is merely a zone in which the material of the rod is under longitudinal compression.

In front of the zone and behind the zone the material is entirely uncompressed and at rest. At each instant new material in front of this zone, as the portion ab, is just about to be compressed, while at the same time material at the rear of the zone, cd, is losing its compression. Thus the *zone* can move along the rod with considerable velocity, although individual particles have only the small motion involved in compression and decompression as the zone passes.

Explanation. The explanation of this phenomenon is roughly as follows: When the force f is applied it compresses the material near the end of the rod. The compressed portion is shortened slightly, which means that some of it suffers a displacement, and in being displaced is given a velocity to the right. Due to its inertia it tends to continue moving at this velocity. It strikes the stationary material in front of it and compresses and starts this moving—but in so doing is stopped itself. Thus the zone of compression moves forward.

It should be noted that the material in the compressed zone always has a certain amount of velocity,[3] while the material in front of and behind it is at rest. Hence we could equally well speak of this zone as a "velocity wave" or zone. We shall see that for a given bar there is a definite relation between the magnitude of the compression and this velocity of the particles; also both are respectively equal to the compression f and the velocity which f produced at the end of the bar when the wave was started.

The velocity of propagation of the wave or zone along the bar, on the other hand, is independent of f and is constant for a given material.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] The effect of lateral expansion, which would be of more importance in a thick bar, will be neglected.

[3] This velocity of *particles* of material in the zone must not be confused with the velocity of *propagation* of the zone along the bar.

Quantitative Demonstration

Let f = magnitude of compressive force acting on bar
t = time it acts
L = length of pressure zone
v = velocity of individual particles in the zone
V = velocity with which the zone moves along the bar
m = "inertia constant" of the bar, in this case the mass of the bar per unit length
k = "stiffness constant" of the bar, in this case the modulus of elasticity of the material times the cross-sectional area of the bar.

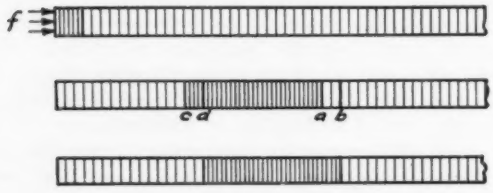


FIG. 1

Assume that, when the constant force f acts on the end of the bar for the time t, the result is a moving pressure zone of length L, all parts of which are under a uniform compression equal to f and have a uniform velocity v. The shortening of the bar due to the compression of this zone will be fL/k. The work done by the external force is therefore:

$$\frac{f^2L}{k} \quad \ldots\ldots [1]$$

This must be equal to the energy stored in the pressure zone, which, no matter where the zone happens to be, consists of the

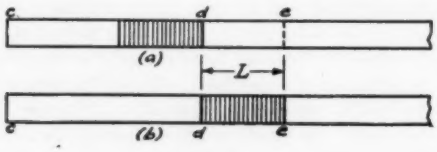


FIG. 2

potential strain energy $f^2L/2k$ (assuming that the material obeys Hooke's law[4]) plus the kinetic energy $mLv^2/2$. Hence:

$$\frac{f^2L}{k} = \frac{f^2L}{2k} + \frac{mLv^2}{2} \quad \ldots\ldots [2]$$

from which

$$v = f\sqrt{\frac{1}{mk}} \quad \ldots\ldots [3]$$

that is, the velocity of particles in a wave equals the compression in the wave times a quantity which depends only on the bar. It is interesting to note that the energy in the zone is, from [2], half potential and half kinetic.

In Fig. 2 consider as a free body a definite portion of the bar de of length L. At a certain instant the pressure zone will be in the position shown in Fig. 2 (a), and, after an interval of time which we shall call for the present t', it will have moved the distance L to the position shown in Fig. 2 (b). During this time t' the element de has had only one external force acting on it, namely, a force f acting on the end d to the right. The total change in momentum of the material in the element during this time is $mL(v - 0) = mLf\sqrt{1/mk}$ (from [3]). The impulse of the external force must equal this change in momentum, or

$$ft' = mLf\sqrt{\frac{1}{mk}}$$

from which

$$\frac{L}{t'} = V = \sqrt{\frac{k}{m}} \quad \ldots\ldots [4]$$

that is, the velocity of propagation[5] of the wave along the bar is $\sqrt{k/m}$, a constant depending only on the material of the bar as both k and m are proportional to the area of the bar. Also the length of the zone is

$$L = Vt' = Vt = \sqrt{\frac{k}{m}}t \quad \ldots\ldots [5]$$

as t' is obviously equal to t, the time the external force f acted on the end of the bar as the wave was started.

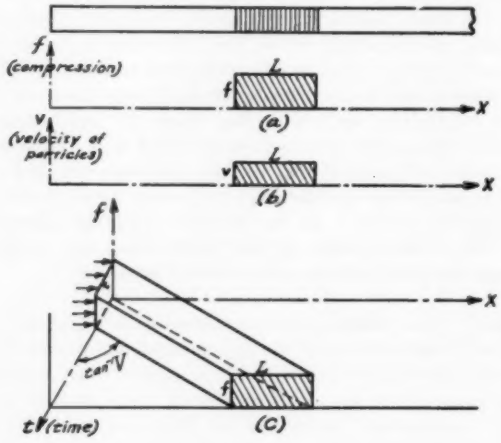


FIG. 3

We have still to prove our assumption that the velocity and compression are constant in all parts of the zone and that the compression equals f. This assumption is certainly true when the wave starts—that is, as it "crosses" the section c at the end of the bar, Fig. 2. Moreover, at the start at least, all longitudinal elements of the bar within the zone[6] have on either side of them elements with the same velocity and pressure. Hence there is no tendency to change conditions within the zone either at this moment or later, and consequently the conformation of the zone remains the same as it started, throughout its motion along the rod (provided the rod is freely supported and no frictional resistances act).

The foregoing equations apply equally well if a *tensile* force acts on the end of the rod, starting a tension wave or zone in which the material is under tension. Tension may be considered to be a negative compression. Equations [3] and [4] show that when the sign of f is changed the sign of v is changed, but V remains the same as before. Thus in a compression wave the

[4] Pressure waves in materials which do not obey Hooke's law are discussed later on.

[5] This is the velocity of the wave relative to the bar. If the bar is moving, then the absolute velocity of the wave will be the vectorial sum of this velocity and the velocity of the bar. Thus sound has a greater absolute velocity *with* the wind than against it.

[6] According to our assumption of a suddenly applied force, elements suffer an infinite acceleration when the *edge* of the pressure zone crosses them. However, the mass involved at any instant is infinitesimal, so it is possible for the unbalanced force f which acts on it at this instant to produce such an acceleration.

velocity of individual particles is in the same direction as the wave moves; but in a tension wave the velocity of the particles is in the opposite direction from that of the wave.

Graphical Representation of Waves

The conditions in a wave such as has been discussed can be shown by diagrams, as in Fig. 3. Fig. 3(*a*) shows the compression f at all points in the rod at a given instant, ordinates above the x-axis representing compression and ordinates below representing tension. Fig. 3(*b*) shows the longitudinal velocity v of particles of the rod at a given instant, ordinates above the x-axis representing velocity to the right and ordinates below representing velocity to the left.

The conditions at all instants can be represented by a three-dimensional diagram such as shown in perspective in Fig. 3(*c*). In this diagram the conditions at any instant are given by a section parallel to the fx plane, as shown shaded.

Methods for Simplifying Complex Problems

If we establish the laws governing simple cases, such as the effect of a constant force under various conditions, it is possible in many cases to study more complex problems by replacing the complex action by an equivalent series of simpler actions and superposing their results.

When conditions are variable it is sometimes convenient to consider the variation to take place in several sudden steps;

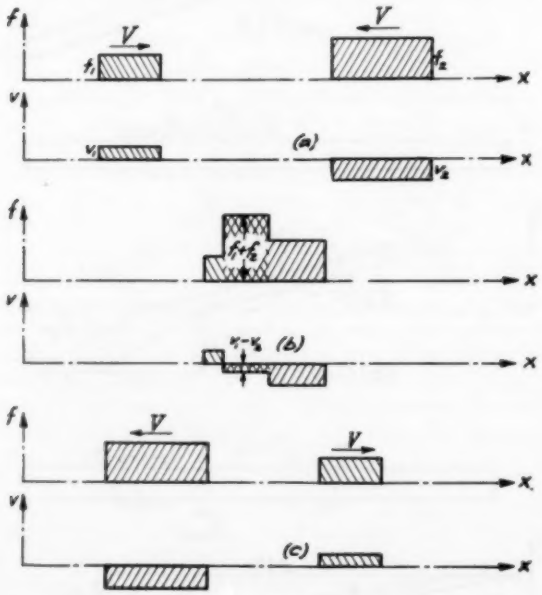


Fig. 4

it is then frequently obvious what the result would be if the number of steps were made infinite.

Superposition of Waves

When two waves such as we have been considering come together, the principle of superposition holds as in other branches of mechanics, if there are no frictional resistances and the material follows Hooke's law. Hence the resulting compression or velocity is the vectorial sum (which in this case is the same as the algebraic sum) of the compressions or velocities in each wave.

When two waves coming from opposite directions meet, as shown in Fig. 4, the compression in the resulting zone is $f = (f_1 + f_2)$, and the velocity is

$$v = (v_1 - v_2) = \frac{(f_1 - f_2)}{\sqrt{mk}} \quad \ldots\ldots\ldots\ldots [6]$$

It is evident that in this zone the relation between f and v given by [3] for a simple wave does not hold, and hence the energy is no longer half kinetic and half potential. However, the total energy in this zone is unchanged; it consists of the potential energy $(f_1 + f_2)^2L/2k$ and the kinetic energy $mL(v_1 - v_2)^2/2 = (f_1 - f_2)^2L/2k$ (from Equations [2] and [3]). The sum of these is $(f_1^2 + f_2^2)L/k$, which equals the energy before meeting. After

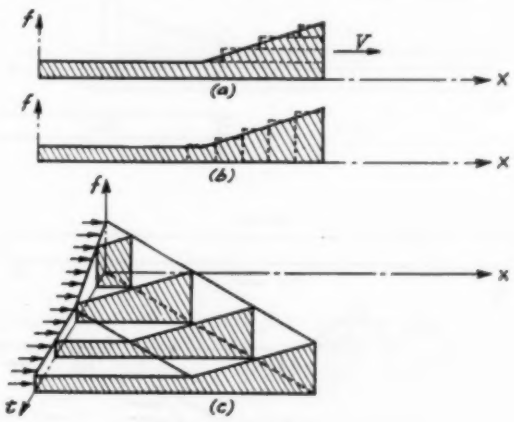


Fig. 5

passing, the waves return to their original conformation the same as if they had never met, as in Fig. 4(*c*).

If two waves going in the same direction superpose (as is possible when waves are started by forces at intermediate points along the rod) the resulting compression, by the principle of superposition, is $(f_1 + f_2)$ and the velocity $(v_1 + v_2)$. The relation between f and v in this case is the same as in a simple wave produced by a single force, Equation [3], and the energy will be half potential and half kinetic. However, the energy will be found to be considerably more than it would be if the superposed waves were separated. (This could also be predicted from the fact that, from [2], the energy of a wave varies as the square of f or v.) The explanation is that additional work is done by one external force due to the particle motion in the wave started by another external force.

Waves Produced by Variable Forces

If the external force producing the wave varies—suppose, for example, it decreases uniformly for a while and then remains constant—we can imagine it to be replaced by a series of constant forces, either as indicated by Fig. 5 (*a*) or Fig. 5 (*b*). In this manner, if we know the laws governing waves produced by constant forces, we can apply them to cases where the forces vary.

It is very useful, for visualizing the action in complex cases, to note that as the wave moves along the rod it keeps the same conformation—just as though the fx curve were constructed on paper and drawn across the end of the rod and out along the rod. Thus in Fig. 6 imagine the shaded figure drawn on a piece of cardboard *abcd* and the cardboard moved to the right past the fixed end of the rod *e*. The ordinate *pq* of the diagram passing the end of the rod represents the magnitude of the external force acting on the end of the rod at the instant. The fx diagram is thus seen to be a moving record of the value of the external force on the end of the rod; the ordinate *rs*, farthest

from the end, represents the initial value of this force, and, as we go back, the ordinates between give a complete history of the value of the external force up to the present. This is also shown by Fig. 5(*c*); the fx diagram at any instant is the same as the ft diagram up to that point (which is the history of the external force) except that the horizontal dimensions are multiplied by V.

Waves Due to a Force Applied at an Intermediate Section

Consider a force f applied at section d, Fig. 7, for a time t. The effect is the same as if the force f acted partly on the end of

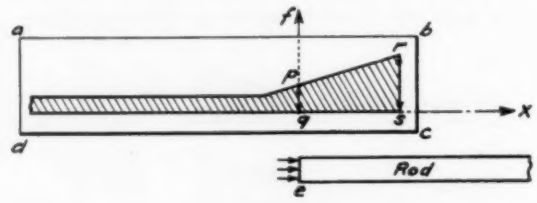


Fig. 6

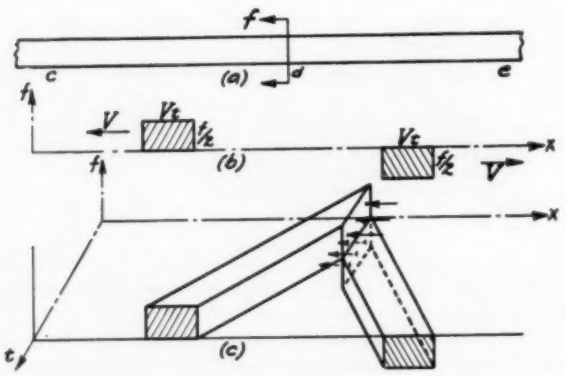


Fig. 7

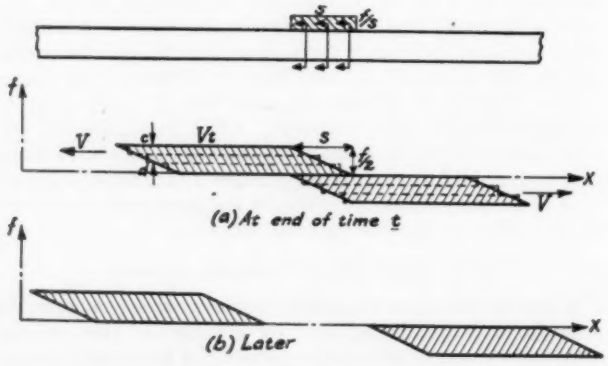


Fig. 8

a rod dc and partly on the end of a rod de, sending a compression wave of length $L = Vt$ along dc to the left, and a tension wave of the same length along de to the right. Since the velocity of particles at the section d is the same for both rods when the waves are started, the force f must, from [3], be equally divided between dc and de.[7] Hence the wave traversing dc will be under a compression of $f/2$ and the one traversing de under a tension of $f/2$, as shown in Fig. 7 [*b*] and [*c*].

If the force f is distributed, say, uniformly over a length s of the rod (that is, a distributed force f/s per unit length, acts for a time t), the result will be the same except that portions of the wave due to different parts of the force will be displaced lengthwise with respect to each other, as shown in Fig. 8. (In these figures the compression at any point is the vertical intercept of the shaded figures, as, for instance, the length cd.) The

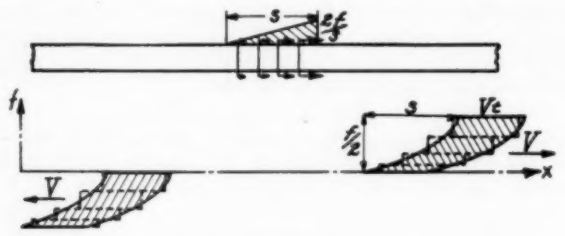


Fig. 9

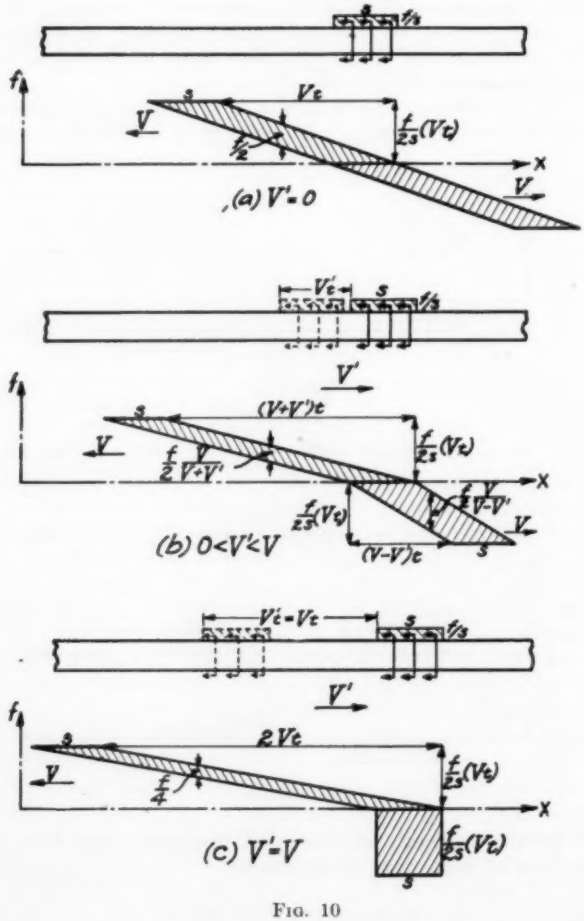


Fig. 10

broken lines show what the result would be if the distributed force were replaced by three concentrated forces; if it were replaced by a larger number of concentrated forces we should obviously approach the condition shown by the full lines.

Similarly a force with triangular distribution would give rise to the waves shown in Fig. 9. These waves, produced by distributed forces, have the same area of fx diagram as if the forces were concentrated. However, their energy is less, because the

[7] If two different rods are joined at d, the condition for equal velocities at d is, from [3]:

$$f_{cd}\sqrt{\frac{1}{m_{cd}k_{cd}}} = f_{de}\sqrt{\frac{1}{m_{de}k_{de}}}$$

and the force f would be divided between the two rods in the ratio given by this equation.

energy depends on the square of f and the first power of L, from Eq. [1]; the explanation is that conditions are here the reverse of those considered before, under superposition of forces—there we were superposing the effects of forces, and here we are separating them.

Waves Produced by Traveling Forces

These are of interest in connection with the study of frictional resistances, longitudinal waves in railway trains, etc. Suppose that a force f is uniformly distributed over a length s and that its place of application moves uniformly along the rod (to the right in Fig. 10) at the rate V'. If the force is applied for a time t, then its place of application will move a distance $V't$ during this time.

When $V' = 0$ we have the case shown in Fig. 8(a); this can evidently be equally well represented in the form shown in Fig. 10 [a]. When V' is not zero we have the result shown in Fig. 10(b); this can be shown by replacing the moving force by a series of fixed forces acting successively at different points along the rod.

If $V' = V$ the conditions become as shown in Fig. 10(c). In this case the effect is cumulative on the wave going in the same direction as the forces. The compression in this wave, $(f/2s) \times (Vt)$, increases with time and is inversely proportional to s. Hence a concentrated force ($s = 0$) whose point of application moves with the velocity V would immediately produce infinite stresses—in practice failure or plasticity—and even a force not perfectly concentrated would soon produce such conditions.

Longitudinal Waves in Railway Trains

When a train passes over an unevenness in the track the effect is the same as that of a force whose point of application travels along the train at the speed at which the train is moving. A long train may be considered roughly as a bar with uniformly distributed mass (at least the part of the train behind the locomotive) and more or less uniformly distributed flexibility (due almost entirely to the coupling springs). It will have a definite rate of propagation of tension or compression waves, depending on the mass of the cars and the characteristics of the couplings.

The longitudinal external forces on a train are: (1) The frictional resistances to the motion of the train, which never change suddenly and so can hardly produce any of the effects we are considering. (2) Braking forces, which may be changed suddenly but which are usually uniformly distributed along the length of the train and hence balanced at each point by the resulting inertia forces, in which case they would not produce waves. (3) The pull of the engine, which may change suddenly and produce waves in the train of the same type as when a force is applied to the end of a bar. (4) The longitudinal component of gravity forces, which may act as a traveling force, as stated at the beginning. It should be noted that what we are interested in here is sudden *changes* in these forces, which are not immediately balanced locally as was assumed to be the case with braking forces. Thus if conditions are steady, with the engine exerting a pull of 5 tons, and its pull is suddenly increased to 8 tons, the effect is to superpose on the previous steady conditions a suddenly applied tensile force of 3 tons; if the pull is decreased the effect is that of a suddenly applied compressive force. If the change is spread over a considerable period of time the result is not so great, because by the time the full change is in effect there are likely to be other counterbalancing forces locally applied and the wave started by the first part of the change may have come back, reflected from the free end of the train as a wave of *opposite* sign, as considered later.

From the previous discussion it would seem that bad effects may be produced by the cumulative effect of gravity components when the train traverses a change in the slope of the track at the same speed as the speed of propagation of longitudinal waves. Such effects might of course be controlled by designing the couplings so that the speed of propagation of waves would be above the speed range of the trains. Let b = length of each car, B = length of the train, W = weight of the train, and K = spring constant of a complete coupling (including the portion

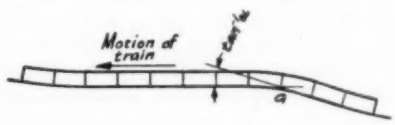


Fig. 11

on both of the cars connected). Then the average inertia and stiffness constants are respectively

$$m \triangleq \frac{W}{gB} \quad \text{and} \quad k = bK$$

The velocity of propagation of waves is

$$V = \sqrt{\frac{k}{m}} = \sqrt{\frac{bKgB}{W}}$$

When the train passes a *change* in slope of the tracks equal to α, as shown at a, Fig. 11, the change in longitudinal gravity component is a distributed force of approximately $W\alpha/B$ per unit length. If the train is moving to the left in Fig. 11, this is equiva-

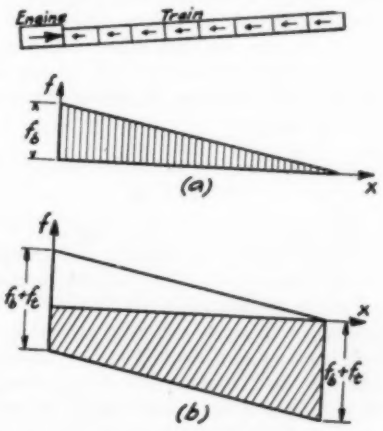


Fig. 12

lent to a traveling force acting to the left and moving along the train to the right a distance equal to the length of the train. Comparing this with Fig. 10(c) it is evident that the greatest effect is a tension in the rear of the train as it passes a, whose magnitude is $W\alpha B/2B = W\alpha/2$. If α is negative, this is a compression.

The greatest load on a coupling under steady conditions is likely to be at the front of a train going up a grade. The maximum tension would be $W'\alpha'$, where W' is the weight of the train minus the engine, and α' the absolute slope of the grade. The dynamic tension $W\alpha/2$ was calculated on the assumption that the train was homogeneous longitudinally. Actually it is a series of distinct units, and it is reasonable to suppose that as the wave front passes along such a series of units the tension will fluctuate above and below the average value $W\alpha/2$, so that the maximum tension may be greater than that calculated. The conditions in practice are more complicated than those assumed.

As another example consider the case of a railway train made up of cars having no individual brakes, so that any braking effort must be applied entirely at the engine. If such a train is descending a grade at a uniform speed, being held back by the braking force f_b applied by the engine (which will be equal to the longitudinal component of the weight of the train minus the frictional resistance), there will be a static state of compression along the train as indicated in Fig. 12(*a*). If now the engineer suddenly releases the brakes and opens the throttle sufficiently to apply a tractive force f_t (as is not unusual, when approaching the bottom of the grade), this is equivalent to superposing upon the foregoing condition a suddenly applied tensile force of $f_b + f_t$. This sends a tension wave along the train which, superposing on the previous condition of compression, produces the condition indicated in Fig. 12(*b*) when it reaches the end of the train. It will be observed that the coupling of the rear car is subjected to a tensile load of approximately $f_b + f_t$ at this instant, which may be sufficient to produce fracture.

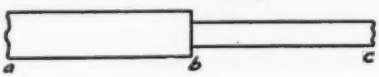


FIG. 13

In the propagation of pressure waves in trains the damping effect of friction is probably not as great as might be thought. If the velocity of the train is in the same direction as the velocity of particles v produced by a wave, or if it is in the opposite direction but is larger than v (as will usually be the case), then the total velocity of any part of the train will always be in the same direction, whether a wave is passing or not. Consequently that part of the frictional resistances which is of the type exhibited by solids rubbing on each other, being independent of the velocity, will exert a constant force unaffected by waves, and can play no part whatever in either the propagation or dissipation of such waves.

EFFECT OF A SUDDEN CHANGE IN CROSS-SECTION OR MATERIAL OF A BAR[8]

Section *b*, Fig. 13, represents the junction of two uniform portions *ab* and *bc* having the inertia and stiffness constants m_a, k_a and m_c, k_c, respectively. Let a pressure wave having the compression f strike the section *b* from the left. We may consider that this wave is wiped out as it crosses *b*, but that the pressure which it exerts at *b* starts two new waves from *b*—one which we shall call the "transmitted" wave moving to the right along *bc* with a compression f_t; and one which we shall call the "reflected" wave moving back along *ab* with a compression f_r. This is shown in Fig. 14.

Equilibrium of an element of the rod at *b* during this process requires that[9]

$$f + f_r = f_t \qquad [7]$$

The resultant velocity of particles to the left of section *b* during this process is, from [6],

$$v = \frac{f - f_r}{\sqrt{m_a k_a}}$$

and this must be equal to the velocity to the right of *b*, or

$$\frac{f - f_r}{\sqrt{m_a k_a}} = \frac{f_t}{\sqrt{m_c k_c}} \qquad [8]$$

Solving [7] and [8] for f_t and f_r, and letting $\sqrt{m_c k_c / m_a k_a} = r$, we find the compression in the transmitted wave is

$$f_t = \frac{2r}{r+1} f \qquad [9]$$

and the compression in the reflected wave is

$$f_r = \frac{r-1}{r+1} f \qquad [10]$$

The velocity of particles in these waves can be found from [3]. The length of the reflected wave is the same as that of the original wave; the length of the transmitted wave is V_c/V_a or $\sqrt{m_a k_c / m_c k_a}$ times the length of the original wave. If the sudden change is one of area only, from A_a to A_c, then the lengths of all three waves are the same and $r = A_c/A_a$.

Equation [10] shows that if $r < 1$ (as when a wave strikes a reduction in area) the reflected wave is of opposite sign to the original—tension if the latter is compression, and compression if the latter is tension. If $r > 1$ (as when a wave strikes an enlargement) the reflected wave is of the same sign as the original. The transmitted wave is, from [9], always of the same sign as the original wave.

If $r = 1$ (that is, *b* is merely a section in a uniform bar) then [9] and [10] reduce to $f_t = f$ and $f_r = 0$, as expected.

FREE AND FIXED ENDS

If m_c and k_c are zero (that is, *b* is a free end of the bar) then r is zero, and we find from [9] and [10] that $f_t = 0$ and $f_r = -f$. This means that a wave is reflected back from a free end unchanged except for sign; a compression wave is reflected as a similar tension wave, and vice versa. Superposition of the incoming and outgoing waves at the free end makes the resultant compression there zero, while the resultant velocity of the particles at this point is

$$\frac{f - (-f)}{\sqrt{m_a k_a}} = \frac{2f}{\sqrt{m_a k_a}}$$

or twice the velocity in the original wave.

If k_c is infinite (that is, *b* is a rigidly fixed end) then r is infinite and we find $f_r = f$; hence the wave is reflected back entirely unchanged. Also $f_t = 2f$; however, the energy in this transmitted wave is, from [1], $f_t^2 L/k_c = 0$. Half of f_t can be considered as produced by the incoming wave, while the other half starts the outgoing wave. The resultant velocity of particles at this point is $(f - f_t)/\sqrt{m_a k_a} = 0$, as would be expected.

LONGITUDINAL IMPACT ON A PRISMATICAL BAR BY AN INFINITE MASS

If a perfectly rigid mass moving with a longitudinal velocity v_0 strikes the end of a bar squarely,[10] a pressure wave will be started having a velocity of particles equal to v_0, and hence a compression, from [3], of

$$f_0 = v_0 \sqrt{mk} \qquad [11]$$

This force will also be acting on the mass, tending to decelerate it. If the mass is of infinite size, however, it will continue to

[8] It will be assumed for the present that stresses are at all times uniformly distributed over the cross-sections, although this is of course not true locally when there is a change in the cross-sections. The effect of non-uniform distribution is discussed under "Gradual Changes."

[9] There are no changes of velocity *within* the waves considered, hence no inertia forces enter this equation of equilibrium.

[10] We must assume that contact takes place at the same instant over the whole surface of the end of the rod, although this condition is difficult to realize in practice.

move at the velocity v_o and the wave sent out will have a uniform compression f_o.

If the other end of the bar is free, this wave, on reaching the other end, will be reflected back with opposite sign. The portion coming back will superpose on the portion going out, producing a resultant compression $f_o - f_o = 0$ and a resultant velocity of particles $v_o + v_o = 2v_o$. When the reflected wave gets back to the starting point (which will be after a total time $2b/V$, where b is the length of the rod), the whole rod will be under zero stress and moving with a velocity twice that of the mass, so that it will separate from the mass at this instant and all waves will cease. This condition will be approximated if a large mass strikes a small free rod.

We also have a condition like this if two equal free rods c and d strike each other longitudinally. If the rods have initial velocities of v_c and v_d, respectively, the plane of contact will have a constant velocity of $(v_c + v_d)/2$ during the impact.[11] Hence the effect on each rod is the same as being struck by an infinite mass moving at this speed, $(v_c + v_d)/2$. Now for rod c,

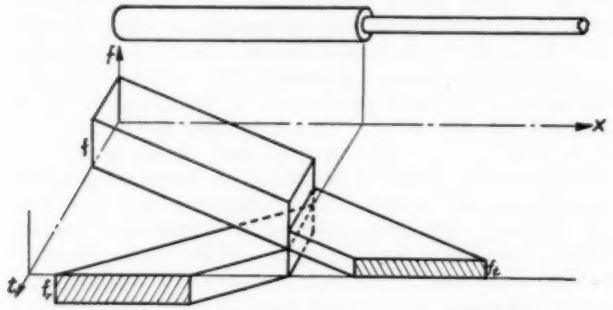


FIG. 14

this would be the same as if it were stationary and the infinite mass had a velocity of

$$\frac{v_c + v_d}{2} - v_c = \frac{v_d - v_c}{2},$$

within an isolated system having a uniform velocity of v_c. From the preceding paragraph the velocity of rebound inside this system will be twice the velocity of the infinite mass, or $2(v_d - v_c)/2 = v_d - v_c$. This, added to the velocity of the system, gives the absolute final velocity of c as $(v_d - v_c) + v_c = v_d$. Similarly, the final velocity of d will be v_c. The length of time the bars are in contact is of course $2b/V$, and the maximum compressive force is $\frac{v_d - v_c}{2}\sqrt{mk}$.

If the bars are of unequal length, b_c and b_d (suppose $b_c < b_d$) but are of the same materials and cross-sections, the conditions will be just the same as above at first, because the lengths can have no effect until the reflected waves get back. The pressure between the bars will become zero when the reflected wave returns in the shorter bar (after a time $2b_c/V$) and the conditions in the shorter bar c will be just the same as above; that is, all waves will have ceased and it will be moving at a velocity v_d. As the end of the longer bar in contact with it is now moving with the same velocity the bars will remain in contact, but without pressure, until the reflected wave returns in the longer bar (after a total time $2b_d/V$), and at this moment the longer bar will have a wave moving in it of total length $2b_c$ and a uniform compression $\frac{v_d - v_c}{2}\sqrt{mk}$. This wave will continue to move along the rod after separation, being reflected from the two free ends with a change in sign at each reflection, until it is damped out by friction. From the law of conservation of momentum, after separation the total momentum of the longer bar divided by the mass of the bar (let us call this the "average" velocity of the bar, v_d') must remain constant and is found from

$$mb_cv_c + mb_dv_d = mb_cv_d + mb_dv_d'$$

which gives $v_d' = v_d + (b_c/b_d)(v_c - v_d)$. This is the velocity which d will have when the waves are all damped out; the kinetic energy in the system will then be $(m/2g)(b_cv_d^2 + b_dv_d'^2)$,

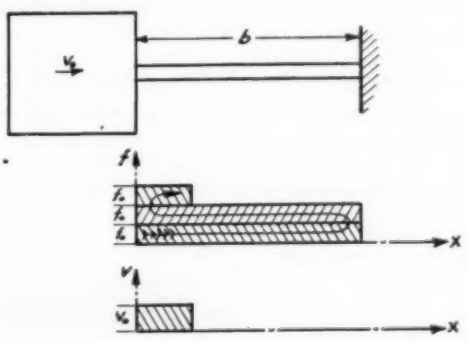


FIG. 15

while the original energy was $(m/2g)(b_cv_c^2 + b_dv_d^2)$. The ratio of lost energy to the original energy is therefore, using the above value of v_d':

$$\frac{(b_cv_c^2 + b_dv_d^2) - (b_cv_d^2 + b_dv_d'^2)}{(b_cv_c^2 + b_dv_d^2)} = \frac{\left(1 - \frac{b_c}{b_d}\right)(v_c - v_d)^2}{\left(v_c^2 + \frac{b_d}{b_c}v_d^2\right)}$$

If the bars are of unequal materials or cross-sections, and have the constants m_c, k_c, and m_d, k_d, then the plane of contact, during the impact, will have a constant velocity v_s of such a magnitude that the compressions of the waves started in the two bars will be the same, or:

$$f_c = (v_s - v_c)\sqrt{m_ck_c} = -f_d = (v_d - v_s)\sqrt{m_dk_d} \quad \ldots [12]$$

from which

$$v_s = \frac{\alpha v_c + v_d}{\alpha + 1}, \text{ where } \alpha = \sqrt{\frac{m_ck_c}{m_dk_d}} \quad \ldots [13]$$

As before, the pressure between the bars will drop to zero when the reflected wave returns in one of the bars, and actual separation will occur as soon as it returns in both of them, that is, after a time $2b_c/\sqrt{k_c/m_c}$ or $2b_d/\sqrt{k_d/m_d}$, whichever is larger. Following the same reasoning as before, if the reflected wave returned first in bar c, then this bar will be unstressed after separation and will have a velocity of $2v_s - v_c$. The bar d will have a wave moving in it, at separation, of length $2b\sqrt{k_d/m_d}/\sqrt{k_c/m_c}$ and with a uniform compression given by [12]. Using the law of conservation of momentum, the "average" velocity of this bar will be constant and equal to $v_d + 2\beta(v_s - v_s)$, where $\beta = m_cb_c/m_db_d$. The maximum compression in each bar is given by [12], and the lost-energy ratio, obtained as before, is

$$\frac{4(\alpha - \beta)(v_c - v_d)^2}{(\alpha + 1)^2(v_c^2 + \frac{1}{\beta}v_d^2)}.$$

[11] If we consider the rods to be parts of an isolated system having a uniform velocity of $(v_c + v_d)/2$, then they will have symmetrical motion within this system, and from symmetry within this system their plane of contact must be stationary with respect to the system—and will thus have an absolute velocity of $(v_c + v_d)/2$.

If a perfectly rigid infinite mass strikes one end of a bar which is fixed at the other end, the resulting wave (with compression given by [11]) will, on reaching the fixed end, be reflected with the same sign. The reflected portion, superposed on the portion going out, produces a resultant compression $f_o + f_o = 2f_o$ and a resultant velocity of particles $v_o - v_o = 0$. When the reflected wave gets back to the end struck, the velocity of this end must remain v_o; hence the wave must be reflected again as from a fixed end, so as to make the resultant velocity due to the three superposed waves $v_o - v_o + v_o = v_o$; the resultant compression is $f_o + f_o + f_o = 3f_o$, as shown in Fig. 15. The wave will continue to emerge from the end struck and to be reflected from both ends without change of sign, so that the compression at the end struck will increase by the amount $2f_o$ at the end of every interval of time $T = 2b/V$.

Longitudinal Impact by a Finite Mass

If the mass is rigid but of a finite magnitude M, then the compression in the rod at the end in contact with it will decelerate it, and its velocity (and hence the velocity of particles in the rod at this point) will gradually decrease, as indicated

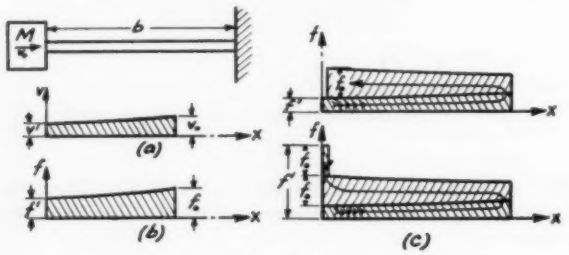


Fig. 16

by Fig. 16(*a*). Also, from [3], the compression in the rod at this point must decrease proportionally, as indicated by Fig. 16(*b*). Calling the compression and velocity of particles at the end struck f' and v', the equation of equilibrium of the mass M is

$$f' + M\frac{dv'}{dt} = 0$$

From [3], $v' = \frac{1}{\sqrt{mk}} f'$, and this equation becomes, letting $\frac{\sqrt{mk}}{M} = B$:

$$\frac{df'}{f'} = -Bdt$$

Integrating,

$$\log_e f' = -Bt + c, \text{ or: } f' = C_e{}^{-Bt}$$

Using the initial condition that $f' = f_o$ when $t = 0$, we find that $C = f_o$, hence

$$f' = f_o e^{-Bt} \ldots\ldots\ldots\ldots\ldots\ldots [14]$$

After an interval of time $T = 2b/V$ (that is, just before the reflected wave returns), f' and v' will be

$$f' = f_o e^{-BT} = f_o e^{-2/\mu},\ v' = \frac{f'}{\sqrt{mk}} = \frac{f_o}{\sqrt{mk}} e^{-2/\mu} = v_o e^{-2/\mu}$$

where $\mu = M/mb$ is the ratio of the mass of the striking body to the mass of the rod, so that

$$BT = \frac{\sqrt{mk}}{M}\frac{2b}{V} = \frac{\sqrt{mk}}{M} 2b\sqrt{\frac{m}{k}} = \frac{2mb}{M} = \frac{2}{\mu}$$

If the other end of the rod is free, the reflected wave will be of opposite sign and, when it returns, v' will be suddenly increased by v_o, the particle velocity of the wave front. As M will still be moving at the above velocity $v_o e^{-2/\mu}$, separation will occur at this instant. The "average velocity" of the rod after separation, found from the law of conservation of momentum, is $\mu v_0 (1 - e^{-2/\mu})$. A wave of length $2b$ and a compression which can be found from [14] will be left moving in the rod. The maximum compression is of course f_0 and the time of contact T. The lost-energy ratio, obtained as before, is:

$$1 - e^{-4/\mu} - \mu(1 - e^{-2/\mu})^2$$

If the other end of the rod is fixed the reflected wave will be of the same sign and f' will be suddenly increased by $2f_0$ at the end of every interval of time T, due to the return of the wave front, as shown in Fig. 16(*c*). We must therefore obtain a separate expression for f' for each one of these intervals. After the first interval there will be waves going out from the end struck and the same waves coming back to this end an interval of time T later, having been reflected from the opposite end. Let us designate by F the total compression, at the end struck, of all waves moving *away* from this end. For the reasons given above we need a separate expression for F for each interval. Let us designate the expressions for F after the intervals of time $0T$, $1T$, $2T$, $3T$, $\ldots\ldots nT$ by: $F_0, F_1, F_2, F_3 \ldots F_n$. F_0 is given by [14]. The resultant wave coming back *toward* the end struck is merely the wave *sent out* during the preceding interval, delayed, however, a time T, due to its travel across the rod and back. The compression which it produces at the end struck is obtained by substituting $(t - T)$ for t in the expression for the wave sent out during the preceding interval. The general expression for the total compression at the end struck during any interval is therefore

$$f' = F_n + F_{n-1}(t - T) \ldots\ldots\ldots\ldots [15]$$

where $F_{n-1}(t - T)$ means the expression F_{n-1} with t replaced by $(t - T)$. The net velocity of particles at the end struck is, from (6),

$$v' = \frac{1}{\sqrt{mk}}\left[F_n - F_{n-1}(t - T)\right] \ldots\ldots\ldots [16]$$

The equation of equilibrium of the mass M during this interval is, as before:

$$f' + M\frac{dv'}{dt} = 0$$

Using [15] and [16] and $B = \sqrt{mk}/M$, this becomes

$$dF_n + BF_n dt = dF_{n-1}(t - T) - BF_{n-1}(t - T)\,dt$$

This can easily be integrated by multiplying through by e^{Bt}:

$$[e^{Bt}dF_n + BF_n e^{Bt}dt] = [e^{Bt}dF_{n-1}(t - T) + BF_{n-1}(t - T)\,e^{Bt}dt] - 2BF_{n-1}(t - T)\,e^{Bt}dt$$

$$d[e^{Bt}F_n] = d[e^{Bt}F_{n-1}(t - T)] - 2BF_{n-1}(t - T)\,e^{Bt}dt$$

Integrating and dividing by e^{Bt},

$$F_n = F_{n-1}(t - T) - 2Be^{-Bt}\int F_{n-1}(t - T)\,e^{Bt}dt + C \ldots [17]$$

Using this we can find the expressions F_0, F_1, F_2, etc., in turn starting with the known value of F_0, the right-hand side of [17] being always given by the preceding expression which we have just determined. The integrals will be of known functions of t, which in our case are easily integrated. The integration constant C is found in each case by using the condition that

f' at the beginning of any interval is equal to $2f_0$ plus the value of f' at the end of the last interval, that is, when $t = nT$,

$$F_n + F_{n-1}(t - T) = 2f_0 + F_{n-1} + F_{n-2}(t - T)$$

In this manner we find that:

$$F_0 = f_0 e^{z_0}(1), \qquad \text{where } z_0 = \frac{2}{\mu}\left(-\frac{t}{T}\right)$$

$$F_1 = F_0 + f_0 e^{z_1}(1 + 2z_1), \qquad \text{where } z_1 = \frac{2}{\mu}\left(1 - \frac{t}{T}\right)$$

$$F_2 = F_1 + f_0 e^{z_2}(1 + 2{\cdot}2z_2 + 2z_2^2), \qquad \text{where } z_2 = \frac{2}{\mu}\left(2 - \frac{t}{T}\right)$$

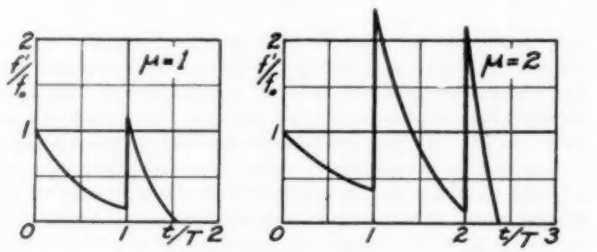


FIG. 17

$$F_3 = F_2 + f_0 e^{z_3}\left(1 + 2{\cdot}3z_3 + 2{\cdot}3z^2 + \frac{2{\cdot}2{\cdot}3}{3{\cdot}3}z^3\right)$$

$$\text{where } z_3 = \frac{2}{\mu}\left(3 - \frac{t}{T}\right)$$

etc. The general expression for any interval is:

$$F_n = f_0 \sum_{m=0}^{m=n} e z_m \sum_{h=0}^{h=m} \frac{m!(2z_m)^h}{(h!)^2(m-h)!} \quad \text{where } z_m = \frac{2}{\mu}\left(m - \frac{t}{T}\right) \qquad \ldots\ldots [18]$$

and where h, m, n are integers, and $0! = 1$ (see footnote 12), $1! = 1$, $2! = 1{\cdot}2$, $3! = 1{\cdot}2{\cdot}3$, etc.

Using these expressions in [15] and [16], the values of f' and v' at all instants are determined for any value of μ. Fig. 17 shows the variation of f' with time, obtained in this way, for $\mu = 1$ and $\mu = 2$.

Separation occurs when f' becomes zero. The time of contact is therefore obtained by setting the various expressions for f' equal to zero and solving for t/T, using only such values of μ as give values of t/T within the interval for which each expression applies. In some cases, for a given value of μ, several values of t/T are obtained (in successive intervals) at which $f' = 0$. In such cases the smallest value of t/T only is used, because separation of course takes place the *first* time f' becomes zero. Subsequent values of t/T at which it becomes zero would have a meaning only if M were attached to the end of the rod. The results of such calculations are plotted in Fig. 18, which also shows an approximate expression for the time of contact, calculated empirically to match the actual curves as closely as possible and, when μ is very large, to approach the value obtained by neglecting the mass of the rod entirely (in which case the end of the rod would undergo a half oscillation of simple harmonic motion):

$$\text{Time of contact} = \frac{b}{V}\left[\pi\sqrt{\mu + \frac{1}{2}} - \frac{1}{2}\right]$$

Method of Obtaining Maximum Compression or Unit Stress in the Rod During Impact

The maximum compression or unit stress in the rod during impact is obtained as follows: The total compression at any point in the rod during impact is always the sum of two values of F, one in the resultant wave going in one direction and one in the resultant wave going in the opposite direction. When the portion of wave with a compression equal to the *maximum* value of F strikes the fixed end and is reflected there, both waves will have this maximum value; the total compression at this point and at this instant is as great as can occur during the impact because at no other point or instant can we have two values

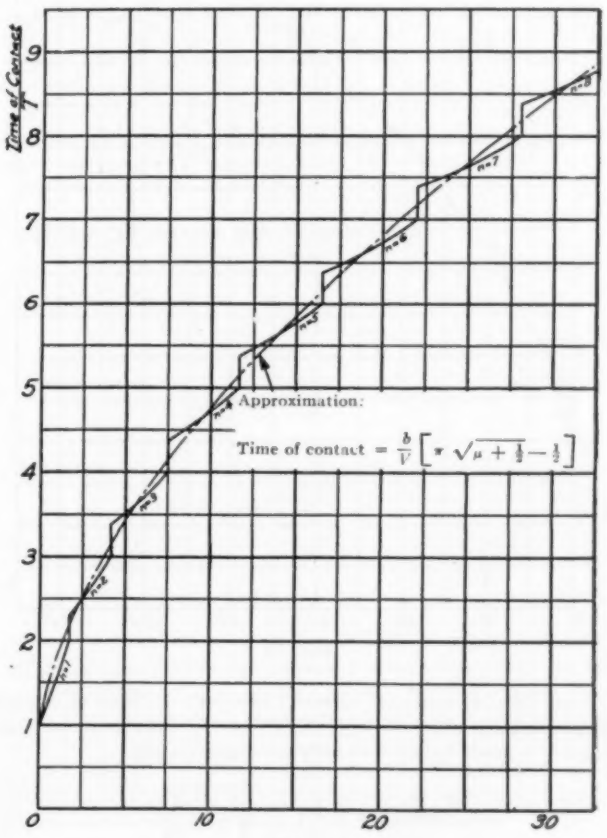


FIG. 18

of F greater than this.[13] Hence the maximum compression during impact occurs in general at the fixed end and is equal to twice the maximum value of F. Now F varies in something the same fashion as f', shown in Fig. 17, rising to a peak at the beginning of each interval and then decreasing. Hence in searching for the maximum value of F we need consider only the values at the beginning of each interval; the general expression for these is obtained by substituting $t/T = n$ in [18], which makes $z_m = 2/\mu(m - n)$. This gives us an expression for F/f_0 as a function of μ and n only. It is then only a question of what

[12] That this is not merely a convention can be shown as follows: $m! = \frac{(m+1)!}{m+1}$, and if $m = 0$, we have $0! = \frac{(0+1)!}{0+1} = 1$.

[13] F may have two equal maximum values an interval of T apart which combine at one instant at the end struck.

value of n to use to get the greatest F for any particular value of μ. This is solved by plotting F/f_0 against μ for *each* value of n (which must be an integer) and selecting only those portions of the resulting curves which rise above their fellow-curves. This has been done in Fig. 19, except that $2F/f_0$ is plotted instead of F/f_0, so that the resulting curves give the maximum compression in the rod for any value of μ.

The empirical curves Max. $f = f_0\,(1 + \sqrt{\mu + 1})$ and Max. $f = f_0(1 + \sqrt{\mu})$, calculated as above, practically form an en-

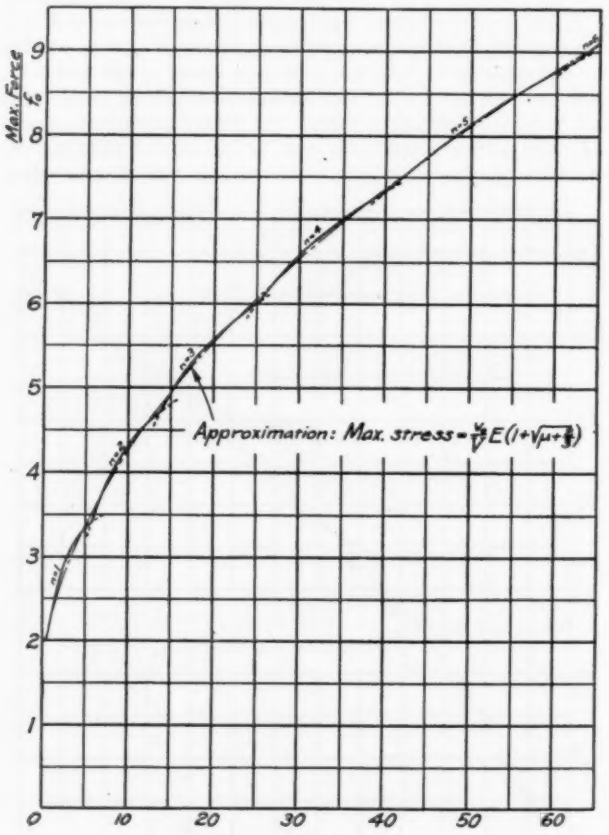


FIG. 19

velope for this series of curves. The first of these is a good approximation which is on the safe side, but the closest approximation is given by a curve between these, namely

$$\text{Max. } f = f_0\left(1 + \sqrt{\mu + \frac{2}{3}}\right)$$

as shown on the chart.

The velocity of the mass M at separation (and from this the lost-energy ratio) can be obtained by taking values of μ, t/T, and n from Fig. 18 and substituting them in [18], and then in [16]. The results of such calculations are plotted on Fig. 20. A rough approximation to this very irregular function, calculated as above and also plotted on the chart, is:

$$\text{Velocity of } M \text{ at separation } = v_0\sqrt{1 - \frac{2}{\mu^{1.3} + 3}}$$

With this approximation the lost-energy ratio, obtained as before, is $\dfrac{2}{\mu^{1.3} + 3}$.

EFFECT OF A GRADUAL CHANGE IN CROSS-SECTION OR MATERIAL OF BAR[14]

A gradual change in section or material may be considered to be brought about in a series of small, sudden steps such as previously considered. As a wave is transmitted through such a series of steps there will be reflection at each step. The reflected waves must travel back through the steps, and in so doing give rise to doubly reflected waves, and these to trebly reflected waves, and so on. The original transmitted wave combines with all waves due to an even number of reflections to form a resultant forward-moving wave, which we shall call the resultant transmitted wave f_t. The waves due to an odd number of reflections combine to form a backward-moving wave which we shall call the resultant reflected wave f_r.

SMALL CHANGE WITH COMPARATIVELY LONG WAVES

If the change is a comparatively small one (say, $4 > r > 1/4$, where r is defined as before), the waves due to successive reflections quickly approach zero. If a long, uniform wave passes through such a change, as, for instance, the change in area shown in Fig. 21, the conditions in the varying portion change at first due to the successive reflections, but quickly approach a stable condition as the successive reflections become smaller. After a length of wave equal to several times the length of the varying portion has passed, we can consider that conditions in this portion of the bar are constant with time. If they are constant with time there are no accelerations or inertia forces, and equilibrium requires that the total compression must be constant along the length of the varying portion. The total velocity of particles must also be constant along the length, as otherwise the compression would change with time. Consequently the compression of the resultant reflected wave f_r at the beginning of the varying portion and of the resultant

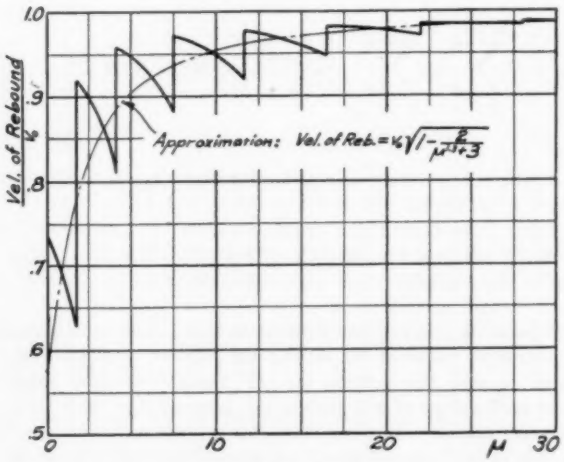


FIG. 20

transmitted wave f_t at the end of the varying portion are found by the same reasoning as for a sudden change, and are given by [9] and [10]. The magnitudes of f_r and f_t at any intermediate section such as b, Fig. 21, are found from the conditions that the resultant compression remains constant, or

$$f_t + f_r = f + \frac{r - 1}{r + 1} f$$

[14] It is assumed that the waves are started on a plane perpendicular to the axis of the rod and remain so. This will not involve great error if the changes in section are gradual.

and that the resultant velocity remains constant, or

$$\frac{f_t - f_r}{\sqrt{m_b k_b}} = \frac{f - \dfrac{r-1}{r+1} f}{\sqrt{m_a k_a}}$$

where m_b and k_b are the constants at section b. Solving for f_t and f_r and letting $\sqrt{\dfrac{m_b k_b}{m_a k_a}} = r_b$,

$$f_t = \frac{r + r_b}{r+1} f \quad \text{and} \quad f_r = \frac{r - r_b}{r+1} f$$

It is well known that when a bar with a *sudden* change in cross-section (such as shown by the broken lines in Fig. 21) is under compression, the material at the corners (shown shaded in the figure) is more or less dead material which takes little part in the action. It may be presumed, therefore, that in the case of a sudden change of cross-section, conditions are actually of the nature of those just described,[15] instead of as previously obtained under the assumption that stresses were uniformly distributed over the cross-section at all points.

Gradual Change With Comparatively Short Waves

Let us consider that portion of the wave which is directly transmitted without any reflection, calling its compression f_t'. We can consider that the change is brought about in n successive steps, as suggested by the broken lines in Fig. 22, where n is to be made infinite. We can take these steps such that, letting $r = \sqrt{m_c k_c / m_a k_a}$ as before:

$$r^{1/n} = \sqrt{\frac{m_1 k_1}{m_a k_a}} = \sqrt{\frac{m_2 k_2}{m_1 k_1}} = \sqrt{\frac{m_3 k_3}{m_2 k_2}}, \text{ etc.}$$

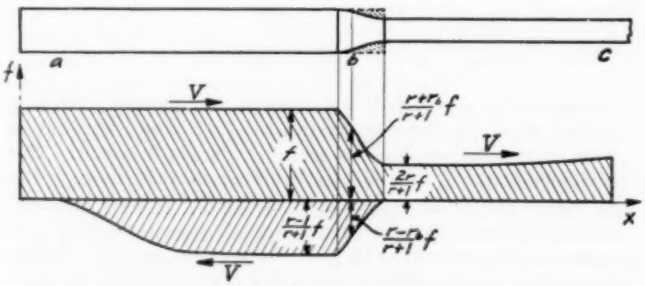


Fig. 21

If a wave with the compression f at section a travels to section c, the compression in the *directly* transmitted wave will, from [9], be multiplied by $2r^{1/n}/(r^{1/n}+1)$ at each step, so that after passing all n steps, the final value of the directly transmitted wave is

$$f_t' = \left[\frac{2r^{1/n}}{r^{1/n}+1}\right]^n f = \left[\frac{2}{2 + (r^{1/n} - 1)}\right]^n rf = \frac{rf}{[1 + \frac{1}{2}(r^{1/n} - 1)]^n}$$

When n becomes infinite, $(r^{1/n} - 1)$ becomes infinitesimal, so we can write $[1 + \frac{1}{2}(r^{1/n} - 1)]^n = [1 + (r^{1/n} - 1)]^{n/2} = [r^{1/n}]^{n/2} = \sqrt{r}$.

Hence:

$$f_t' = \sqrt{r}\, f \qquad [19]$$

If the change is one of cross-section only, and the cross-sections are all geometrically similar, f_t' will therefore vary directly as the linear dimensions of the cross-section, as shown in Fig. 23. The intensity of stress, on the other hand, will vary *inversely* as the linear dimensions of the cross-section. A wave moving toward a pointed end would thus theoretically produce an infinite stress at the end. In a conical bar ft' varies directly as the distance from the apex, and the unit stress inversely as this distance.

The reflected waves, whatever they are, obviously start from zero at the front of the transmitted wave, as suggested by the line gh, Fig. 23. Consequently the *resultant* transmitted wave f_t must have the value given by [19] at the front of the wave (as suggested by the broken lines in Fig. 23), and the resultant reflected wave must have the value zero at the front of the wave.

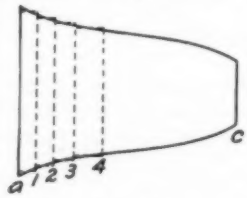


Fig. 22

If the wave is very short, such as the portion bc of the wave shown in Fig. 23, the resultant transmitted wave will be nearly as given by [19] throughout, as well as at the front.[16] Moreover the resultant reflected wave will be of minor importance in this case, as can be shown by considering the energies in the waves. The energy in the resultant transmitted wave as it passes any section c, from [1] and [19], will be nearly

$$\frac{(\sqrt{r}\, f)^2 L_c}{k_c} = \frac{rf^2 L_c}{k_c} = \frac{\sqrt{\dfrac{m_c k_c}{m_a k_a}}\, f^2 \sqrt{\dfrac{k_c}{m_c}}\, t}{k_c} = \frac{f^2 \sqrt{\dfrac{k_a}{m_a}}\, t}{k_a} = \frac{f^2 L_a}{k_a}$$

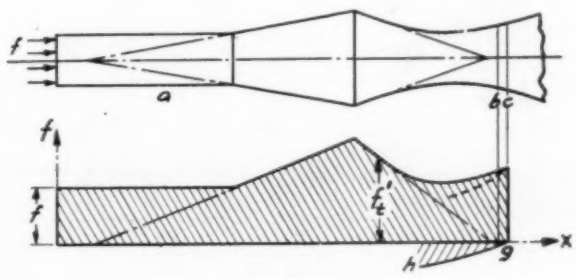


Fig. 23

which is the energy of the original wave starting at a. As the total energy cannot change, the energy in the resultant reflected wave must be small (it has been pointed out before that the energies of waves going in opposite directions can be calculated independently). It can be shown that the foregoing reasoning applies also to a series of short waves of *opposite* sign, such as are produced by a harmonic force of high frequency.

General Case of a Gradual Change

To study the problem in more detail we must use the differential equation of the rod. Let $f = (f_t + f_r)$ represent the total compression at any section, u the longitudinal displacement of the section, and t the time. In general f varies both with time and with x. The rate of variation along the length (that is with x) is represented by $\partial f/\partial x$. The difference between the compressions at the two ends of an elemental length of the bar, of length dx, is therefore $(\partial f/\partial x)dx$. This represents an unbalanced force acting on the element, which must equal the mass

[15] However, we are still making the approximate assumption that wave fronts are perpendicular to the axis of the rod.

[16] It will be the same whether the portion bc is alone or at the front of a long wave, because the effect of the portion of the wave behind section b cannot catch up with the portion bc.

of the element times its acceleration $(mdx)\frac{\partial^2 u}{\partial t^2}$. Hence, taking the direction in which x is measured as positive,

$$-\frac{\partial f}{\partial x}dx = mdx\frac{\partial^2 u}{\partial t^2}, \text{ or } -\frac{\partial f}{\partial x} = m\frac{\partial^2 u}{\partial t^2} \quad [20]$$

The unit strain is

$$\frac{\partial u}{\partial x} = -\frac{f}{k} \quad [21]$$

The negative sign is due to the fact that we have taken f as compression. In these equations m and k are of course no longer constants but are functions of x. From [21],

$$f = -k\frac{\partial u}{\partial x}$$

Differentiating with respect to x,

$$\frac{\partial f}{\partial x} = -k\frac{\partial^2 u}{\partial x^2} - \frac{dk}{dx}\cdot\frac{\partial u}{\partial x}$$

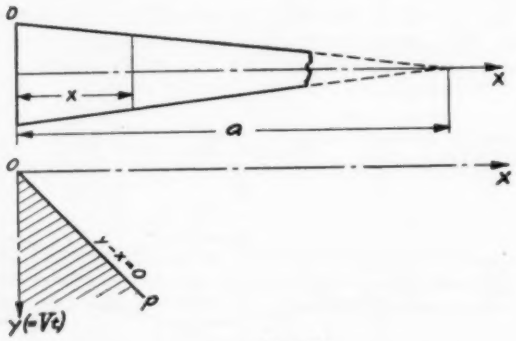


FIG. 24

and substituting this in [20] we have the general differential equation of the bar

$$k\frac{\partial^2 u}{\partial x^2} + \frac{dk}{dx}\cdot\frac{\partial x}{\partial x} = m\frac{\partial^2 u}{\partial t^2}$$

or

$$\frac{m}{k}\frac{\partial^2 u}{\partial t^2} = \frac{\partial^2 u}{\partial x^2} + \frac{1}{k}\frac{dk}{dx}\cdot\frac{\partial u}{\partial x} \quad [22]$$

Also, from [21],

$$f_i + f_r = f = -k\frac{\partial u}{\partial x}$$

and using [6],

$$f_i - f_r = \sqrt{mk}\,v = \sqrt{mk}\frac{\partial u}{\partial t} \quad [23]$$

from which

$$f_i = -\frac{k}{2}\left(\frac{\partial u}{\partial x} - \sqrt{\frac{m}{k}}\frac{\partial u}{\partial t}\right)$$

$$f_r = -\frac{k}{2}\left(\frac{\partial u}{\partial x} + \sqrt{\frac{m}{k}}\frac{\partial u}{\partial t}\right)$$

CONICAL BAR

As an example of the application of these equations, consider the case of a conical bar of homogeneous material, dimensioned as shown in Fig. 24. If m_o and k_o are the constants at section o, then the constants at any other section will be[17]

$$m = \frac{(a-x)^2}{a^2}m_0 = \left(1-\frac{x}{a}\right)^2 m_0; \quad k = \left(1-\frac{x}{a}\right)^2 k_0. \quad [24]$$

In this case $V = \sqrt{\frac{k}{m}} = \sqrt{\frac{k_0}{m_0}}$ is a constant. Using the notation $y = Vt$, Equation [22] becomes:

$$\frac{\partial^2 u}{\partial y^2} = \frac{\partial^2 u}{\partial x^2} - \frac{2}{a-x}\frac{\partial u}{\partial x} \quad [25]$$

The general solution of this can be expressed as follows:

$$u = \frac{\phi(y-x) + \psi(y+x)}{1-\frac{x}{a}} \quad [26]$$

where ϕ and ψ are any functions; they must of course be determined to fit the other conditions. Using Equations [23],

$$f = -k\frac{\partial u}{\partial x} = k_o\left[\left(1-\frac{x}{a}\right)(\phi'-\psi') - \frac{1}{a}(\phi+\psi)\right] \quad [27]$$

$$v = \frac{\partial u}{\partial t} = V\frac{\partial u}{\partial y} = V\frac{\phi'+\psi'}{1-\frac{x}{a}} \quad [28]$$

$$f_i = -\frac{k}{2}\left(\frac{\partial u}{\partial x} - \frac{\partial u}{\partial y}\right) = k_o\left[\left(1-\frac{x}{a}\right)\phi' - \frac{1}{2a}(\phi+\psi)\right] \quad [29]$$

$$f_r = -\frac{k}{2}\left(\frac{\partial u}{\partial x} + \frac{\partial u}{\partial y}\right) = k_o\left[\left(\frac{x}{a}-1\right)\psi' - \frac{1}{2a}(\phi+\psi)\right] \quad [30]$$

where $\phi' = \frac{d\phi}{d(y-x)}$, $\phi'' = \frac{d^2\phi}{d(y-x)^2}$, etc.

Suppose for example that a constant compressive force f_o is applied at o. In determining the value of u in the resulting wave an xy diagram, such as shown in Fig. 24, is useful. This is similar to the xt diagrams used previously but here $y = Vt$ represents the distance traveled by the wave. If we start measuring the time when the force is applied then the line $y = x$ represents the position of the wave front at any instant. Points in the area xop represent points along the rod before the wave reaches them, that is, beyond the wave front. The portion in which we are interested is the shaded area poy, which represents points within the wave. There are no discontinuities within this region and we can avoid discussion of discontinuities by making use of our knowledge, previously obtained, of the conditions at the wave front—that is, along the line op. We need only satisfy the known conditions along the lines op and oy, and the condition for variation between; the latter condition, represented by [25], has already been satisfied by using [26]. The condition along oy—that is, at the end of the bar—is that $(f)_{x=o} = f_o$. From [19] the condition along op, at the wave front, is that $(f)_{y=x} = \frac{a-x}{a}f_0 = \left(1-\frac{x}{a}\right)f_0$. It is found that ψ may be zero in this case, so that these conditions become

$$k_0\left[\phi'(y) - \frac{\phi(y)}{a}\right] = f_0 \quad [31]$$

[17] The following analysis applies to any case where [24] holds whether the bar is actually conical or not. It is assumed, as before, that cross-sections and the rate of change, or angle of taper, are small.

$$k_0\left[\left(1-\frac{x}{a}\right)\phi'(o)-\frac{\phi(o)}{a}\right]=\left(1-\frac{x}{a}\right)f_o \quad \ldots\ldots [32]$$

Solving the differential Equation [31], and using [32] to determine the resulting constant of integration,[18]

$$\phi=\frac{f_0 a}{k_0}\left(e^{\frac{y-x}{a}}-1\right),\ \psi=0 \quad \ldots\ldots [33]$$

Substituting this in [24],

$$f=\left(1-\frac{x}{a}e^{\frac{y-x}{a}}\right)f_0$$

If a is made very large we approach the condition in a uniform bar, $f = f_0$.

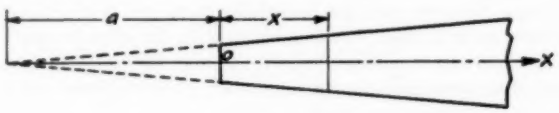


FIG. 25

If the force is removed after a length of time equal to L/V, this is equivalent to superimposing on the above wave a negative wave produced by a force $-f_0$, and having $(y - L)$ instead of y as the distance traveled by its wave front. This gives

$$(y-L)<x<y\ :f=\left(1-\frac{x}{a}e^{\frac{y-x}{a}}\right)f_0, \text{ as above}$$

$$0<x<(y-L)\ :f=\frac{x}{a}\left(e^{\frac{y-x-l}{a}}-e^{\frac{y-x}{a}}\right)f_0$$

The latter expression represents a sort of tail strung out behind the main part of the wave, due to reflections. All these waves are of course made up of two waves, one with compression f_i moving forward, and one with compression f_r moving backward. These can be investigated by substituting [33] in [29] and [30].

If a is negative we have the case shown in Fig. 25. The above formulas can be used for this case also, using a *negative* value for a.

WAVES IN MATERIALS WHICH DO NOT FOLLOW HOOKE'S LAW

Suppose that a uniform bar is made of a material such that, in a tensile or compressive test, we obtain a load vs. unit strain curve as shown in Fig. 26. The slope of this curve corresponds to our stiffness constant k. If a constant force f_1 is applied to the end of the bar, a wave with this compression will move along the bar with a velocity $V_1 = \sqrt{k_1/m}$. If now another force f_2 is superposed on f_1, the bar, being already under a compression f_1, will follow the load-strain curve with slope k_2, and the wave started by f_2 will move along the bar with the velocity $V_2 = \sqrt{k_2/m}$. If the two forces are applied at the same time the action will be the same, and the complete wave front will present the appearance shown in Fig. 26(*b*), with the gap between the two parts of the front increasing with time.

If we imagine k_2 gradually decreased we see that when it becomes zero the second portion of the wave, having zero velocity, will never get started. This means that, if a force exceeding the yield-point force is applied to a bar of steel having a sharp, horizontal yield point, a wave will move along the rod with a compression equal to the yield-point force only. The work done by the remainder of the force is obviously absorbed as plastic work around the point of application.

[18] It would ordinarily require two arbitrary constants to satisfy [32]; the fact that it can be done with only one is a check on our previous analysis of the conditions at the wave front.

If the load-strain curve of the material has a gradual decrease of slope with increasing stress (as in the case of cast iron, for example) it is evident, from the above reasoning, that the wave front will have a gradual backward slope, as in Fig. 27, increasing with time. Its exact shape will depend on the stress-strain diagram. The simplest result would be to have it a straight line, as indicated by the broken line in Fig. 27. In order for this to be the case the distance $\frac{t}{\sqrt{m}}\left(\sqrt{k_0}-\sqrt{k}\right)$ in the figure would have to be proportional to f as well as to t, as for instance,

$$\frac{t}{\sqrt{m}}\left(\sqrt{k_0}-\sqrt{k}\right)=\frac{Ct}{\sqrt{mk_0}}f \quad \ldots\ldots [34]$$

where C is a constant to be determined. Substituting $k = df/de$ in this equation, and integrating and determining the constant of integration from the condition $f = 0$, $e = 0$, we find:

$$f=\frac{k_0 e}{Ce+1} \quad \ldots\ldots [35]$$

This equation represents the experimental results obtained with such materials as cast iron and annealed copper very well.

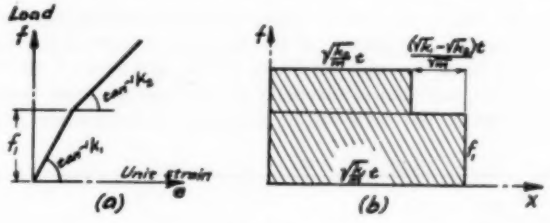


FIG. 26

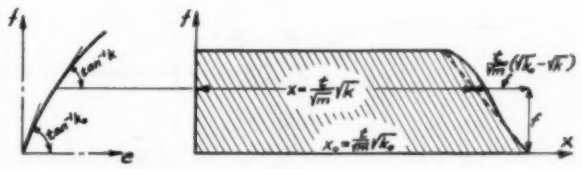


FIG. 27

The constants k_0 and C can be found from any two experimental values of f and e: f_1, e_1, and f_2, e_2 (where f_1 is preferably the largest load used and f_2 about half the largest). Substituting these in [35] and solving for k_0 and C,[19]

$$k_0=\frac{f_1 f_2(e_2-e_1)}{e_1 e_2(f_2-f_1)},\quad C=\frac{f_1 e_2-f_2 e_1}{e_1 e_2(f_2-f_1)} \quad \ldots\ldots [36]$$

Hence we may assume that in bars of such materials the wave produced by a suddenly applied constant force will have, on the f, x diagram, an inclined front, the slope of which is $Ct/\sqrt{mk_0}$ (from [34]), where k_0 and C are given by [36]. If, upon *unloading*, the material shows a straight load-strain line having the slope k_0—as seems to be approximately the rule—the rear of the wave will evidently be vertical in the f, x diagram and will move at the velocity $\sqrt{k_0/m}$, so that the total length of the wave will remain constant.

COMPRESSION WAVES IN HELICAL SPRINGS AND COLUMNS OF LIQUIDS

All the foregoing theory applies without change to longitudinal

[19] k_0 should be the slope of the experimental f, e curve at the bottom but it is difficult to measure this accurately.

waves in helical springs. The stiffness constant k in this case is the spring constant of a unit length of the spring.

That portion of the foregoing which deals with prismatical bars can be applied to longitudinal pressure waves in inclosed columns of gases and liquids, of uniform section. It also applies approximately to the longitudinal propagation of surface waves in liquids in long uniform tanks; in this case, however, the lateral motions involved at the ends of the waves produce a large frictional effect which modifies conditions at these points.

In the case of columns of gases in rigid pipes, k is the area of the cross-section times the modulus of volume elasticity. The latter

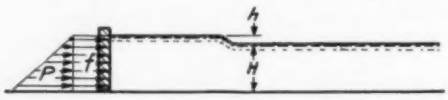


FIG. 28

for adiabatic conditions, is λp, where λ is the ratio of the specific heats of the gas under constant pressure and under constant volume ($\lambda = 1.4$ for air) and p is the unit pressure of the gas. For very long waves there will be some deviation from adiabatic conditions in the middle of the wave, which will complicate the problem.

In the case of columns of liquids in pipes, the flexibility of the walls must be considered. If p is the unit pressure of the liquid, α the ratio of the wall thickness to the diameter of the pipe, and E the modulus of elasticity of the wall material, then the hoop stress in the walls will be $p/2\alpha$ and the hoop strain $p/2\alpha E$.

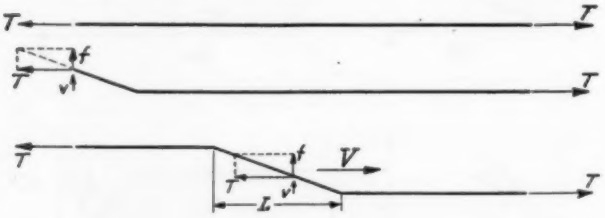


FIG. 29

This produces a unit increase in volume inside the pipe of

$$\left(1 + \frac{p}{2\alpha E}\right)^2 - 1 = \frac{p}{\alpha E}$$

approximately. This has the same effect as a unit decrease in volume of the liquid. This decrease must be added to that due to the compressibility of the liquid, which is p/E_v, where E_v is the modulus of volume elasticity of the liquid. The stiffness constant k, being the longitudinal force per unit longitudinal shortening, will be equal to the area of the cross-section times

$$\frac{p}{\frac{p}{E_v} + \frac{p}{\alpha E}} = \frac{E_v}{1 + \frac{E_v}{\alpha E}}$$

The inertia of the pipe walls and the stiffening effect of the portion of the pipe just outside the wave will modify conditions near the ends of the waves and will be of importance if the waves are very short.

The case of surface waves in long tanks is illustrated by Fig. 28. If a uniformly distributed longitudinal force f (superposed on the total static pressure, P) is exerted on the water at a certain point, say through a movable bulkhead as shown in the figure, the water will rise a distance h, such that

$$\frac{(h + H)^2}{H^2} = \frac{P + f}{P}$$

where H is the original depth. Assuming that the volume of water remains the same, there will be a unit shortening in the lengthwise direction, due to this increase in depth, of h/H. And k, the force per unit shortening, will be $f/(h/H)$. Using the above equation this reduces to $k = 2P$, approximately (if h/H is small).

Applications to Other Types of Waves

The foregoing theory can be applied also to the case of small *lateral* waves in a stretched chain or flexible rope. In this case f is to be taken as a lateral force acting on the rope, superposed on the initial constant tensile force T, and v is to be taken as the resulting lateral velocity of particles of the rope, as indicated in Fig. 29. The lateral displacement per unit length of rope can be taken as f/T, if f is small compared to T. In this case k, being the force per unit *lateral* displacement, is $f/(f/T) = T$. In this, as in all the preceding cases, m is the same as originally defined, the mass per unit length. It may be noted, in this last case, that superposing waves may be in different planes and must then be combined vectorially.

All the preceding theory can be applied to the case of torsion waves in bars of uniform or variable section (with even more exactitude than for the original case of longitudinal pressure waves, as there is no lateral expansion here to affect the conditions at the ends of waves). In this case f is to be interpreted as the torque applied to the bar (or the internal torque on sections within the wave) and v is to be interpreted as the resulting angular velocity of sections of the bar. For this case m must be taken as the moment of inertia of mass of the bar (about the longitudinal axis) per unit length of the bar; and k will be the torque required to twist unit length of the bar one radian, which can be determined by the usual methods.

Discussion

THOS. C. RATHBONE.[20] Professor Donnell has opened up a most fascinating subject which has practical application in many fields of mechanical engineering. The transmission of impulses in materials is usually associated with the case of relatively small parts involving extremely short periods of time. When these studies are applied to large central-station turbines, however, both the time intervals and distances become appreciable. For example, the distance between the bearings of a large generator rotor may reach 35 ft. or more, and a length of 75 ft. would not be unusual between the turbine thrust bearing and the outboard bearing. These units are normally mounted on a continuous steel foundation. Although the forces at the bearing due to unbalance in the rotor are usually considered as acting in a plane normal to the turbine axis, it is not unusual to find axial disturbances.

In his investigation of the vibration phenomena, the writer has made use of stroboscopic vibrometers to explore in some detail the characteristic of vibration of large installations as a whole, with particular regard to the phase relation between the vibratory motion and references on the rotor. These studies were directed toward arriving at a method of correcting unbalance on a complicated structure where the vibratory disturbances at the several bearings are transmitted throughout the unit and affect each other. On such an installation the motion observed at any bearing represents a composite which includes disturbances emanating from the other bearings.

Many unusual conditions were disclosed. For example, it was found that the same unbalance in a rotor would produce a

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given set of vibratory characteristics when rotating in the opposite direction at the same speed. It was also found that the so-called "angle of lag," between a point in the vibratory motion of the bearing and a datum on the rotating spindle, would vary through two full circles or more in going through a speed range up to 2000 r.p.m. These results are entirely at variance with the commonly accepted notion that the "angle of lag" at the critical speed is 90 deg., becoming 180 deg. at an infinite speed.

On these very large units the phase relation of the vibratory motion at the various bearings is affected by the length of time required for an impulse to travel from one point to another. For example, on an installation of the size mentioned, running at 1800 r.p.m., the spindle will turn through approximately 50 deg. during the time required for an impulse to travel from the thrust bearing, for example, to the outboard bearing. Such large angles are of course very appreciable when attempting to analyze the vibratory characteristics in order to determine the proper locations for balance corrector weights.

The velocity of propagation in steel is about one-third higher than in concrete. It is interesting to note that a disturbance at the thrust end of such a unit would reach the outboard end through the steel and iron turbine structure about 17 deg. ahead of the wave coming through the concrete foundation.

The study of resonances in a large turbine-generator installation from the viewpoint of reflections and standing waves offers an interesting field of investigation. The modes of vibration commonly encountered on such structures are very complex and cannot be predicted by ordinary methods.

APM-52-15

Steady Forced Vibration as Influenced by Damping

An Approximate Solution of the Steady Forced Vibration of a System of One Degree of Freedom Under the Influence of Various Types of Damping

By LYDIK S. JACOBSEN,[1] STANFORD UNIVERSITY, CALIF.

In this paper the author presents a general method of obtaining approximate solutions of the steady forced vibration of a damped system of one degree of freedom for the case of sinusoidally varying disturbing forces.

The approximation consists in expressing all the damping terms of the original differential equation by a single equivalent damping term, proportional to the first power of the velocity of motion.

In the case of a system influenced by a centrifugal disturbing force and damped by constant friction and by friction proportional to the first power of the velocity, experimental evidence is in good agreement with the approximate solution.

While the paper deals solely with a system of one degree of freedom, there is no reason aside from complexity why the approximate solution cannot be applied to systems of several degrees of freedom.



THE analytical treatment of forced vibrations occurring in mechanical structures usually assumes that the dissipative forces can be regarded as being proportional to the first power of the velocity of motion. By making this postulate, the investigator arbitrarily limits his analysis to apply to physical problems of a special class, but he gains a commensurate amount of ease and comfort in his mathematical expression for the resulting vibratory motions. If, however, the problem is to investigate the forced vibrations of structures in which only a small proportion of the dissipated energy is due to viscous forces, it becomes necessary to make approximations in such a way that actual physical conditions are simulated and that the resulting differential equations of motion do not defy easy solutions.

The experimental records of free vibrations of mechanical structures very fortunately furnish us with information about the type of dissipative forces that predominate in particular cases. An examination of a free-vibration record will disclose the extent to which viscous damping occurs, it will indicate whether or not constant dissipative forces are in action, and it will, if closely scrutinized, betray the existence of damping forces proportional to the nth power of the velocity of motion. At any rate, it will usually be possible to fit an empirical curve to the free-vibration curve if combinations of constant friction, viscous friction, and friction proportional to the nth power of the velocity be at our disposal.

As examples involving the three types of damping, the following may be mentioned:

a Constant friction occurs in nearly all mechanical structures; the greater the number of friction joints, the more predominant the constant-friction term in the equation of motion.

b Viscous friction, proportional to the first power of the velocity, is usually present to a greater or smaller extent in all mechanical structures. Air damping at small velocities belongs to this type.

c Friction, proportional to the nth power of the velocity, occurs in hydraulic systems and in air-damped systems where the velocity of motion is relatively large.

Historical Notes

While the solution of the free vibration of a system with constant friction is well known, the problem of the forced vibration of the same system involves so much difficulty that a rigorous analytical solution has not been obtained until very recently. A paper by Herr W. Eckolt[2] considers in detail, but from a semi-graphical point of view, the wave shape of a forced vibration with constant damping. His method involves a great number of steps that are not sufficiently obvious, or well explained, to be of practical use for engineering purposes. During the summer of 1929, after the first draft of this paper had been made, and submitted to Dr. S. Timoshenko for criticism, the author learned that Dr. J. P. Den Hartog[3] had just succeeded in obtaining an exact analytical solution of the forced vibration with constant friction and also of the forced vibration with constant friction and with first-power-velocity damping. It is, indeed, very fortunate for the approximate method developed in this paper that Dr. Den Hartog's exact solution is available. A comparison of the results obtained by applying the exact and the approximate method to four special cases discloses the fact that there is excellent numerical agreement between them. Moreover, the exact as well as the approximate solution is in good accord with experimental evidence for the four special cases mentioned above.

The solution of the free vibration with damping proportional to the square of the velocity is well known, though not very simple. Professor Milne's tables,[4] recently finished, give a complete and rigorous solution of the problem. The case of damping proportional to the nth power of the velocity has not yet been solved analytically; it is, however, as is pointed out by Professor Milne, possible to get a solution for a certain region by considering a combination of first- and second-power-velocity damping.

Simplifying Assumptions

An approximate solution of a physical problem usually requires that uneven emphasis be placed upon a trustworthy

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[2] *Zeitschrift für Technische Physik*, vol. 7 (1926), p. 226.

[3] *Philosophical Magazine*, 1930.

[4] W. E. Milne, "Damped Vibrations," Univ. of Oregon Publications, No. 1 (1923) and No. 2 (1929).

answer to the various aspects of the physical problem. In the following approximate solution of steady forced vibrations, primary interest will be attached to the question of amplitude of the resulting motion. It is, of course, realized that the question of wave form enters in and, in the case of large damping, may influence the amplitude appreciably, but as a first approximation it will be assumed that a displacement of sinusoidal form will yield results that cannot, except in extreme cases, be much at variance with actual phenomena.

The approximate solution centers on the simple and well-known solution of a vibrating system of one degree of freedom, influenced by a sinusoidally varying disturbing force and by a viscous friction proportional to the first power of the velocity of motion. In considering other types of damping it will be assumed that they all can be reduced to an equivalent viscous damping involving the term dx/dt and a coefficient which is a function of the maximum displacement and the angular velocity of the motion. The reason for doing this is obviously simplicity; its justification as an approximation lies in the fact that we are considering the steady state of the forced vibration, so that the coefficient of dx/dt becomes a constant for the constant value of the maximum displacement and the angular velocity under observation.

In defining the term "equivalent viscous damping," it is necessary to state the criterion upon which equivalence is established. It must be admitted that it is somewhat arbitrary to say that the criterion of equivalent dissipative work done during a cycle is a better criterion than, for instance, that of equivalent time average of the damping force. Intuitively, however, we recognize the great importance of having a general reckoning in regard to work done at the end of every cycle, so we feel warranted in pdstulating as a criterion of equivalence of viscous damping force the equality of dissipative work done during a cycle by the real force and by the equivalent viscous force. Before this criterion had been adopted, the approximate solution of the forced vibration of a system with constant friction and velocity damping had been carried out for four special cases on three different criteria of equivalent viscous damping force. The results were then compared to experimental evidence, with the remarkable outcome that the three criteria gave approximately the same agreement with experiment in all regions of the frequency-amplitude plane, with the exception of the one of resonance. At or near resonance the criterion of dissipative work done during a cycle was far superior to the other two in regard to numerical agreement with experiment. Furthermore, a comparison of Mr. Den Hartog's exact solution with the approximate one gave a very gratifying agreement, provided that the criterion of dissipative work was used for establishing the equivalent viscous damping.

I--THEORETICAL WORK

Derivation of the General Equation

Consider a vibrating system of one degree of freedom. The mass is m, and the spring factor is k. Let the system be acted upon by a sinusoidally varying disturbing force of the constant circular frequency ω, and of a constant maximum magnitude Q. If the system be influenced by a viscous damping equal to a constant c multiplied by the velocity dx/dt, the differential equation of motion is:

$$m\frac{d^2x}{dt^2} + c\frac{dx}{dt} + kx = Q\sin\omega t \quad \ldots\ldots\ldots\ldots [1]$$

The solution of this equation is simple and well known. Since we are limiting this investigation to the steady forced vibration, we are interested in the particular integral only. This integral is:

$$x = \frac{Q}{\sqrt{(k - m\omega^2)^2 + c^2\omega^2}}\sin(\omega t - \psi) = X\sin(\omega t - \psi) \quad \ldots [2]$$

where

$$\tan\psi = \frac{c\omega}{k - m\omega^2}$$

Let us now consider a slightly different case where the same system is influenced by a damping force which is proportional to the nth power of the velocity; the equation of motion is then:

$$m\frac{d^2x}{dt^2} \pm c_n\left(\frac{dx}{dt}\right)^n + kx = Q\sin\omega t \quad \ldots\ldots\ldots [3]$$

The plus or minus sign before the damping term indicates that the force always opposes the motion. In other words, the sign of the damping term must always be the same as that of dx/dt. Let us now replace the damping term in Equation [3] by an equivalent viscous damping term

$$c_1\frac{dx}{dt}$$

where c_1 is to be determined by the criterion of equivalent dissipative work done during a cycle,

$$w = 4c_n\int_0^{\frac{\pi}{2\omega}}\left(\frac{dx}{dt}\right)^n dx = 4c_1\int_0^{\frac{\pi}{2\omega}}\frac{dx}{dt}\,dx$$

If we retain our previous assumption that the effect of the friction on the wave shape of the displacement is negligible, we can perform the integration above since

$$x = X\sin(\omega t - \psi)$$

Consequently

$$c_n\int_0^{\frac{\pi}{2\omega}} X^{n+1}\,\omega^{n+1}\cos^{n+1}\omega t\,dt = c_1\int_0^{\frac{\pi}{2\omega}} X^2\omega^2\cos^2\omega t\,dt$$

giving

$$c_n X^{n-1}\,\omega^{n-1}\int_0^{\frac{\pi}{2\omega}}\cos^{n+1}\omega t\,dt = \frac{\pi}{4}c_1$$

The integral above is most generally expressed as follows:

$$\int_0^{\frac{\pi}{2\omega}}\cos^{n+1}\omega t\,dt = {}^1/_2\sqrt{\pi}\,\frac{\Gamma\left(\frac{n+2}{2}\right)}{\Gamma\left(\frac{n+3}{2}\right)}$$

where $\Gamma(n)$ means the gamma function for the value n. The equivalent friction coefficient is therefore

$$c_1 = c_n\frac{2}{\sqrt{\pi}}\,\frac{\Gamma\left(\frac{n+2}{2}\right)}{\Gamma\left(\frac{n+3}{2}\right)}\,X^{n-1}\,\omega^{n-1} = c_n\gamma X^{n-1}\,\omega^{n-1} \quad \ldots [4]$$

Fig. 1 gives in curve form the value of γ for values of n between zero and four.

Equation [3] can be replaced by

$$m\frac{d^2x}{dt^2} + (c_n\gamma X^{n-1}\,\omega^{n-1})\frac{dx}{dt} + kx = Q\sin\omega t \quad \ldots\ldots [5]$$

Since, as before, we are interested in the steady forced vibration only, X as well as ω is a constant, and the particular integral, in accordance with Equation [2], becomes:

$$x = \frac{Q}{\sqrt{(k - m\omega^2)^2 + c_n^2\gamma^2X^{2n-2}\omega^{2n}}} \sin(\omega t - \psi)$$
$$= X \sin(\omega t - \psi) \ldots\ldots [6]$$

where

$$\tan \psi = \frac{c_n\gamma X^{n-1}\omega^n}{k - m\omega^2}$$

In case n is equal to unity, this result coincides with the exact solution of the vibrating system with damping proportional to the first power of the velocity. If n be greater than zero but different from one, we have approximate solutions, the approximation being very close for values of n near unity, and less close for values of n near zero and for values of n somewhat greater than unity.

Fig. 2 shows a family of equivalent damping-force curves as

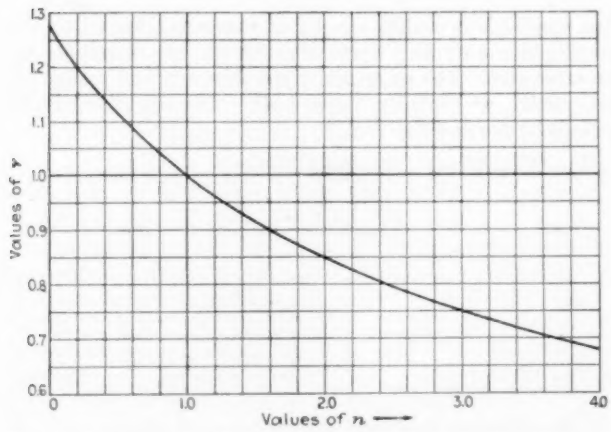


FIG. 1 SHOWING THE EQUIVALENCE FACTOR γ AS A FUNCTION OF THE VELOCITY-DAMPING POWER n

functions of time, assuming that the work per cycle is the same for each value of n. The curve $n = 1$ is the simple sine curve for which we have the exact solution of the equation of motion. As n decreases, the peak of the curve is lowered until, for $n = 0$, the curve degenerates into a straight line parallel to the time axis. This case corresponds to constant friction independent of the velocity. If the constant friction force be F, it is seen that

$$c_1 = \frac{4}{\pi} c_0 X^{-1}\omega^{-1} = \frac{4}{\pi} FX^{-1}\omega^{-1}$$

For values of n greater than unity, the peaks of the curves increase, so that for the limiting case of $n = \infty$ the curve becomes a straight line perpendicular to the time axis. For the case of damping proportional to the second power of the velocity, the following relation exists between the coefficient of the actual damping term, c_2, and the coefficient of the equivalent damping term, c_1,

$$c_1 = \frac{8}{3\pi} c_2 X\omega$$

AN ARBITRARY CRITERION OF VALIDITY

Before the physical significance of the above approximate solutions of the steady forced vibration is discussed, let us consider the criterion for a free periodic motion of a system of one degree of freedom and with damping proportional to the first power of the velocity. In this case the condition for a system having a periodic free vibration requires that:

$$\frac{k}{m} \geqq \frac{c_1^2}{4m^2} \ldots\ldots [7]$$

In the approximate solution of the steady forced vibration, the equivalent damping factor c_1 varies with the amplitude and the angular velocity. This means that, in the general case of n different from unity, the condition for a periodic free vibration cannot be satisfied for all regions of the frequency-amplitude plane. Interpreted physically, this fact indicates that there

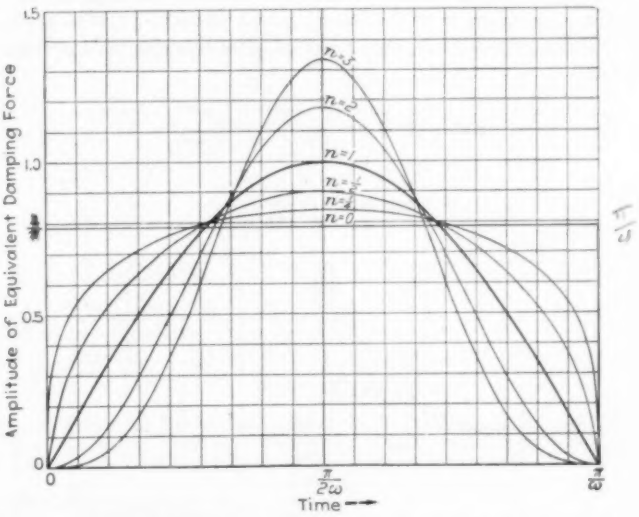


FIG. 2 SHOWING TIME-AMPLITUDE CURVES OF THE EQUIVALENT DAMPING FORCE FOR DIFFERENT VALUES OF THE VELOCITY-DAMPING POWER n

will be regions where our approximately equivalent dynamical vibrating system is not a free vibrating system. Manifestly, in these regions we must be careful in interpreting the results of the approximate solution. Thus, for the case of constant friction when n is zero, the value of c_1 will be large for simultaneously small values of the amplitude and the frequency; the condition of free periodicity will therefore not be fulfilled at small values of ω unless the damping factor F is extremely small. We therefore infer that in the general case an orderly motion, expressible by our approximate solution, will not occur until ω has reached a finite value dictated by the criterion of free periodicity expressed in Equation [7]. For values of n greater than zero but smaller than one, the same characteristics of the free-periodicity criterion prevail. At $n = 1$, our solution is, as said before, exact, and we have nothing unusual to be concerned about. When n is greater than unity, the aperiodic condition is sometimes reached for simultaneously large values of the amplitude and the angular frequency.

APPLICATIONS OF THE THEORY

In the following applications of the above-explained theory, it will be advisable to distinguish between two types of sinusoidal disturbing forces, the one of constant magnitude Q, and the one of a variable magnitude $P\omega^2$. The practical importance of considering the variable or the "centrifugal" disturbing force is derived from the fact that this type of disturbing force is realized approximately in a rotating, unbalanced machine, whose unbalanced mass moment is P, when the amplitude of the forced vibration is small compared to the radius of the unbalanced

moment. To facilitate a discussion as well as to make the curves applicable to similar vibrating systems with different physical constants, it will be found desirable to express all the constants by ratios. Let:

$$\left.\begin{aligned} &\frac{Q}{k} = q; \quad \frac{P}{m} = s; \quad \frac{k}{m} = \omega_c; \quad \frac{\omega}{\omega_c} = \tau \\ &f_n = c_n \gamma_n \omega_c{}^n k^{-n} Q^{n-1} \\ &h_n = c_n \gamma_n \omega_c{}^{2n-2} k^{-n} P^{n-1} \end{aligned}\right\} \quad \ldots\ldots\ldots [8]$$

γ_n to be found from Fig. 1.

The general Equation [6] for the maximum displacement and the phase angle ψ of the steady forced vibration, due to a disturbing force $Q \sin \omega t$, then becomes, after squaring and rearranging:

$$X^{2n} + X^2 \frac{q^{2n-2}(1-\tau^2)^2}{f_n{}^2 \tau^{2n}} - \frac{q^{2n}}{f_n{}^2 \tau^{2n}} = 0 \qquad (9)$$

$$\tan \psi = \frac{f_n \tau^n X^{n-1}}{q^{n-1}(1-\tau^2)}$$

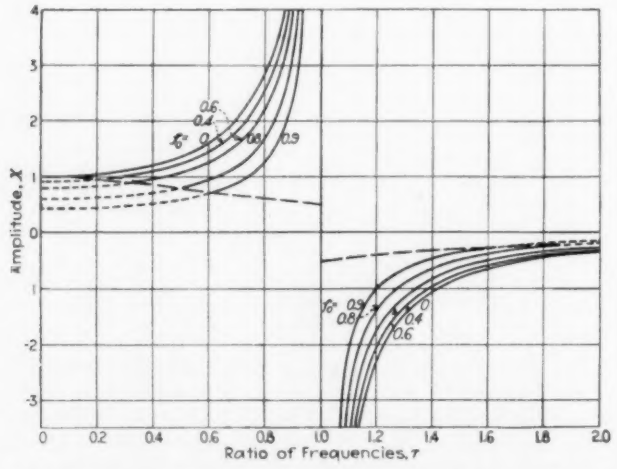


FIG. 3 SHOWING FREQUENCY-AMPLITUDE CURVES OF THE FORCED VIBRATION WITH COULOMB DAMPING AND "CONSTANT" DISTURBING FORCE, FOR DIFFERENT VALUES OF THE DAMPING FACTOR f_0

The arbitrary condition of free periodicity, Equation [7], requires that

$$4km \geqq c_n{}^2 \gamma_n{}^2 X^{2n-2} \omega^{2n-2} \text{ or } X\tau \leqq q \sqrt[n-1]{\frac{2}{f_n}}$$

If, instead of the "constant" disturbing force, the "centrifugal" disturbing force be used, the general equation assumes the form:

$$X^{2n} + X^2 \frac{s^{2n-2}(1-\tau^2)^2}{h_n{}^2 \tau^{2n}} - \frac{s^{2n}\tau^4}{h_n{}^2 \tau^{2n}} = 0 \qquad (10)$$

$$\tan \psi = \frac{h_n \tau^n X^{n-1}}{s^{n-1}(1-\tau^2)}$$

with the arbitrary free periodicity condition of

$$4km \geqq c_n{}^2 \gamma_n{}^2 X^{2n-2} \omega^{2n-2} \quad \text{or} \quad X\tau \leqq S \sqrt[n-1]{\frac{2}{h_n}}$$

In the general cases where n is greater than zero, Equations [9] and [10] must, of course, be solved by trial.

Let us see what the general equations become when different values are assigned to n.

Case I—A System With Viscous Damping

If $n = 1$, we have a system with damping proportional to the first power of the velocity; in other words, the solution is exact. From Fig. 1 we see that $\gamma = 1$.

(*a*) "Constant" Disturbing Force:

$$f_1 = c_1 \omega_c k^{-1} = \frac{c_1}{m\omega_c}$$

The general Equation [9] then becomes

$$\left.\begin{aligned} &X^2 + X^2 \frac{(1-\tau^2)^2}{f_1{}^2\tau^2} - \frac{q^2}{f_1{}^2\tau^2} = 0 \\ &X = q[f_1{}^2\tau^2 + (1-\tau^2)^2]^{-\frac{1}{2}} \\ &\tan \psi = \frac{f_1 \tau}{1-\tau} \end{aligned}\right\} \quad \ldots\ldots\ldots [11]$$

The amplitude is q when $\tau = 0$; at resonance when $\tau = 1$, the amplitude is q/f_1; as τ approaches infinity, the amplitude approaches zero. The phase angle ψ is zero, $\pi/2$, and π when τ is zero, unity, and infinity, respectively. The condition of free periodicity requires that $f_1 \leqq 2$.

(*b*) "Centrifugal" Disturbing Force:

$$h_1 = c_1 \omega_c k^{-1} = \frac{c_1}{m\omega_c}$$

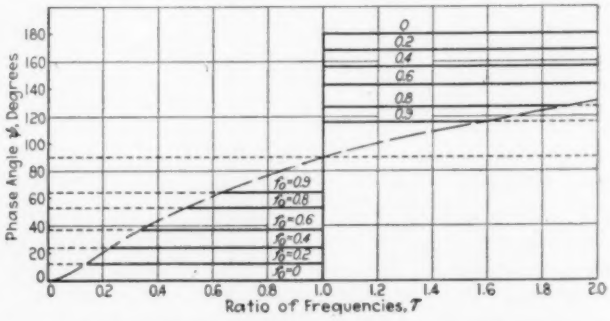


FIG. 4 SHOWING FREQUENCY-PHASE-ANGLE CURVES CORRESPONDING TO THE AMPLITUDE CURVES IN FIG. 3

The general Equation [10] then becomes

$$\left.\begin{aligned} &X^2 + X^2 \frac{(1-\tau^2)^2}{h_2{}^2\tau^2} - \frac{s^2\tau^4}{h_1{}^2\tau^4} = 0 \\ &X = s\tau^2[h_1{}^2\tau^2 + (1-\tau^2)^2]^{-\frac{1}{2}} \\ &\tan \psi = \frac{h_1 \tau}{1-\tau^2} \end{aligned}\right\} \quad \ldots\ldots\ldots [12]$$

showing that the amplitude is zero, s/h_1, and $-s$, when τ is zero, unity, and infinity, respectively. The phase angle shows the same characteristics as before. Also, the condition of free periodicity requires that $h_1 \leqq 2$.

The results stated under case I are so well known that it has been deemed unnecessary to draw curves exhibiting the characteristics.

Case II—A System With Coulomb Damping

If $n = 0$, we have the case of a system with constant friction.

Our solution by means of the general equations is, of course, only approximate. From Fig. 1 we see that $\gamma = 4/\pi$.

(*a*) "Constant" Disturbing Force:

$$f_0 = c_0 \frac{4}{\pi} Q^{-1} = \frac{4}{\pi} \frac{F}{Q}$$

The general Equation [9] then becomes,

$$1 + X^2 \frac{q^{-2}(1 - \tau^2)^2}{f_0^2} - \frac{1}{f_0^2} = 0$$

$$X = q \frac{(1 - f_0^2)^{\frac{1}{2}}}{1 - \tau^2}$$

$$\tan \psi = \frac{f_0 X^{-1}}{q^{-1}(1 - \tau^2)} = \frac{f_0}{(1 - f_0^2)^{\frac{1}{2}}}$$

The arbitrary condition of free periodicity requires that

$$X\tau \leqq \frac{f_0}{2} q$$

giving

$$f_0 \geqq 2\tau[(1 - \tau^2)^2 + 4\tau^2]^{-\frac{1}{2}}$$

to which corresponds a minimum amplitude of

$$X_{\text{min.}} = q[(1 - \tau^2)^2 + 4\tau^2]^{-\frac{1}{2}}$$

The above results are shown in Fig. 3 where a family of curves has been drawn for q equal to unity and for different values of f_0. Infinite amplitude occurs at resonance when $\tau = 1$; as τ approaches infinity, the amplitude approaches zero from the negative side. The arbitrary condition of free periodicity is shown by the dotted part of the curves; thus when $f_0 = 0.8$, the system does not emerge from the aperiodic state until τ is greater than 0.5; moreover, at a value of $\tau = 1.85$ approximately, the system returns to the aperiodic state. This does not mean that no motion takes place except when τ lies between the two values, but it means that our approximate solution cannot be expected to give very trustworthy results except for the range of $\tau = 0.5$ to $\tau = 1.85$. Fig. 4 shows the corresponding phase angles. They are constant until $\tau = 1$, at which point they change suddenly to other constant values, symmetrically distributed about $\pi/2$. The condition of free periodicity cuts off certain parts of the curves as shown by the dotted lines.

(*b*) "Centrifugal" Disturbing Force:

$$h_0 = c_0 \frac{4}{\pi} \omega_c^{-2} P^{-1} = \frac{4}{\pi} \frac{F}{P\omega_e^2}$$

The general Equation [10] then becomes,

$$\left.\begin{aligned} 1 + X^2 \frac{s^{-2}(1 - \tau^2)^2}{h_0^2} - \frac{\tau^4}{h_0^2} &= 0 \\ X = s \frac{(\tau^4 - h_0^2)^{\frac{1}{2}}}{1 - \tau^2} & \\ \tan \psi = \frac{h_0 X^{-1}}{s^{-1}(1 - \tau^2)} = \frac{h_0}{(\tau^4 - h_0^2)^{\frac{1}{2}}} & \end{aligned}\right\} \quad \ldots\ldots [14]$$

The condition of free periodicity requires that

$$X\tau \leqq \frac{h_0}{2} s$$

giving

$$h_0 \geqq 2\tau^3[(1 - \tau^2)^2 + 4\tau^2]^{-\frac{1}{2}}$$

to which corresponds a minimum amplitude of

$$X_{\text{min.}} = s\tau^2[(1 - \tau^2)^2 + 4\tau^2]^{-\frac{1}{2}}$$

Fig. 5 shows a family of amplitude curves, constructed for different values of h_0 and for s equal to unity. The frequency-amplitude plane is divided into four regions by the zero ordinate and by the abscissa for $\tau = 1$; two of these regions are used for plotting the real values of the amplitudes; in the other two regions the amplitude is imaginary and has no physical significance. It is interesting to note that, unless h_0 is zero, the amplitude is imaginary and negative at start when $\tau = 0$. If, for instance, h_0 be 0.4, the curve starts at minus imaginary 0.4, it crosses the zero axis at right angles when $\tau = 0.633$, and enters the region of real values. On account of the arbitrary criterion of free periodicity, the curve is still shown dotted until $\tau = 0.66$ approximately; from then on the amplitude increases rapidly and becomes infinite at the resonance value of τ. For values of τ greater than unity, the amplitude is real and negative, approaching the limit $-s$ as τ approaches infinity. When h_0 is greater than unity, the amplitude curve has no positive real values. Thus, for $h_0 = 2$, the curve starts at minus imagi-

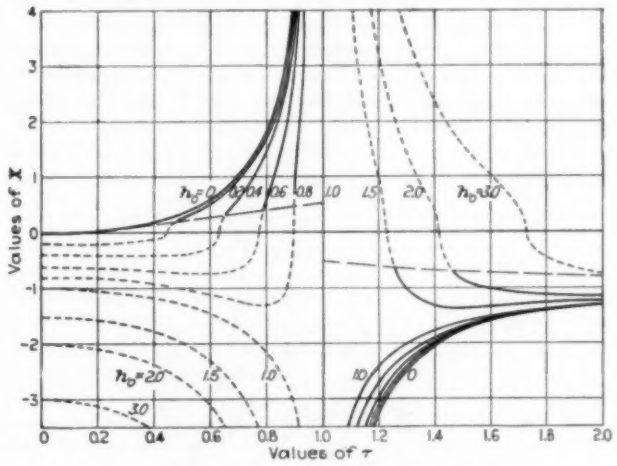


FIG. 5 SHOWING FREQUENCY-AMPLITUDE CURVES OF THE FORCED VIBRATION WITH COULOMB DAMPING AND "CENTRIFUGAL" DISTURBING FORCE FOR DIFFERENT VALUES OF THE DAMPING FACTOR h_0

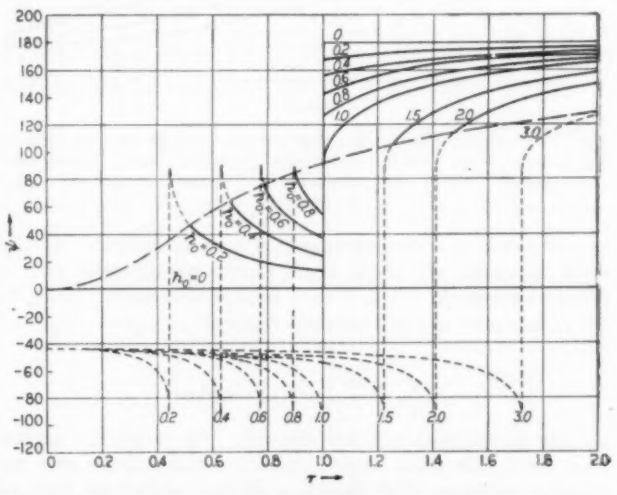


FIG. 6 SHOWING FREQUENCY-PHASE-ANGLE CURVES CORRESPONDING TO THE AMPLITUDE CURVES IN FIG. 5

nary 2, at $\tau = 1$ the amplitude is plus or minus imaginary infinite, and the first zero value is reached when $\tau = \sqrt{2}$; from then on the curve enters the region of negative real values of the amplitude, and overcomes the requirement of the condition for free periodicity when $\tau = 1.46$ approximately. The limit $-s$ is approached from the zero side as τ approaches infinity. The interesting characteristic shown by this family of curves is that no motion occurs until the frequency of the "centrifugal" disturbing force has reached the value $\omega_c\sqrt{h_0}$; further increases in the frequency result, first in forced vibrations in the aperiodic realm, and then in forced vibrations in the free periodic realm for which our approximate solution gives usable results.

The phase angles ψ corresponding to the amplitude curves in Fig. 5 are given in Fig. 6. The ordinate $\psi = 0$ divides the frequency-phase-angle plane into two regions, the upper being used for real values, the lower for imaginary values of ψ. All the curves, with the exception of the one for $h_0 = 0$, start at minus imaginary $\pi/4$; they then decrease to minus imaginary $\pi/2$

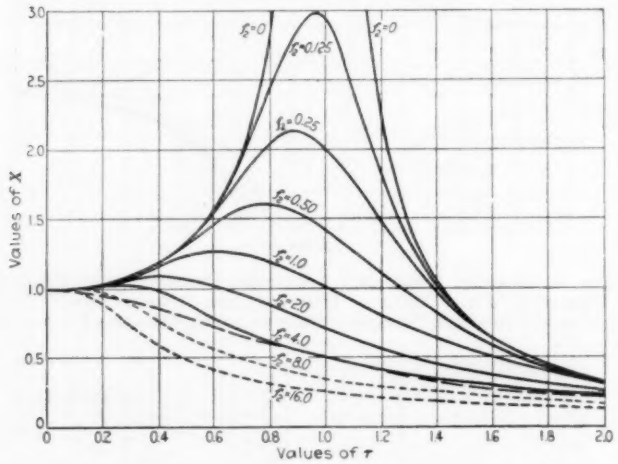


FIG. 7 SHOWING FREQUENCY-AMPLITUDE CURVES OF THE FORCED VIBRATION WITH VELOCITY-SQUARE DAMPING AND "CONSTANT" DISTURBING FORCE, FOR DIFFERENT VALUES OF THE DAMPING FACTOR f_2

at the value of τ corresponding to zero amplitude. At this point, however, the phase angle is also positive real $\pi/2$; a further increase in τ will, if h_0 is smaller than unity, make the curve drop until at $\tau = 1$ the finite angles of

$$\psi = \tan\left[\pm \frac{h_0}{(1 - h_0^2)^{\frac{1}{2}}}\right]$$

equally spaced from $\pi/2$, are reached. For values of τ greater than unity, the phase angle approaches the value π as τ approaches infinity. The condition of free periodicity cuts off certain parts of the curves in the region of real values of the phase angle.

Case III—A System With Velocity-Square Damping

If $n = 2$, we have the case of a system influenced by friction proportional to the second power of the velocity. The application of the general equations to this problem will also give approximate solutions. From Fig. 1 it is seen that $\gamma = 8/3\pi = 0.847$.

(*a*) "Constant" Disturbing Force:

$$f_2 = c_2 \frac{8}{3\pi} \omega_c^2 k^{-2} Q = \frac{\frac{8}{3\pi} c_2 Q}{m^2 \omega_c^2}$$

The general Equation [9] becomes

$$X^4 + X^2 \frac{q^2(1 - \tau^2)^2}{f_2^2\tau^4} - \frac{q^4}{f_2^2\tau^4} = 0$$

$$X = \frac{q}{\sqrt{2}\, f_2\tau^2} \left[\left\{(1 - \tau^2)^4 + 4f_2^2\tau^4\right\}^{\frac{1}{2}} - (1 - \tau^2)^2\right]^{\frac{1}{2}}$$

$$\tan\psi = \frac{f_2\tau^2 X}{q(1 - \tau^2)} = \frac{1}{\sqrt{2}}\left[\left\{1 + \frac{4f_2^2\tau^4}{(1 - \tau^2)^4}\right\}^{\frac{1}{2}} - 1\right]^{\frac{1}{2}}$$

The arbitrary condition of free periodicity requires that

$$X\tau \leqq \frac{2}{f_2} q$$

giving

$$f_2 \leqq \left[16 + \frac{4(1 - \tau^2)^2}{\tau_2}\right]^{\frac{1}{2}}$$

which tells us that for free periodicity the maximum value of f_2 is 4 and occurs at resonance. It therefore appears that only in the case of very high values of c_2 the vibrating system with velocity-square damping is likely to fall into the category of free aperiodic systems.

The limiting value of the amplitude as τ approaches zero is

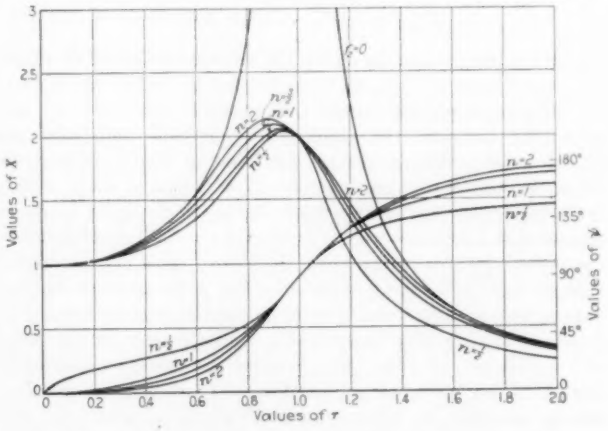


FIG. 8 SHOWING FREQUENCY-AMPLITUDE AND FREQUENCY-PHASE-ANGLE CURVES FOR THE FORCED VIBRATION WITH "CONSTANT" DISTURBING FORCE AND FOR DIFFERENT VALUES OF THE VELOCITY-DAMPING POWER n

easily shown to be q, just as in the system where n is unity. At resonance the amplitude becomes $q/\sqrt{f_2}$. As τ approaches infinity, the limiting value of the amplitude is zero or imaginary q/f_2, which means, physically speaking, that it is zero. The phase angle behaves qualitatively as the phase angle for the case of first-power velocity damping.

Fig. 7 shows a family of curves for different values of f_2. With the exception of the resonance range, the curves follow the zero friction curve more closely than the frequency amplitude curves for a system with first-power velocity damping. The maxima of the curves are shifted a considerable distance toward the lower frequencies. In order to illustrate the peculiar conditions for free periodic motion, three curves for large values of the damping factor f_2 have been drawn. The curve for $f_2 = 4$ becomes tangent to the boundary curve for the free periodic realm at the resonance point. The curve for $f_2 = 8$ enters the aperiodic region when $\tau = 0.28$ approximately; it leaves that region (not shown on curves) when $\tau = 3.5$ approximately. Similarly, the amplitude curve for $f_2 = 16$ plunges into the region

of free aperiodic motion when $\tau = 0.13$, and leaves it when $\tau = 7.8$. It is doubtful, however, whether or not this criterion of free periodicity has much physical significance for our approximate solution of systems whose damping terms are proportional to powers of the velocity higher than the first.

(*b*) "Centrifugal" Disturbing Force:

$$h_2 = c_2 \frac{8}{3\pi} \omega_c{}^4 k^{-2} P = \frac{\frac{8}{3\pi} c_2 P}{m^2}$$

The general Equation [10] becomes,

$$X^4 + X^2 \frac{s^2(1-\tau^2)^2}{h_2{}^2\tau^4} - \frac{s^4\tau^4}{h_2{}^2\tau^4} = 0$$

$$X = \frac{s}{\sqrt{2h_2\tau^2}} \left[\left\{ (1-\tau^2)^4 + 4h_2{}^2\tau^8 \right\}^{\frac{1}{2}} - (1-\tau^2)^2 \right]^{\frac{1}{2}}$$

$$\tan \psi = \frac{h_2\tau X}{s(1-\tau^2)} = \frac{1}{\sqrt{2}} \left[\left\{ 1 + \frac{4h_2{}^2\tau^8}{(1-\tau^2)^4} \right\}^{\frac{1}{2}} - 1 \right]^{\frac{1}{2}}$$

The condition of free periodicity requires that

$$X\tau \leqq \frac{2}{h_2} s$$

giving

$$h_2 \leqq \frac{1}{\tau^2} \left[16 + \frac{4(1-\tau^2)^2}{\tau^2} \right]^{\frac{1}{2}}$$

As τ increases, the critical value of h_2 decreases from infinity to zero. Thus, when $\tau = 1$, $h_2 = 4$; when $\tau = 2$, $h_2 = 1.25$; and when $\tau = \infty$, $h_2 = 0$.

No curves are drawn for this case, as it is clear that they will resemble those of Fig. 7 in all important deviations from the ones of first-power velocity damping. When $\tau = 0$, the amplitude will be zero; at resonance the amplitude will be $s/\sqrt{h_2}$; when τ approaches infinity, this system will, unlike its first-power-velocity relative, have a limiting value of the amplitude of:

$$-s \frac{\left(\sqrt{h_2{}^2 + \frac{1}{4}} - \frac{1}{2} \right)^{\frac{1}{2}}}{h_2}$$

For smaller values of n, i.e., for $0 < n < 2$, the limiting value of the amplitude is $-s$; for $n > 2$, the limiting value of the amplitude will be zero.

Comparison of Systems With *n*th-Power Velocity Damping

In order to make clear the influence of changing the exponent of the velocity-damping term on the amplitude curve, a set of curves has been drawn in Fig. 8. The curves have been drawn so as to give the same resonance amplitude of $2q$. Since the exponents are $^1/_2$, 1, $^2/_3$, and 2, the requirement of equal resonance amplitude necessitates that the friction factors f be 0.707, 0.500, 0.353, and 0.250, respectively. It is seen that there is a considerable change in the shape of the amplitude curves as the exponent of the friction term is varied. The phase-angle curves show a flattening-out tendency accompanying a decrease in the value of n; the limiting form of the phase-angle curve for $n = 0$ is, as shown in Fig. 4, a straight line parallel to the abscissa.

Case IV—Systems With Composite Damping

Let us now consider the case where the damping is proportional to a sum of terms, each one involving the velocity to some power. It is interesting that also in this very general case our approximate method will give results.

The differential equation of motion, before we have applied our approximations, may look like this:

$$m \frac{d^2x}{dt^2} \pm c_{n1} \left(\frac{dx}{dt} \right)^{n1} \pm c_{n2} \left(\frac{dx}{dt} \right)^{n2} + \ldots . kx = Q \sin \omega t . \quad [17]$$

By the criterion of equivalent dissipative work done during a cycle, we obtain:

$$\frac{1}{4} W = c_{n1} \int_0^{\frac{\pi}{2\omega}} \left(\frac{dx}{dt} \right)^{n1} dx + c_{n2} \int_0^{\frac{2\pi}{\omega}} \left(\frac{dx}{dt} \right)^{n2} dx + \ldots .$$

$$= c_1 \int_0^{\frac{\pi}{2\omega}} \frac{dx}{dt} \, dx$$

From which it follows that

$$c_1 = c_{n1} \gamma_{n1} x^{n_1 - 1} \omega^{n_1 - 1} + c_{n2} \gamma_{n2} X^{n_2 - 1} \omega^{n_2 - 1} + \ldots \quad [18]$$

If this value of c_1 be inserted in the solution for the simple system, involving first-power velocity damping only, an expression results which, in the general case, can be solved by successive approximations.

Due to the frequent occurrence of constant friction and first-power velocity damping in engineering structures, let us see what the approximate solution gives in this case.

Case V—A System With Coulomb and Viscosity Damping

Since $n_1 = 0$ and $n_2 = 1$, we find, by means of Fig. 1 that

$$\gamma_{n1} = \frac{4}{\pi} \qquad \gamma_{n2} = 1$$

Inserting this into Equation [18]

$$c_1 = c_0 \frac{4}{\pi} X^{-1} \omega^{-1} + c = \frac{4}{\pi} F X^{-1} \omega^{-1} + c$$

Substitution of this value of c_1 into the general solution, Equation [2], will yield

$$x = \frac{Q}{\sqrt{(k - m\omega^2)^2 + \left(\frac{4}{\pi} F X^{-1} \omega^{-1} + c \right)^2 \omega^2}} \sin(\omega t - \psi)$$

$$= X \sin(\omega t - \psi)$$

$$\tan \psi = \frac{\left(\frac{4}{\pi} F X^{-1} \omega^{-1} + c \right) \omega}{k - m\omega^2}$$

Squaring both sides and rearranging terms, the following quadratic equation in X is obtained:

$$X^2 + \frac{\frac{8}{\pi} cF}{c^2\omega^2 + (k - m\omega^2)^2} X + \frac{\frac{16}{\pi^2} F^2 - Q^2}{c^2\omega^2 + (k - m\omega^2)^2} = 0$$

from which

$$X = \frac{\left[Q^2 \left\{ c^2\omega^2 + (k - m\omega^2)^2 \right\} - \frac{16}{\pi^2} F^2 (k - m\omega^2)^2 \right]^{\frac{1}{2}} - \frac{4}{\pi} cF\omega}{c^2\omega^2 + (k - m\omega^2)^2}$$

. . . . . [19]

For the sake of simplicity and generality, let us use the ratio symbols that were employed before, Equation [8]; then

$$f_0 = \frac{4}{\pi}\frac{F}{Q} \qquad f_1 = \frac{c}{m\omega_c}$$

and Equation [19] becomes

$$X = \frac{q[\{f_1^2\tau^2 + (1 - f_0^2)(1 - \tau^2)^2\}^{\frac{1}{2}} - f_0 f_1 \tau]}{f_1^2\tau^2 + (1 - \tau^2)} \quad \ldots [19a]$$

$$\tan\psi = \frac{f_1\tau + f_0 X^{-1}}{1 - \tau^2}$$

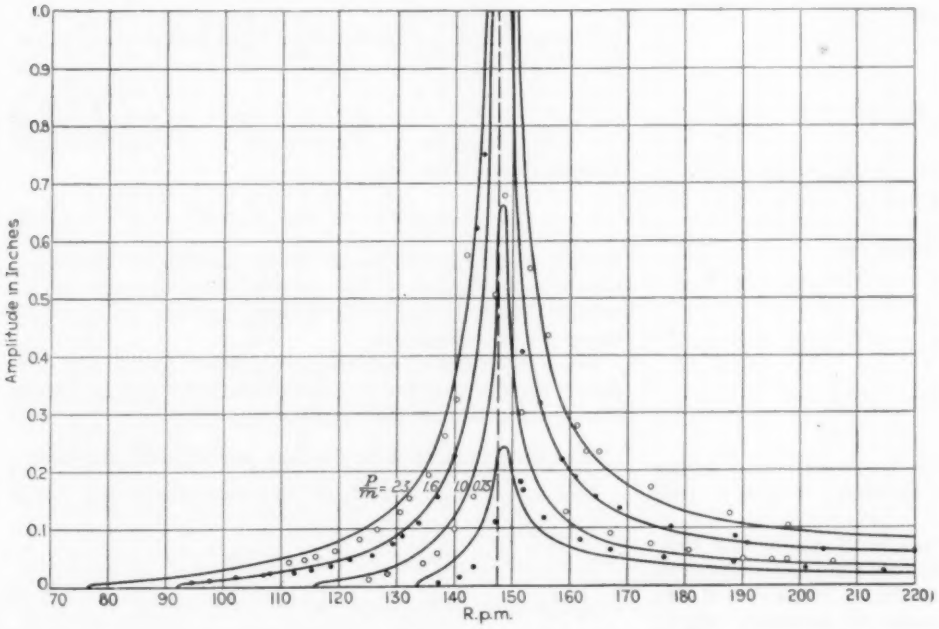


FIG. 9 SHOWING FREQUENCY-AMPLITUDE CURVES OF THE FORCED VIBRATION WITH A COMBINATION OF COULOMB AND VISCOUS DAMPING AND "CENTRIFUGAL" DISTURBING FORCE FOR DIFFERENT VALUES OF THE RATIO P/m (POINTS DETERMINED EXPERIMENTALLY)

It is seen that the motion is similar to the one depicted in Figs. 3 and 4, with the exception that the resonance amplitude is limited to $q(1 - f_0)/f_1$, and that the phase angle does not change abruptly at the resonance point. If, instead of a "constant" disturbing force, a "centrifugal" disturbing force be assumed, the results become

$$\left.\begin{aligned} X &= \frac{\left[P^2\omega^4\{c^2\omega^2 + (k - m\omega^2)^2\} - \frac{16}{\pi^2}F^2(k - m\omega^2)^2\right]^{\frac{1}{2}} - \frac{4}{\pi}cF\omega}{c^2\omega^2 + (k - m\omega^2)^2} \\ \tan\psi &= \frac{c\omega + \frac{4}{\pi}FX^{-1}}{k - m\omega^2} \end{aligned}\right\} \quad [20]$$

At values of ω for which the expression $P^2\omega^2$ is smaller than $(4/\pi)F$, the above equation has no physical meaning; in other words, motion does not begin until

$$\omega = \sqrt{\frac{\frac{4}{\pi}F}{P}}$$

This is in agreement with the results shown in Fig. 5, namely, that the forced vibration with constant friction and "centrifugal" disturbing force requires a certain minimum angular velocity of the disturbing force in order to show any motion at all. At resonance, the value of the amplitude is:

$$X = \pm\left(\frac{P\omega_c}{c} - \frac{\frac{4}{\pi}F}{c\omega_c}\right)$$

It is interesting to note that the amplitude depends not only on P and c, but also on F, a characteristic which is easily recognized in experiments. Above resonance, when ω approaches infinity, the amplitude approaches the asymptotic value of $-P/m$.

In regard to the phase angle, it is seen that, at the value of ω for which motion begins, the amplitude is zero and consequently $\psi = \pi/2$. As ω increases toward the resonance value, the magnitude of the displacement increases with the effect that the expression for $\tan\psi$ first decreases and then increases, giving a minimum somewhere between ω_0 and ω_c. At resonance, ψ is seen to be $\pi/2$, and as ω approaches infinity, the phase angle approaches the value π.

Expressing the motion by ratio symbols, Equation [8], we obtain:

$$h_0 = \frac{4}{\pi}\frac{F}{P\omega_c^2}; \quad h_1 = \frac{c}{m\omega_c}$$

giving

$$X = \frac{s[\{h_1^2\tau^2 + (\tau^4 - h_0^2)(1 - \tau^2)^2\}^{\frac{1}{2}} - h_0 h_1\tau]}{h_1^2\tau^2 + (1 - \tau^2)^2} \quad \ldots [20a]$$

$$\tan\psi = \frac{h_1\tau + h_0 X^{-1}}{1 - \tau^2}$$

The free-periodicity condition gives

$$X\tau \geqq \frac{sh_0}{2 - h_1} = \text{a constant}$$

a requirement which can easily be superimposed on the curves after they have been constructed.

Fig. 9 shows a series of four amplitude curves for the following values of s or P/m: 2.3, 1.6, 1.0, and 0.75. The points plotted on the same figure are the experimental results. Neglecting for the present these points, we notice that the theoretical curves start with finite slopes at definite angular frequencies or r.p.m. The arbitrary condition of free periodicity cuts off a small part of the curves at start. It is noticeable that the resonance amplitudes of the curves are not simply proportional to P/m, being 2.84 in., 1.665 in., 0.665 in., and 0.249 in., respectively. Attention should be called to the fact that the r.p.m. scale does not start at zero but at 70 cycles per minute, or approximately at $\tau = 0.5$. Fig. 10 shows the phase-angle curves corresponding to the amplitude curves of Fig. 9. Here again the criterion of free periodicity discredits our faith in the physical approximation of our method for the

part of the curves shown dotted. Minimum phase angles are obtained at approximately 133, 136, 138, and 142 cycles per minute; they amount to approximately 21.5, 31, 46, and 67 deg. Above the resonance frequency, all the curves approach π as a limit.

II—EXPERIMENTAL WORK

Description of Apparatus Employed

In order to test the foregoing conclusions concerning the steady forced vibration of a system under the influence of constant friction and first-power velocity damping, a series of experiments was performed on a large vibrating table[5] belonging to the School of Engineering at Stanford University. Fig. 11 shows a sketch of the table. The following dimensions and properties of the table have been measured or disclosed by free-vibration tests:

Weight of vibrating mass	19,800 lb.
Spring characteristic	12,320 lb. per in.
Constant friction force	114 lb.
Coefficient of first-power velocity damping	9.3 lb. per in. per sec.
Resonance frequency	148 cycles per min.

The disturbing force was produced by the centrifugal reaction of a rotating unbalanced flywheel, mounted rigidly on the table. Four different unbalanced moments of the flywheel were used: 890 lb-in., 620 lb-in., 390 lb-in., and 290 lb-in. The unbalances as well as the friction constants were determined with an accuracy of ± 3 per cent. A continuous record of the amplitude and the phase angle was obtained by means of a simple recording mechanism.

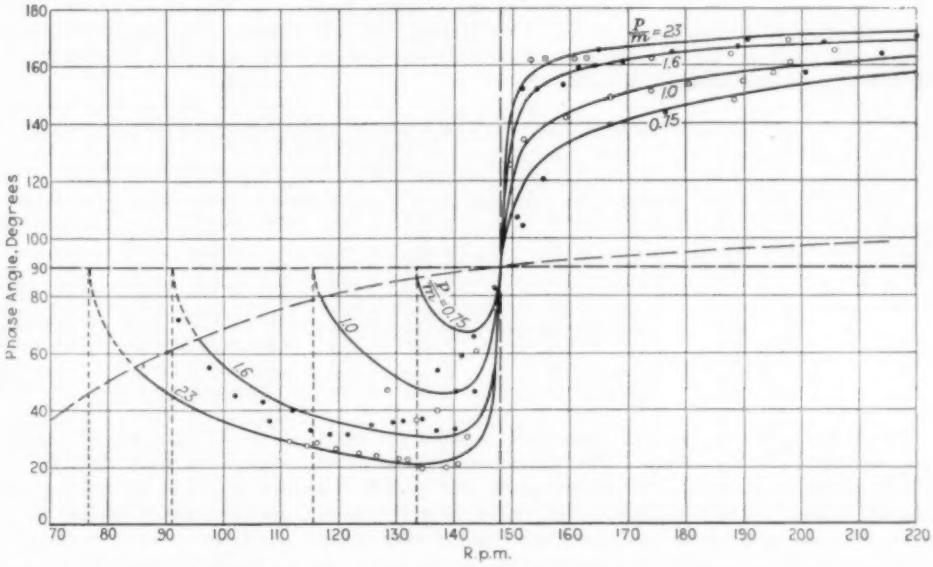


Fig. 10 Showing Frequency-Phase-Angle Curves Corresponding to the Amplitude Curves in Fig. 9

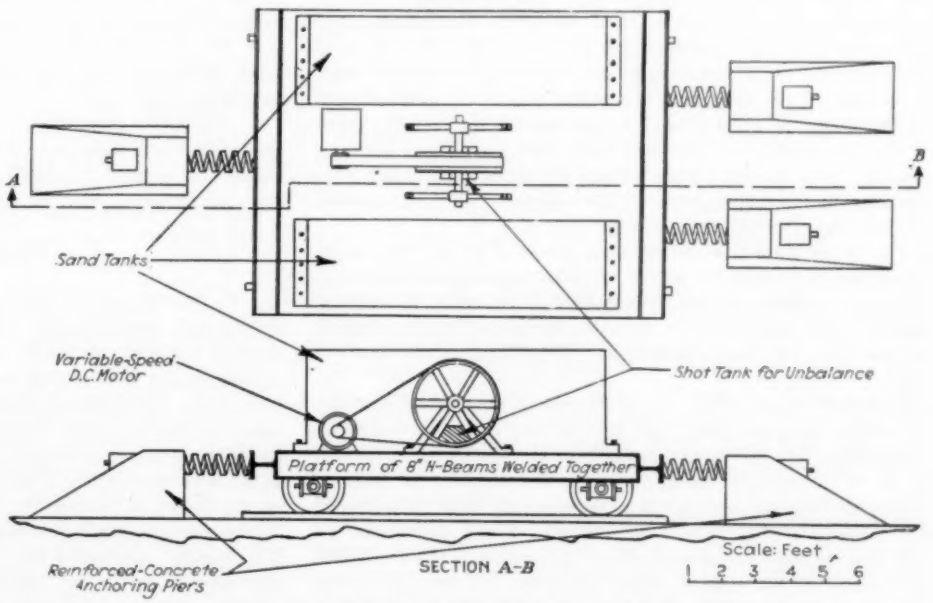


Fig. 11 Diagrammatic Sketch of Vibrating Table at Stanford University

Discussion of Experimental Results

Referring to Fig. 9, it will be seen that the experimentally determined points follow the theoretical, approximate curves very closely. As the unbalance is decreased, the experimental points seem to shift slightly to the right of the curves. This is quite easily explained by a small variation of the spring constant, a thing very likely to happen as the four curves were not obtained on the same day. Near resonance it was found very difficult to keep the frequency of the impressed force constant, especially so when the unbalance was large, hence very few points were obtained at 148 cycles per minute. Considering that the constant friction force is rather large compared to the first-power velocity damping coefficient, one is impressed with the agreement of experimental data and approximate theory.

The phase angles shown in Fig. 10 are also, considering their capricious nature, in good agreement with the approximate theory. All the records, with the exception of those at the starting frequencies, show wave shapes that approximate the sinusoid very closely. The phase angles were determined with reference to the zero-amplitude time position.

[5] For a detailed description of the vibrating table, see author's paper on "Vibration Research at Stanford University," *Bulletin of Seismological Society of America*, March, 1929.

Comparison of the Den Hartog Exact Theory and the Approximate Theory

It was stated earlier in this paper that, fortunately for the approximate theory, an exact theory for the case of constant friction and first-power velocity damping had been developed. The four special cases, for which experimental evidence had been obtained, were computed by Mr. Den Hartog, and his results are given in Table 1 for comparison with the results obtained by the approximate theory.

TABLE 1 COMPARISON OF THE RESULTS OBTAINED BY THE EXACT THEORY AND BY THE APPROXIMATE METHOD

	$P/m = 2.3$			$P/m = 1.6$			$P/m = 1.0$			$P/m = 0.75$		
R.p.m.	Exact	Approx.	Diff., per cent	Exact	Approx.	Diff., per cent	Exact	Approx.	Diff., per cent	Exact	Approx.	Diff., per cent
113.5	0.061	0.056	−8	0.039	0.034	−13	0.017	.....	....	.....	.....	....
123	0.094	0.093	−1	0.061	0.058	−5	0.025	0.021	−16	.....	.....	....
128.5	0.129	0.131	+2	0.083	0.083	0	0.038	0.035	−8	.....	.....	....
134.5	0.197	0.200	+2	0.126	0.130	+3	0.056	0.061	+9	.....	.....	....
141	0.403	0.420	+4	0.260	0.273	+5	0.130	0.133	+2	0.055	0.055	0
145	0.920	0.962	+4	0.581	0.615	+6	0.277	0.292	+5	0.111	0.127	+14
148	2.840	2.840	0	1.660	1.665	0	0.660	0.665	+1	0.242	0.249	+3
151	0.964	0.903	−6	0.618	0.588	−5	0.303	0.298	−2	0.146	0.153	+5
156	0.433	0.420	−3	0.288	0.279	−3	0.154	0.151	−2	0.088	0.089	+1
164.5	0.227	0.226	0	0.152	0.153	0	0.086	0.086	0	0.053	0.055	+2
174	0.157	0.157	0	0.106	0.107	+1	0.061	0.062	+1	0.040	0.041	+2
185	0.122	0.121	−1	0.083	0.083	0	0.049	0.049	0	0.033	0.034	+3

The table shows an excellent agreement between the exact and the approximate theory. The difference is greatest at start; this is to be expected, inasmuch as the assumption of a sinusoid wave form of the displacement cannot be a very good approximation until the motion has reached a fair magnitude; in some cases the difference amounts to about 15 per cent. At resonance the two methods give practically the same results, and, above the resonance point, the error of the approximate method may be said to be within 5 per cent. Considering that the agreement is as satisfactory in this case where the values of n are zero and unity, the term with $n = 0$ being the predominant, it is easy to see that in other cases, where the predominant friction term is proportional to powers of the velocity near unity, the approximate method must give even closer approximations than recorded here. Even the case of damping proportional to the second power of the velocity should be expected to give very close approximations.

Let us consider the assumptions involved in the theoretical treatment of system for which we have experimental evidence; they are constant friction independent of the velocity, constant spring characteristic, and a vibrating system of just one degree of freedom. Of these assumptions let us examine the first one only.

Experiments on coefficients of friction of dry and of partially lubricated surfaces have been carried out by many early investigators with the general result that the coefficient of friction is never completely independent of the velocity of motion. Professor Fleming Jenkin,[6] in experimenting with velocities in the range of 0.0024 to 0.12 in. per sec., found that there did not seem to exist a discontinuity in the value of the coefficient of friction at rest and at small velocities. If this be so, the free-vibration tests will give average values of the coefficient of friction, but the averages will, of course, be higher the smaller the amplitudes of the free vibration. It is therefore very likely that the assumption of a constant friction force is not in complete harmony with experiment, and one invariably asks the question if, from a pragmatic point of view, an approximate theory might not give as accurate results as an exact theory? From an academic point of view, however, the exact theory, even if based on assumptions that are not strictly physical realities, is vastly superior to an approximate theory or solution; in fact, it is the only one that has any claim to recognition.

[6] Proceedings of the Royal Society, March, 1877.

Conclusions

1 A general method of obtaining approximate solutions of the steady forced vibration of a damped system of one degree of freedom has been developed for the case of sinusoidally varying disturbing forces.

2 The approximation consists in expressing all the damping terms of the original differential equation by a single equivalent damping term, proportional to the first power of the velocity of motion.

3 The criterion of equivalence is taken to be the amount of dissipative work done per cycle. If in addition to this it be assumed that the friction does not change the wave shape of the displacement from that of a sinusoid, the equivalent damping coefficient is easily evaluated.

4 In the case of a system influenced by a centrifugal disturbing force and damped by constant friction and by friction proportional to the first power of the velocity, experimental evidence is in good agreement with the approximate solution.

5 Mr. Den Hartog's exact solution for the above system is in good agreement with the experimental results and with the approximate solution.

6 While this paper has dealt solely with a system of one degree of freedom, there is no reason why the approximate solution cannot be applied to systems of several degrees of freedom; such applications will, of course, become relatively involved.

Discussion

J. P. Den Hartog.[7] The paper solves in one stroke and with very simple mathematics a number of problems of which the solution has been wanted for many years.

Since in the general case no exact solution is available to compare with, the question naturally arises whether we can form some qualitative ideas regarding the accuracy of the procedure. In this respect it is noted first that in Fig. 2 the sine curve ($n = 1$) is the first term in the Fourier development of any of the other curves shown in that figure. There is a definite reason for this. The author has made the assumption of pure harmonic motion. The energy dissipated by a non-harmonic periodic force in combination with a pure harmonic motion of the same frequency can be shown to be due to the fundamental component of the force only, while its higher harmonics do not perform any work during a complete cycle.

The author has replaced the curves of Fig. 2 by the particular curve $n = 1$, thus neglecting the higher harmonics of the damping force. As said before, this does not affect the energy relations. The main influence of these higher harmonics will then consist of a distortion of the wave form of the motion.

Let us try to extend the process somewhat by dropping the assumption of a harmonic motion, but by retaining a damping

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force, determined by the curves of Fig. 2, which will be correct only for harmonic motion.

Then let us investigate how this damping force distorts the motion. On account of their symmetry the curves of Fig. 2 can be expressed by sine series with odd components only. The differential equation of the motion thus becomes:

$$m\ddot{x} + kx = Q \cos(\omega t + \varphi) + \sum_{n=1,3,5\ldots} a_n \sin n\omega t \ldots\ldots [21]$$

where φ is as yet undetermined and the a_n's are the Fourier coefficients of the damping curve in Fig. 2. The steady-state solution of this equation is:

$$x = \frac{Q}{k - m\omega^2} \cos(\omega t + \varphi) + \sum_{n=1,3,5\ldots} \frac{a_n \sin n\omega t}{k - n^2 m\omega^2} \ldots [22]$$

It is seen that the first two terms have the period ωt. In the neighborhood of resonance these terms are large on account of the $k - m\omega^2$ in the denominator. The remaining terms with periods $3\omega t$, $5\omega t$, etc., are smaller and represent the distortion in the motion. Therefore, comparing the accuracies of the results of the paper for the various cases of Fig. 2, it can be roughly said that the Fourier coefficients a_3, a_5, etc., of the various curves give an estimate of the error committed. The Fourier expansions for the two principal cases are:

$$\text{For } n = 0: \quad \frac{4}{\pi}\left[\sin \omega t + \frac{1}{3}\sin 3\omega t + \frac{1}{5}\sin 5\omega t \ldots\right] \ldots [23a]$$

$$\text{For } n = 2: \quad \frac{8}{3\pi}\left[\sin \omega t - \frac{1}{5}\sin 3\omega t - \frac{1}{35}\sin 5\omega t \ldots\right]. \quad [23b]$$

Therefore, the conclusion seems justified that the approximation in the neighborhood of resonance is even better for the square velocity damping case than for the Coulomb damping case.

An examination of [22] leads to yet another fact. It is seen that near one-third, one-fifth, etc., of the resonant frequency, the corresponding term in the series becomes large, so that the motion is very much distorted. Near such frequencies, therefore, the approximation given should be poor. Indeed the exact solution for the Coulomb case shows that stops in the motion occur at those frequencies even for infinitesimal small damping. Happily, however, these cases are of little practical importance.

A third interesting property of Equation [22] is that it enables us to arrive at an exact solution in series form for the case of Coulomb friction as long as no stops occur during the cycle. In this case, the magnitude of the damping force is independent of the velocity, that is of the wave form, so that [21] is the true differential equation.

In the Fourier expansion of the damping curve the origin of time was taken at the left-hand side of Fig. 2, at the moment that the damping force passes through zero. This means that at this moment the motion must be at its maximum or minimum position, and the velocity must be zero. Differentiating [22], after substituting the series [23a] for the a_n in it, we obtain:

$$\dot{x} = \frac{-Q\omega}{k - m\omega^2} \sin(\omega t + \varphi) + \frac{4F}{\pi} \sum_{n=1,3,5\ldots} \frac{\omega \cos n\omega t}{k - n^2 m\omega^2}$$

By equating this to zero for $t = 0$, a condition for φ is found:

$$(\dot{x})_{t=0} = 0 \quad \frac{-Q\omega}{k - m\omega^2} \sin\varphi + \frac{4F}{\pi} \sum_{1,3,5\ldots} \frac{\omega}{k - n^2 m\omega^2}$$

or

$$\sin\varphi = \frac{4}{\pi}\frac{F}{Q}\left(\frac{k}{m} - \omega^2\right) \sum_{n=1,3,5\ldots} \frac{1}{k/m - n^2\omega^2} \ldots\ldots [24]$$

The maximum amplitude X also occurs at $t = 0$; then the entire series in [22] vanishes:

$$X = \frac{Q}{k - m\omega^2} \cos\varphi \ldots\ldots\ldots\ldots\ldots [25]$$

It is seen that [24] and [25] constitute an exact solution of the amplitude and phase angle, while [22] gives the wave form.

It is appreciated that this is true only when the series in [21] actually represents the damping force. When the friction becomes so large that stops occur during the motion this ceases to be correct, so that the solution is limited to motions without stops.

It is evident that when [24] and [25] are restricted to only the first term of the series, the result:

$$X = \frac{Q}{k - m\omega^2} \sqrt{1 - \left(\frac{4}{\pi}\frac{F}{Q}\right)^2}$$

coincides with the solution given in the paper.

The solution [24], [25] naturally gives the same numerical results as the exact solution obtained before,[8] though the latter appears in the finite form:

$$X = \frac{Q}{k} \sqrt{\left(\frac{\beta^2}{\beta^2 - 1}\right)^2 - \left(\frac{F}{Q}\right)^2 \left(\frac{\beta \sin \beta\pi}{1 + \cos \beta\pi}\right)^2} \ldots [26]$$

where

$$\beta = \frac{1}{\omega}\sqrt{\frac{k}{m}}$$

A comparison of [25] with [26] incidentally leads to the interesting series:

$$\sum_{n=1,3,5\ldots} \frac{1}{n^2 - \beta^2} = \frac{\pi}{4} \frac{\sin \beta\pi}{\beta(1 + \cos \beta\pi)} \ldots\ldots\ldots [27]$$

which is true for all values of β and which would probably be difficult to prove mathematically, without having recourse to the physical problem from which it was derived.

Summarizing, the following statements can be made:

1. For any n not larger than 2, the results obtained by the approximate method are more accurate than those for the case $n = 0$, at least in a sufficiently broad region near resonance. Since for $n = 0$ the error is very small, the method is reliable for all practical cases.

2. Larger errors are to be expected at frequencies near one-third, one-fifth, etc., of the natural frequency. Since these cases are of no practical importance, it does not affect the usefulness of the solution.

3. For the special case of Coulomb damping ($n = 0$) the method employed in the paper can be extended to give an exact solution in series form for friction forces sufficiently small to permit motions without periodic stops.

E. O. Waters.[9] The author's analysis is a most interesting extension of the theory of vibrations, and the formulations are simple enough to be within the reach of most engineers who have to do with vibration problems. It is indeed un-

[8] *Phil. Mag.*, 1930, p. 803.

[9] Associate Professor of Mechanical Engineering, Yale University, New Haven, Conn. Assoc-Mem. A.S.M.E.

fortunate that only one particular type of vibration—that with viscous damping—should lend itself to an easy and at the same time rigorous mathematical treatment, because the tendency has been to concentrate attention upon this type to the exclusion of others which may be of quite as common physical occurrence. In most of the elementary analyses of the forces and stresses in machines, the friction force is assumed constant; but as soon as periodic motion is encountered, the engineer has not had the means at his disposal to determine the effect of this kind of friction even approximately.

From an abstractly mathematical point of view, there is nothing difficult about the solution of the equations for steady forced vibrations with Coulomb damping, or a combination of viscous and Coulomb damping; the trouble comes in trying to get numerical results. On account of the + sign appearing before the damping term, there is a discontinuity each time the velocity passes through zero, and the constants of integration have to be evaluated anew at each such point. But these constants are contained in the transient portion of the complete expression for the motion, whereas it is only the steady state in which we are interested. Mr. Den Hartog and Professor Milne have shown that the problem can be solved both for the case of constant damping and that of velocity-square damping, but a perusal of their papers leads one to the conclusion that the method outlined in the present paper, with its simplifying assumptions, is better adapted to practical numerical computation. This is especially true for composite Coulomb and viscous damping, for which Table 1 shows a very good agreement between the approximate and the exact theory. It would be interesting to know whether an equally good correlation holds for the case of velocity-square damping.

As the author states, his method does not reveal anything regarding possible complications in the wave shape, and simply assumes that it is sinusoidal. It may be of interest to know that the actual wave shape may be found quite easily, with any desired degree of accuracy, for a given combination of Coulomb and viscous damping. The method is based upon the use of a Fourier series for the damping, and the solution of the equation of motion by the well-known method of undetermined coefficients, together with the condition that $c_0 = 0$ when $x = 0$.

If the friction is constant in magnitude, but varies principally in sign, it may be expressed as

$$\frac{4s_0}{\pi}\left[\sin\omega t + \frac{1}{3}\sin 3\omega t + \frac{1}{5}\sin 5\omega t + \ldots\ldots\right]$$

Four terms give a mean value better than 97 per cent of c_0 and a correspondingly low root-mean-square deviation. The equation of motion is

$$mx - \frac{4c_0}{\pi}\left[\sin\omega t + \frac{1}{3}\sin 3\omega t + \frac{1}{5}\sin 5\omega t + \ldots\right] + bx = Q\sin(\omega t - \psi)$$

the solution of which is (neglecting transients)

$$x = -\frac{4c_0}{\pi}\left(\frac{1}{k - qm\omega^2} + \frac{1}{k - 25m\omega^2} + \ldots\right)\sin\omega t + \sqrt{\left(\frac{Q}{k - m\omega^2}\right)^2 - \frac{16c_0^2}{\pi^2}\left(\frac{1}{k - m\omega^2} + \frac{1}{k - qm\omega^2} + \frac{1}{k - 25m\omega^2}\right)^2}\cos\omega t + \frac{4c_0}{\pi}\left(\frac{\sin 3\omega t}{3(k - qm\omega^2)} + \frac{\sin 5\omega t}{5(k - 25m\omega^2)} + \ldots\right)$$

This becomes identical with the author's solution for a "constant" disturbing force when the higher harmonic terms and corrective coefficients are dropped. At first sight it would seem that there are an infinite number of critical speeds, but an analysis of the cosine term shows that there is only the one critical speed $\omega_c = \sqrt{k/m}$, and a number of aperiodic regions which are reached whenever any of the terms $1/(k - qm\omega^2)$, $1/(k - 25m\omega^2)$, etc., become so large that the coefficient of cos ωt is imaginary.

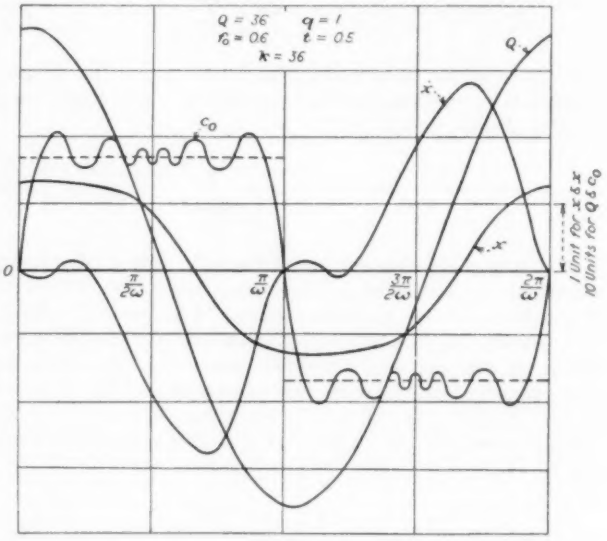


FIG. 12

Fig. 12 illustrates the results for a sample problem, with constants taken close to the aperiodic region as shown in Fig. 3 of the paper. The small oscillations of x about the X-axis are of course incompatible with the damping force as indicated; possibly the inclusion of more harmonics in the analysis, or a choice of constants further from the aperiodic boundary, would iron them out. At any rate, it is evident that the motion is practically composed of sinusoidal swings from one extreme position to the other, interrupted by brief pauses for about one-eighth of a cycle at each end of the travel of the vibrating body.

The foregoing method may be extended to problems in combined Coulomb and viscous damping; it is also applicable to nth-power velocity damping, provided the assumption be made that the damping force is proportional to the nth power of the velocity wave.

A. L. KIMBALL.[10] The author has presented a method of finding the amplitude of a forced vibration at any frequency for any law of damping which he wishes to assume. The method is not exact, but it is near enough so as to be of great value in practical work. He makes the approximation of assuming the vibratory motion to be exactly sinusoidal, regardless of the character of the damping friction. Since the departure from sinusoidal motion is usually very small, the error in this method is correspondingly unimportant.

The paper is of considerable interest to the writer because vibration damping due to internal friction in elastic bodies is a specialty of his. It is interesting that he has used the same method as that used by the author for the particular case of the damping of forced vibrations with "solid friction."[11]

The paper, in the writer's opinion, is outstanding, and it is to be hoped that it will receive a full share of appreciation.

[10] Res. Engr., Gen. Elec. Co., Schenectady, N. Y. Assoc. A.S.M.E.

[11] A. L. Kimball, "Vibration Damping, Including the Case of Solid Friction," Trans. A.S.M.E., paper APM-51-21, pp. 227–236.

Author's Closure

After his paper had been written and submitted to the Committee on Publications of the Applied Mechanics Division of the Society, the author was much interested to learn from Mr. Kimball that he had used a particular case of the general problem dealt with in the paper for expressing the damping due to "solid friction." It is very gratifying for the general problem that the approximation in the case of "solid friction" is found to be satisfactory.

The discussions by Dr. Den Hartog and by Professor Waters are valuable contributions to the paper. If no stops occur in the motion, the exact solution for $n = 0$ is obtainable in series form or in the finite form given by Dr. Den Hartog in his original paper. Of especial interest to the author is Dr. Den Hartog's investigation of the question of the degree of approximation involved when the exponent n is varied.

The author wishes to acknowledge his indebtedness to Prof. L. H. Donnell of the University of Michigan for presenting the paper, and to express his thanks to those who have taken part in the discussion.

APM-52-16

Automotive Brakes With Servo-Action

By DR. F. LOEWENBERG,[1] BERLIN-CHARLOTTENBURG, GERMANY

The retardation produced in braking automotive vehicles results from the pressing of a fixed body against a moving surface. This paper discusses a method in which the friction forces themselves produce higher pressure. This has been called "self-actuation" or "self-energization." Various conditions and arrangements are discussed, and the forces developed are worked out mathematically.

THE design of brakes for automotive vehicles is based on the decelerative effect of friction forces produced by pressing a fixed body against a moving surface. In the case of buses and trucks the physical power of a human being has been found to be insufficient to exert the pressure required for proper brake action. Of course, the power of the driver can be largely increased by using levers, but the leverage is limited by the allowable travel of the pedal. From the total motion of the pedal a sufficient amount has to be deducted for taking up the slack in the brake rigging and the clearance between shoe and drum and to allow for wear of the lining; otherwise, serious difficulties arise in service such as dragging brakes and the necessity of frequent brake adjustments. Only a fraction of the travel can therefore be used for real brake action, which results in a rather definite maximum leverage. Various methods are in use in order to increase or replace the force of the driver, using either the suction of the engine or other means as a source of power. This paper is concerned with another method consisting in the use of the friction forces themselves to produce higher pressure. This has been termed "self-actuation" or "self-energization" and has been used to a certain extent not only in straight mechanical but also in power brakes. It is only in modern brake design, however, that the possibilities of this method have been fully realized.

In order to make clear the meaning of self-actuation and the way it has been used in different brake designs, the basic principle is shown in Fig. 1 in a simplified manner. Here a block is pressed against a straight moving surface, the reaction of the friction forces being taken up by a fixed block with a straight or inclined surface. While this deviates far from the actual design, it gives an idea of the action without the use of formulas, and most of the actual designs can be classified under one of the cases shown. In case (*a*) the reaction of the friction forces has apparently no effect on the pressure between block and moving face (no self-actuation). In case (*b*) the pressure is increased by the reaction of the friction (positive self-actuation), while in case (*c*) it is decreased (negative self-actuation). Under (*d*) an inclined face of very small angle is drawn in order to show that the principle of self-actuation can be brought to the point where the block "wedges itself in," which means that the smallest force exerted upon the block results in a theoretically unlimited pressure between block and moving face. In this case the brake is "grabbing."

The three following cases represent designs with two brakeshoes. In (*e*) two brakeshoes are shown, each having positive self-actuation; in (*f*) and (*g*) one brakeshoe has positive and one negative self-actuation, while (*f*) represents the case in which the force exerted by the brake rigging is equally divided between the two shoes. It can be easily proved that in this case the advantage gained on one shoe exceeds the disadvantage caused by the negative self-actuation of the other shoe, so that the two shoes taken as a whole have positive self-actuation. In case (*g*) the two blocks are connected with the rigging in such a way that the travel of both blocks is equal. This is obtained in actual design, for instance, by a symmetrical fixed cam, and results in the pressure on the sliding face being equal on both blocks. A simple calculation shows that in this case the positive and the negative self-actuation of the two shoes just balance each other, so that the net self-actuation is zero.

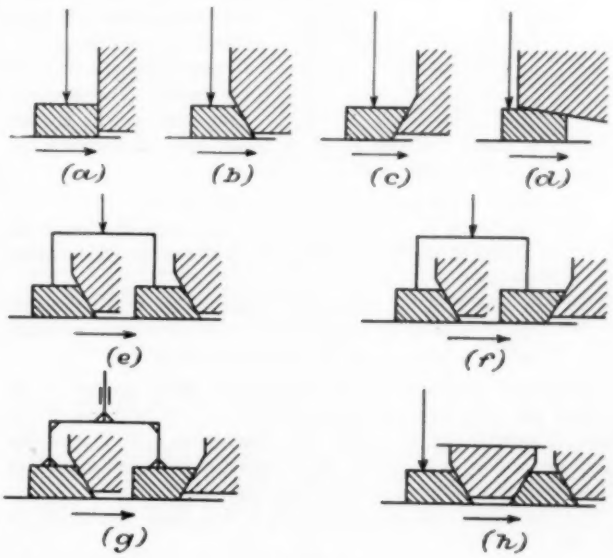


FIG. 1

The principle of another brake design is shown at (*h*), Fig. 1. In this case the pedal pressure acts upon one of the two brakeshoes only. The reaction of the friction forces of this shoe is not taken up by a fixed face, but is transferred to the second shoe, which is actuated by it. Each of the two brakeshoes can be self-actuating in itself, but at the same time we have the action of the "primary" shoe upon the "secondary" shoe, which action is called "servo-action" and forms the main subject of the present paper.

BRAKESHOE WITH FIXED ANCHOR PIN

In actual brake design the self-actuation is not produced by wedge effect as has been assumed thus far for the sake of simplicity. Fig. 2 gives an example of a self-actuating shoe. The

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

shoe swings around a fixed anchor pin G and is pressed against the drum by a force P exerted, for instance, by a cam. F_1 denotes the friction force acting upon the brake drum at a certain point, while F_2 is the reaction of this force upon the brakeshoe. It is obvious from the figure that the reactions of the friction forces in all points of the brakeshoe have a tendency to turn the shoe in the same direction in which it is turned by the action of the cam. They contribute, therefore, to the pressure existing between brakeshoe and drum, and make the shoe self-actuating. This, however, is only true for the drum turning in the direction shown by the arrow. If the drum rotates in the opposite direction the reactions of the friction forces counteract the cam force P. For this direction of rotation the shoe has therefore negative self-actuation. For the brakeshoe shown in Fig. 2, L. C. Huck has developed a formula[2] giving the relation between the cam force P and the total amount of all pressures between shoe and drum. The assumptions on which Huck's formula is based are the same as those used in deriving the equations of this paper, and are as follows:

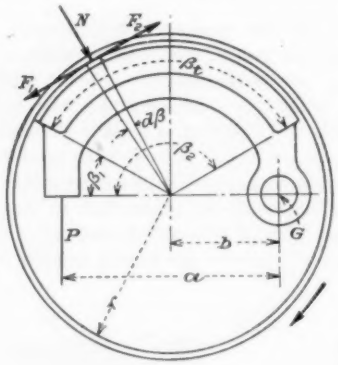


FIG. 2

1 Brake drum and brakeshoe are not flexible
2 The brake is "worn in," i.e., in applying the brake all points of the brake lining touch the drum at the same moment
3 In applying the brake the lining is being compressed, the compression being in proportion to the pressure which is exerted on the lining (Hooke's law).

With these assumptions, Huck's formula gives the sum of all normal pressures between shoe and drum (ΣN) in terms of the force P exerted by the cam, the dimensions a, b, r, and the angles β_1 and β_2 as shown in Fig. 2, and the coefficient of friction f between shoe and drum (see Appendix No. 1).

For the sum of all friction forces acting upon the drum we obtain

$$\Sigma F = f\Sigma N$$

In the case of no-friction, i.e., for $f = 0$, we obtain from Equation [2], Appendix No. 1,

$$\Sigma N' = P\frac{aE}{bB}$$

The negative sign in Equation [2] is valid for the rotation according to the arrow in Fig. 2. In this case $\Sigma N > \Sigma N'$, as our definition of self-actuation called for. For rotation in the opposite direction we have to use the positive sign of the equation and we obtain $\Sigma N < \Sigma N'$. In comparing the self-actuation of different brakeshoes Huck makes use of a certain self-actuation factor giving the self-actuation as a percentage and an actuation constant which is constant for a given brakeshoe. This has not been found to be practical for the calculations which follow. The author will therefore use only one self-actuation factor, which gives the ratio between the sum of all pressures exerted on the drum under the action of the friction forces to the sum of all pressures which would result from the cam force if there were no friction whatsoever. This factor $\Sigma N/\Sigma N'$ is not constant for a given brakeshoe but varies with the coefficient of friction. With the value of ΣN given in Equation [2], Appendix No. 1, we obtain for the brakeshoe swinging around a fixed anchor pin

$$\frac{\Sigma N}{\Sigma N'} = \frac{1}{1 - \frac{f}{B}\left(A + \frac{r}{b}E\right)}$$

This expression becomes infinite when

$$f = \frac{B}{A + \frac{r}{b}E}$$

this being the value of the friction coefficient at which the brake first grabs.

In Fig. 3 the factor of self-actuation is plotted against the coefficient of friction, for a symmetrical brakeshoe in which $b = 7.125$ and $r = 8.45$. The figure shows the curves for different values of $\beta_t = \beta_2 - \beta_1$, the total angle covered by the brakeshoe. In Fig. 4 similar curves are plotted, the difference being that the brakeshoe is here assumed to be symmetrical with respect to a radius drawn at 70 deg. and at 110 deg. to the line connecting the anchor pin and the center of the brake.

The curves show that any amount of self-actuation can easily be obtained, but they show at the same time that the higher the self-actuation the more sensitive is the brake to changes in the friction coefficient. Therefore, if too high a degree of self-actuation is used in a brake design, the brake is liable to grab in some cases, while it will not hold sufficiently in other cases. That is why it is much more important to obtain a brake lining which is affected as little as possible by temperature, moisture, speed, etc., than to obtain one with a high frictional coefficient. Outside of this the curves show that moving the brakeshoe either toward the anchor pin or away from the anchor pin tends to decrease the self-actuation. Lengthening the brakeshoe seems to decrease the self-actuation in most cases, while in a few cases it increases it. There seems to be no way of judging the self-actuation of a brakeshoe or of comparing the self-actuation of two shoes of different design without computation according to Huck's formula.

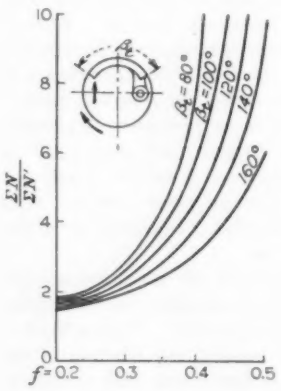


FIG. 3

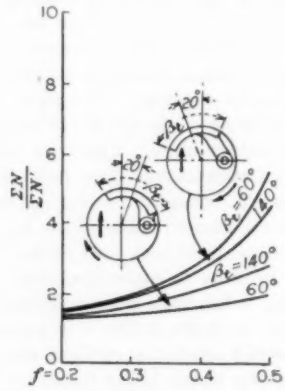


FIG. 4

SERVO-ACTION

Two different designs will be dealt with here. In one design the primary shoe, or the shoe upon which the cam pressure is exerted, is directly connected to the secondary shoe by a pin, the secondary shoe swinging around a fixed anchor pin (Fig. 5). In this case there are only two members between the cam and

[2] *Trans. S.A.E.*, 1926, part 1.

the anchor pin. In the other design (Figs. 10 and 14) both shoes are connected by a link, so that there are three members between the cam and the anchor pin.

Shoes Directly Connected

Referring to Fig. 5, we can apply Huck's formula to the secondary shoe because it swings around the fixed anchor pin G. The primary shoe swings around pin H, which in turn describes a circle around G. In the design shown in Fig. 5 the points G and H are located almost diametrically opposite each other. This means that the motion of point H around G deviates but little from the circular motion around the center of the brake. Consequently the primary shoe can also be considered as swinging around a fixed anchor pin, because a circular motion of point H around the brake center has no effect on the compression of the brake lining of the primary shoe. It would be quite possible to dispense with this approximation and to solve the problem for a primary brakeshoe whose motion is composed of swinging around H and G. The calculations, however, would become far more involved, and it does not seem likely that any brake design could be used successfully in which the points G and H are not located diametrically opposite each other, because it would necessarily result in very unequal wear of the linings of the two shoes.

Huck's formula will therefore be applied to the primary brake-

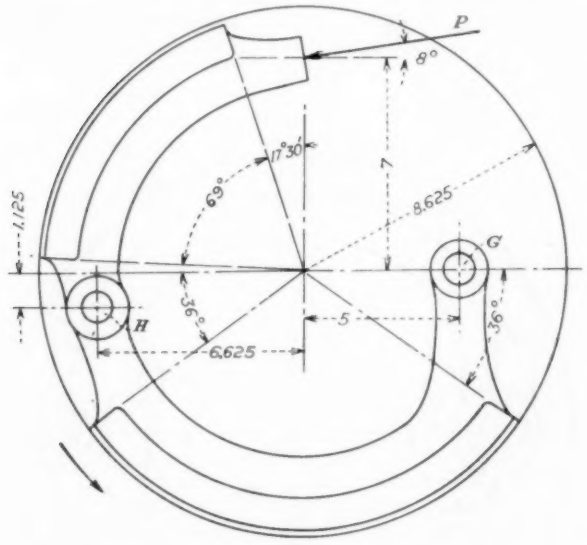


Fig. 5

shoe. However, this formula gives only a means of calculating the sum of all pressures exerted upon the drum and the amount of self-actuation of this shoe. In order to obtain the self-actuation of the total brake, comprising both primary and secondary shoe, it is necessary to determine the force acting upon point H, because it is this force which actuates the secondary shoe. This force can only be calculated from the static equations of the primary brakeshoe.

Referring to Fig. 6, in which the primary brakeshoe is drawn separately, a line is drawn through point H and the center of the brake, and the forces are resolved into their components either parallel or normal to this line. This gives two static equations, while a third equation is obtained from the moments around the center of the brake drum. (See Appendix No. 2.) From these three equations formulas are derived which give the force acting upon the pin H in terms of the cam force P, the coefficient of friction f, and the dimensions of the brake. (Eq. [6] and [7], App. No. 2.) By substituting $f = 0$ in these equations we obtain the relation of these forces in the case of no-friction. Furthermore a coefficient of friction can be computed for which the force on the pin becomes infinite, i.e., for which the brake grabs due to the servo-action. With the force on

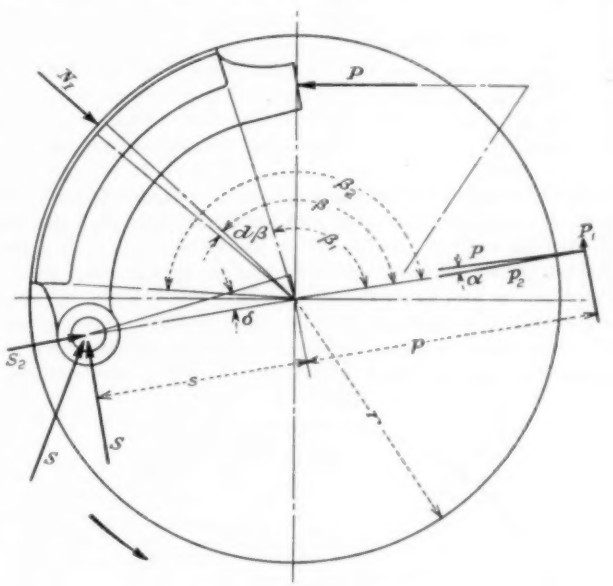


Fig. 6

the pin and the force on the cam we can use Huck's formula to determine the sum of all pressures between the primary shoe and the drum ΣN_{I}, and between the secondary shoe and the drum, ΣN_{II}, and we can furthermore compute these pressures for the case of no-friction, $\Sigma N_{\mathrm{I}}'$ and $\Sigma N_{\mathrm{II}}'$. (Eq. [6a], [6b], [7a], [7b], App. No. 2.) The relation between the sum of all normal pressures for friction and for no friction gives the factor of self-actuation; it is to be noted, however, that the expression $\Sigma N_{\mathrm{II}}/\Sigma N_{\mathrm{II}}'$ found in this way is not the same as if the self-actuation of the secondary shoe had been calculated separately, because it contains at the same time the influence of the servo-action. If now ΣN and $\Sigma N'$ denote the sum of all pressures of the total brake, we have

$$\Sigma N = \Sigma N_{\mathrm{I}} + \Sigma N_{\mathrm{II}}$$
$$\Sigma N' = \Sigma N_{\mathrm{I}}' + \Sigma N_{\mathrm{II}}'$$

and the total self-actuation is

$$\frac{\Sigma N}{\Sigma N'}$$

Example. From the dimensions shown in Fig. 5 the following values can be calculated:

Primary Shoe (Fig. 6):

$p = 264$ $s = 6.75$ $r = 8.625$
$\alpha = 1° 30'$ $\beta_1 = 98°$ $\beta_2 = 167°$
$A = -0.465$ $B = 0.641$ $E = 0.835$

Secondary Shoe (Fig. 7):

$a = 11.625$ $a' = 1.125$
$b = 5$ $r = 8.625$
$\epsilon = 9° 30'$ $\beta_1 = 36°$ $\beta_2 = 144°$
$A = 0$ $B = 1.419$ $E = 1.618$

TABLE 1

		Values of f				
		0	0.2	0.3	0.4	0.5
1	ΣN_I	...	1.58	1.82	2.11	2.46
2	$\Sigma N_I'$	1.37	..	...	...	...
3	$\Sigma N_I/\Sigma N_I'$	...	1.15	1.33	1.54	1.8
4	S_1	...	1.43	1.72	2.1	2.6
5	S_2	...	0.22	0.36	0.45	0.56
6	$S = \sqrt{S_1^2 + S_2^2}$	...	1.45	1.76	2.15	2.66
7	S_1'	1.025	..	...	...	....
8	S_2'	0.225	..	...	...	....
9	$S' = \sqrt{S_1'^2 + S_2'^2}$	1.05	..	...	...	....
10	S/S'	...	1.38	1.68	2.05	2.53
11	ΣN_{II}	...	5.07	8.4	19.3	290
12	$\Sigma N_{II}'$	2.01	..	...	...	...
13	$\Sigma N_{II}/N_{II}'$	...	2.52	4.18	9.6	71
14	ΣN	...	6.65	10.22	21.4	292
15	$\Sigma N'$	3.38	..	...	...	....
16	$\Sigma N/\Sigma N'$	...	1.97	3.03	6.35	86

With these values and by using the above formulas Table 1 has been calculated. In this table all forces are given in multiples of the cam force P. In Fig. 8 the factors of self-actuation are plotted against the coefficient of friction for primary and secondary shoe as well as for the total brake. Fig. 9 shows the sum of all brakeshoe pressures and the force at the connecting pin H. It is found that the self-actuation of the primary shoe is rather low and very little affected by changes in the frictional

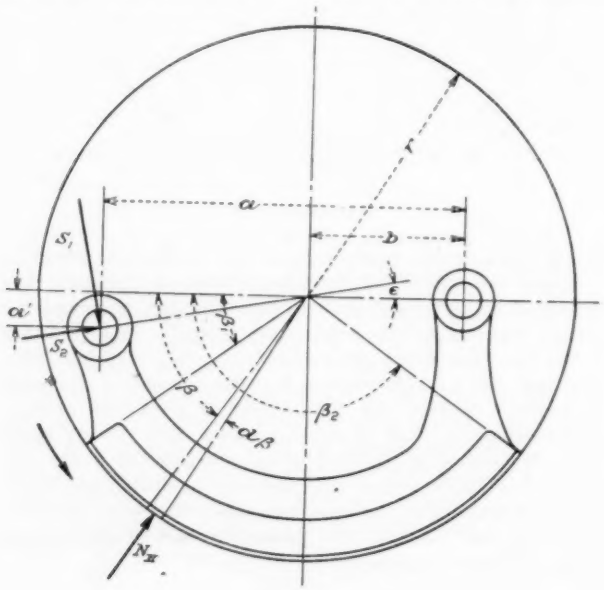


FIG. 7

coefficient. Equally, the servo-action is not influenced much by the friction. The self-actuation of the secondary shoe is very high and correspondingly subjected to great changes. In fact, the secondary shoe is very close to the point of grabbing. According to the curves, one would expect that the lining of the secondary shoe would show most of the wear.

BRAKESHOES CONNECTED BY LINK

Figs. 10 and 14 show brake designs in which a link is inserted between the primary and secondary shoes. The calculation of the primary shoe in this design is essentially different from the one dealt with in the foregoing, because no simplifying assumption can be made regarding the motion of the shoe under the action of the cam force.

In Fig. 11 a primary shoe is shown separately. We shall assume as before that the lining of the shoe touches the drum at all points at the same moment. We shall furthermore consider that point as being the center of the brakeshoe which coincides with the center of the drum when the shoe just touches the drum. If now the shoe is pressed harder against the drum, the center of the shoe will move away from the center of the drum, e denoting the magnitude and γ the direction of this displacement. By resolving all forces into their horizontal and vertical components we obtain three static equations (App. No. 3). After eliminating e and γ a relation between the cam force P and the force in the link S can be found (Eq. [11], App. No. 3). By substituting $P/S = 0$ in this equation we find an equation for that value of the frictional coefficient for which grabbing occurs due to the servo-action. This equation is of the second degree and does not always give a solution, thus indicating that in certain designs grabbing due to the servo-action cannot occur no matter which value of friction prevails.

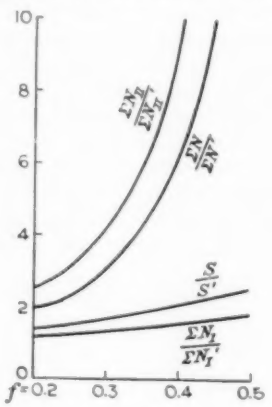


FIG. 8

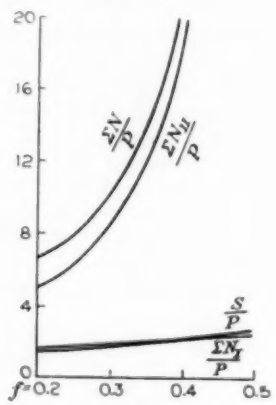


FIG. 9

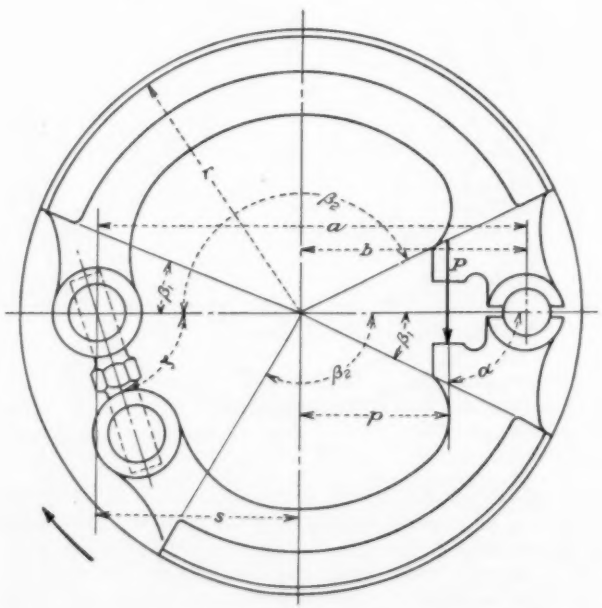


FIG. 10

ΣN_I can be determined in the usual way (Eq. [12], App. No. 3). If we desire to compare the force in the link and the sum of the drum pressures under certain frictional conditions with those which would prevail if there were no friction at all, we easily find a relation between P and S' (Eq. [13], App. No. 3). But we cannot find $\Sigma N_I'$ by substituting $f = 0$ in Eq. [12] because the right-hand member of the equation becomes indefinite. In order to determine $\Sigma N_I'$ we have to go back to the static equations for the case of $f = 0$. The value of the direction

of the displacement for $f = 0$ is given in Eq. [15] (App. No. 3) and $\Sigma N_I'$ in Eq. [14] (App. No. 3).

With these equations the self-actuation of the primary shoe and the force in the connecting link for friction and for no-friction can be determined. The self-actuation of the secondary shoe and its brake-drum pressures can be found in the way already described because the secondary shoe swings around a fixed anchor pin.

Example I (Fig. 10):

PRIMARY SHOE:

p	= 4.625	s	= 6.5625			r	= 8.45
α	= 90°	ζ	= 72°	β_1	= 26° 30′		
A	= 0.28	B	= 1.224	β_2	= 119° 30′	C	= 0.396
D	= 0.424	E	= 1.387				

SECONDARY SHOE:

a	= 13.685	b	= 7.125	r	= 8.45
β_1	= 22°	β_2	= 153°		
A	= 0.033	B	= 1.524	E	= 1.818

With these values Table 2 has been calculated.

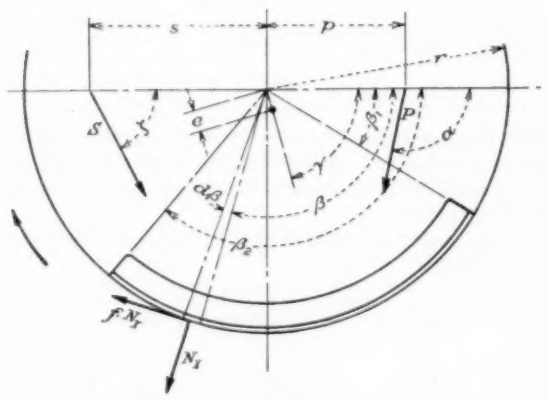


FIG. 11

TABLE 2[a]

		Values of f				
		0	0.2	0.3	0.4	0.5
1	ΣN_I	...	2.44	2.92	3.23	3.45
2	$\Sigma N_I'$	1.88	...	...	...	...
3	$\Sigma N_I/N_I'$	...	1.3	1.55	1.72	1.83
4	S	...	1.4	1.92	2.48	3.07
5	S'	0.735	...	...	...	...
6	S/S'	...	1.91	2.61	3.38	4.18
7	ΣN_{II}	...	4.27	7.32	12.7	23.7
8	$\Sigma N_{II}'$	1.59	...	...	...	...
9	$\Sigma N_{II}/N_{II}'$	...	2.67	4.6	7.95	14.8
10	$\Sigma N = \Sigma N_I + \Sigma N_{II}$	...	6.71	10.24	15.93	27.15
11	$\Sigma N' = \Sigma N_I' + \Sigma N_{II}'$	3.47	...	...	...	...
12	$\Sigma N/\Sigma N'$	...	1.93	2.96	4.6	7.85

[a] All forces in multiples of the cam force P.

Example II (Fig. 14):

PRIMARY SHOE:

p	= 5.25	s	= 5.875	r	= 1	β_1	= 40°
α	= 90°	ζ	= 90°			β_2	= 164°
A	= —0.169	B	= 1.458	C	= 0.702		
D	= —0.367	E	= 1.727				

In this particular instance the equation for P/S cannot be used because the right-hand member of the static Equation [9] disappears for $\alpha = 90°$ and $\zeta = 90°$. The static equations have to be solved for these special conditions. Eq. [16] (App. No. 3) gives the general solution, and with the values given above we obtain:

$$\frac{P}{S} = \frac{4.14 f^2 - 8.05 f + 5.85}{6.92 f^2 + 8.05 f + 5.22}$$

P/S', ΣN_I, and $\Sigma N_I'$ can be calculated in the usual way.

SECONDARY SHOE:

a	= 11.76	b	= 5.62	r	= 1
β_1	= 32° 45′	β_2	= 156° 45′	e	= 16° 45′
A	= —0.068	B	= 1.488	E	= 1.76

The values in Table 3 were then obtained.

Self-actuation and brake-drum pressure are plotted against the coefficient of friction in Figs. 12, 13, 15, and 16. In both cases, but especially in Example I, Figs. 12 and 13, the drum pressure of the primary shoe is considerably higher than in the case of the directly connected shoes (Figs. 8 and 9), while the self-actuation of the secondary shoe is smaller. This seems to be a very desirable condition because, first, it has the result of distributing the wear of the lining more evenly over the two

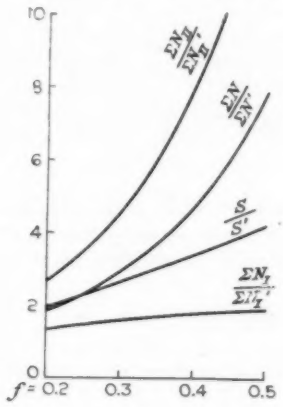


FIG. 12

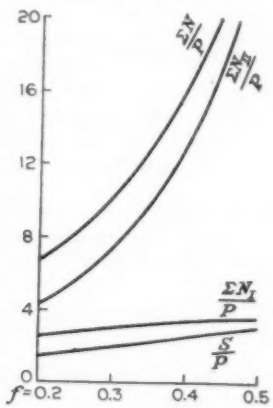


FIG. 13

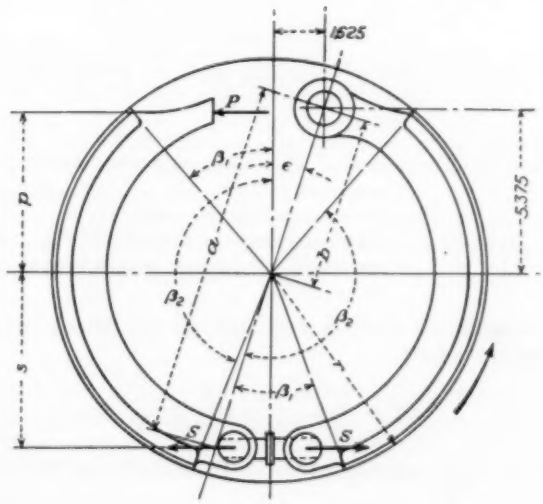


FIG. 14

TABLE 3[a]

		Values of f				
		0	0.2	0.3	0.4	0.5
1	ΣN_I	...	3.03	3.58	4.2	4.9
2	$\Sigma N_I'$	2.15	...	...	...	...
3	$\Sigma N_I/\Sigma N_I'$	...	1.41	1.66	1.95	2.28
4	S	...	1.6	2.17	2.9	3.8
5	S'	0.894	...	...	...	...
6	S/S'	...	1.79	2.43	3.24	4.25
7	ΣN_{II}	...	5.35	9	15.9	31.2
8	$\Sigma N_{II}'$	2.2	...	...	...	...
9	$\Sigma N_{II}/N_{II}'$	...	2.43	4.1	7.24	14.2
10	$\Sigma N = \Sigma N_I + \Sigma N_{II}$	...	8.38	12.58	20.1	36.1
11	$\Sigma N' = \Sigma N_I' + \Sigma N_{II}'$	4.35	...	...	...	...
12	$\Sigma N/\Sigma N'$	...	1.93	2.9	4.63	8.3

[a] All forces in multiples of the cam force P.

shoes, and second, the self-actuation of the primary shoe shows much less tendency to rise suddenly with increasing coefficient of friction. Equally the servo-action seems comparatively little affected by changes in the friction coefficient. In order to obtain reliable and constant brake action, therefore, the aim of the

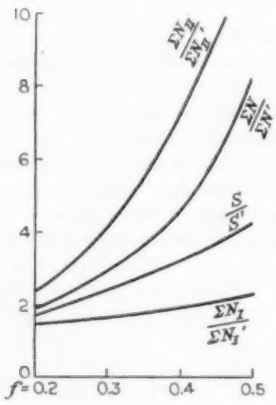


FIG. 15

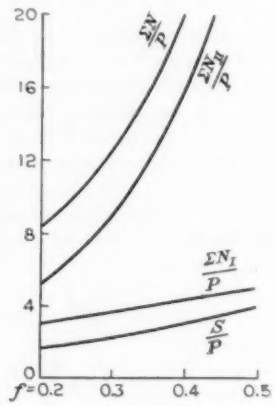


FIG. 16

design should be to have a high self-actuation of the primary shoe and a high servo-action, but to keep the self-actuation of the secondary shoe far enough away from the grabbing point.

Appendix No. 1

For use in this paper Huck's formula has been given the following shape:

$$\Sigma N = P \times \frac{a \int_{\beta_1}^{\beta_2} \sin \beta d\beta}{b \int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta \mp f\left(b \int_{\beta_1}^{\beta_2} \cos \beta \sin \beta d\beta + r \int_{\beta_1}^{\beta_2} \sin \beta d\beta\right)} \quad \ldots\ldots [1]$$

in which ΣN = sum of all normal pressures between shoe and drum
P = force exerted by the cam
a, b, r, β_1 and β_2 = dimensions and angles shown in Fig. 2, and
f = coefficient of friction between shoe and drum.

The integrals can easily be evaluated and the formula can be written in the form

$$\Sigma N = P \times \frac{aE}{bB \mp f(bA + rE)} \quad \ldots\ldots [2]$$

where $A = \int_{\beta_1}^{\beta_2} \cos \beta \sin \beta d\beta = -\frac{1}{4}(\cos 2\beta_2 - \cos 2\beta_1)$

$B = \int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta = -\frac{1}{4}(\sin 2\beta_2 - \sin 2\beta_1) + \frac{1}{2}\beta_2 - \frac{1}{2}\beta_1$

$E = \int_{\beta_1}^{\beta_2} \sin \beta d\beta = -(\cos \beta_2 - \cos \beta_1)$

The formulas become somewhat simpler if the brakeshoe is symmetrical with respect to a line drawn through the center of the drum and normal to the line connecting the center with the anchor pin. In this case $\beta_1 = 2\pi - \beta_2$, $A = 0$, $B = \pi - \beta_1 + \frac{1}{2}\sin 2\beta_1$ and $E = 2 \cos \beta_1$.

Appendix No. 2

The following notation will be used (see Fig. 6):

P, P_1, P_2 = cam force and its components
S, S_1, S_2 = force acting upon pin H and its components
S', S_1', S_2' = same forces for "no-friction"
N_1 = normal force acting between shoe and drum at any point
N_1' = same for "no-friction"
ΣN_1, $\Sigma N_1'$ = sum of all forces N_1 and N_1'
p, s, r = dimensions shown in Fig. 6
w = width of the shoe
α, β, β_1, β_2 = angles shown in Fig. 6
f = coefficient of friction
δ = small angle giving the amount of swing of the shoe when the cam pressure is applied
c = coefficient giving the pressure in lb. per sq. in. which would compress the lining 1 in.

We then have

$$N_1 = wrcs\delta \sin \beta d\beta$$

and we obtain the following three static equations:

$$P_1 + S_1 = wrcs\delta \left(\int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta - f \int_{\beta_1}^{\beta_2} \sin \beta \cos \beta d\beta\right) \quad [3]$$

$$P_2 - S_2 = -wrcs\delta \left(\int_{\beta_1}^{\beta_2} \sin \beta \cos \beta d\beta + f \int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta\right) \quad [4]$$

$$P_1 p - S_1 s = -wrcs\delta r \int_{\beta_1}^{\beta_2} \sin \beta d\beta \quad \ldots\ldots [5]$$

In these equations the expression $wrcs\delta$ can be eliminated and S_1 and S_2 calculated in terms of the cam force P for any given frictional coefficient f, the values of β_1, β_2, p, s, and r being dimensions of the design.

With the substitutions as used in connection with Huck's formula we obtain

$$S_1 = \frac{pB - f(pA - rE)}{sB - f(sA + rE)} P \sin \alpha \quad \ldots\ldots [6]$$

$$S_2 = \left(\cos \alpha + \frac{(p + s)(fB + A)}{sB - f(sA + rE)} \sin \alpha\right) \times P \quad \ldots\ldots [7]$$

For the case of no-friction we have

$$S_1' = \frac{p}{s} \sin \alpha P$$

$$S_2' = \left(\cos \alpha + \frac{(p + s)A}{sB} \sin \alpha\right) P$$

At the same time we find that S_1 and S_2 may become infinite, meaning that grabbing may occur due to excessive servo-action. This would happen when

$$f \geqq \frac{sB}{sA + rE}$$

The sum of all pressures exerted on the drum can be calculated from the three equations. We obtain

$$\Sigma N_1 = wrcs\delta \int_{\beta_1}^{\beta_2} \sin \beta d\beta$$

$$= \frac{(p + s)E}{sB - f(sA + rE)} P \sin \alpha \quad \ldots\ldots [6a]$$

$$\Sigma N_1' = \frac{(p + s)E}{sB} P \sin \alpha \quad \ldots\ldots [6b]$$

Knowing the values for S_1 and S_2 we can now proceed to determine the sum of all pressures exerted by the secondary shoe and the amount of its self-actuation. In Fig. 7 the secondary

brakeshoe is drawn separately. With the notation of that figure and by using Huck's formula we obtain

$$\Sigma N_{II} = (S_1 \cos \epsilon a + S_1 \sin \epsilon a' + S_2 \cos \epsilon a' - S_2 \sin \epsilon a) \frac{E}{bB - f(bA + rE)} \quad [7a]$$

$$\Sigma N_{II}' = (S_1' \cos \epsilon a + S_1' \sin \epsilon a' + S_2' \cos \epsilon a' - S_2' \sin \epsilon a) \frac{E}{bB} \quad [7b]$$

Appendix No. 3

The following notation will be used (Fig. 11):

P = cam force
S = force in the link
N_1 = normal force between shoe and brake drum at any point
$\alpha, \zeta, \beta, \beta_1, \beta_2, \beta, \gamma$ = angles as shown in Fig. 11
p, s, r, e = dimensions as shown in Fig. 11
w = width of the shoe
f = coefficient of friction
c = coefficient of elasticity of the lining in lb. per sq. in. per inch of compression.

N_1', S', ΣN_1, and $\Sigma N_1'$ will be used in the same sense as before.

We then have

$$N_1 = wrce \cos(\beta - \gamma) d\beta$$

and the three static equations:

$$P \sin \alpha + S \sin \zeta = wrce \left[\int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) \sin \beta d\beta - f \int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) \cos \beta d\beta \right] \quad [8]$$

$$P \cos \alpha + S \cos \zeta = wrce \left[\int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) \cos \beta d\beta - f \int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) \sin \beta d\beta \right] \quad [9]$$

$$-Pp \sin \alpha + Ss \sin \zeta = wrcefr \int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) d\beta \quad [10]$$

In these equations the expression $wrce$ as well as the angle γ can be eliminated and S can be calculated in terms of P and the values of f, p, s, r, α, ζ, β_1, and β_2. The following substitutions will be used:

$$A = \int_{\beta_1}^{\beta_2} \cos \beta \sin \beta d\beta = -\tfrac{1}{4}(\cos 2\beta_2 - \cos 2\beta_1)$$

$$B = \int_{\beta_1}^{\beta_2} \sin^2 \beta d\beta = -\tfrac{1}{4}(\sin 2\beta_2 - \sin 2\beta_1) + \tfrac{1}{2}\beta_2 - \tfrac{1}{2}\beta_1$$

$$C = \int_{\beta_1}^{\beta_2} \cos^2 \beta d\beta = \tfrac{1}{4}(\sin 2\beta_2 - \sin 2\beta_1) + \tfrac{1}{2}\beta_2 - \tfrac{1}{2}\beta_1$$

$$D = \int_{\beta_1}^{\beta_2} \cos \beta d\beta = \sin \beta_2 - \sin \beta_1$$

$$E = \int_{\beta_1}^{\beta_2} \sin \beta d\beta = -(\cos \beta_2 - \cos \beta_1)$$

After eliminating $wrce$ we obtain the following two equations:

$$\frac{A - fC + \tan \gamma (B - fA)}{C + fA + \tan \gamma (A + fB)} = \frac{P \sin \alpha + S \sin \zeta}{P \cos \alpha + S \cos \zeta}$$

$$\frac{C + fA + \tan \gamma (A + fB)}{rf(D + \tan \gamma E)} = \frac{P \cos \alpha + S \cos \zeta}{-Pp \sin \alpha + Ss \sin \zeta}$$

from which the following relation between P and S can be obtained:

$$\frac{\frac{P}{S}[C \sin \alpha - A \cos \alpha + f(A \sin \alpha + C \cos \alpha)] + f(A \sin \zeta + C \cos \zeta) + C \sin \zeta - A \cos \zeta}{-\frac{P}{S}[A \sin \alpha - B \cos \alpha + f(B \sin \alpha + A \cos \alpha)] - f(B \sin \zeta + A \cos \zeta) + B \cos \zeta - A \sin \zeta} =$$

$$\frac{\frac{P}{S}[Cp \sin \alpha + f(Ap \sin \alpha + rD \cos \alpha)] + f(rD \cos \zeta - As \sin \zeta) - Cs \sin \zeta}{-\frac{P}{S}[Ap \sin \alpha + f(Bp \sin \alpha + rE \cos \alpha)] - f(rE \cos \zeta - Bs \sin \zeta) + As \sin \zeta}$$

In this form the equation appears to be very involved. In order to obtain P/S a still more complicated equation of the second degree would have to be solved. An essential simplification, however, can be made if instead of dividing the forces into their horizontal and vertical components the axis of the system is chosen in such a way that $\alpha = 90°$. We then have $\sin \alpha = 1$ and $\cos \alpha = 0$ and obtain:

$$\frac{\frac{P}{S}[C + fA] + f(A \sin \zeta + C \cos \zeta) + C \sin \zeta - A \cos \zeta}{-\frac{P}{S}[A + fB] - f(B \sin \zeta + A \cos \zeta) + B \cos \zeta - A \sin \zeta} =$$

$$\frac{\frac{P}{S}[Cp + fAp] + f(rD \cos \zeta - As \sin \zeta] - Cs \sin \zeta}{-\frac{P}{S}[Ap + fBp] - f(rE \cos \zeta - Bs \sin \zeta) + As \sin \zeta} \quad [11]$$

In developing this equation the members containing $(P/S)^2$ disappear and an equation of the first degree is obtained for P/S.

Grabbing of the brake due to excessive servo-action occurs when $P/S = 0$. By substituting $P/S = 0$ in the equation above we obtain an equation of the second degree for f, from which the magnitude of the friction coefficient which will cause grabbing can be calculated. The sum of all brake-drum pressures can be calculated from Equation [10]. We have

$$\Sigma N_1 = wrce \int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) d\beta = \frac{Ss \sin \zeta - Pp \sin \alpha}{rf} \quad [12]$$

In order to compare the force in the link and the sum of the drum pressures under certain frictional conditions with those which would prevail if there were no friction at all, we have to find P/S, and $\Sigma N_1'$. It is evident from Fig. 11 that

$$Pp \sin \alpha = S's \sin \zeta \quad [13]$$

In order to determine $\Sigma N_1'$, however, we have to go back to the static equations, which have the following form for $f = 0$ and $\alpha = 90°$.

$$P + S' \sin \zeta = wrce \int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) \sin \beta d\beta \quad [8a]$$

$$S' \cos \zeta = wrce \int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) \cos \beta d\beta \quad [9a]$$

$$Pp = S's \sin \zeta \quad [10a]$$

We then can write [9a] in the following way:

$$\Sigma N_1' =$$

$$wrce \int_{\beta_1}^{\beta_2} \cos(\beta - \gamma)\, d\beta = S' \cos\zeta \frac{\int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) d\beta}{\int_{\beta_1}^{\beta_2} \cos(\beta - \gamma) \cos\beta d\beta}$$

With the usual substitution we obtain

$$\Sigma N_1' = S' \cos\zeta \frac{D + \tan\gamma\, E}{C + \tan\gamma\, A} \quad \ldots\ldots\ldots\ldots [14]$$

In order to make use of this expression we have to find $\tan\gamma$ for $f = 0$. By eliminating *wrce* in Equations [8*a*] and [9*a*] we obtain the following value:

$$\tan\gamma = \frac{\frac{P}{S'}C + C\sin\zeta - A\cos\zeta}{-\frac{P}{S'}A + B\cos\zeta - A\sin\zeta} \quad \ldots\ldots\ldots [15]$$

For $\alpha = 90°$ and $\zeta = 90°$ the static equations give the following solution:

$$\frac{r\left(1 + \frac{P}{S}\right)}{s - p\frac{P}{S}} = \frac{(A^2 - BC)(1 + f^2)}{f(DA - EC) + f^2(DB - EA)} \quad \ldots [16]$$

Discussion

L. H. Donnell.[3] This very interesting paper shows how comparatively simple analytical methods can greatly clarify a problem which has given considerable difficulty when attacked in a purely empirical manner.

At first thought it appears that in the charts showing the variation of the factor of self-actuation with the coefficient of friction, the ideal condition would be a horizontal line, i.e., no variation with the coefficient of friction. It is interesting to point out that, as a matter of fact, to have the ideal condition of uniform response to a given brake pressure under all conditions, these curves would have to have a *downward* slope to the right, so that a decrease in the coefficient of friction would be made up for by an increase in self-actuation.

All these charts would be clearer if a horizontal line with *unit* ordinate were shown, as this, rather than the line with zero ordinate, represents the real "zero" of self-actuation.

The theory presented in this paper starts with the assumption, first given by Huck, that after the brake lining is worn-in it touches simultaneously at all points and that on further movement the distribution of pressure can be calculated as a statically indeterminate problem, assuming that the brake lining follows Hooke's law and that all other parts are rigid. It seems to the writer that this assumption is likely to be approximately true when the brake lining is nearly new, but that as the brake lining wears there is a strong tendency for the normal pressure to become uniform all over the lining; for if the pressure is greater at some points than it is at others under average brake load, these points will certainly wear faster, and this must tend to relieve the pressure at these points and give a more uniform distribution, under average conditions. It would seem safe to assume that actual conditions will in general lie somewhere between two extremes, one the assumption given in the paper, and the other an assumption of uniformly distributed normal pressures.

[3] University of Michigan. Mem. A.S.M.E.

Author's Closure

The author cannot agree with Mr. Donnell regarding the effect of unequal pressure at different points of the lining. There is a distinct difference between a lining which is worn-in and one which is not. With "worn-in" it is meant that the lining touches the drum in all points at the same time when the brake is being applied. There is very little chance that the lining will do that as long as it is new. There are variations in the curvature of the shoe and in the thickness of the lining. Furthermore, with a fixed anchor pin, there always will be a certain error in the swinging motion of the shoe. Now, as long as this condition exists, in applying the brakes those points which touch the drum first will be subjected to a higher pressure than those which touch later. In this case the author perfectly agrees with Mr. Donnell that the points which carry the higher pressure will wear down faster than the others. This is just what the author calls "wearing-in," and after a certain time it must result in all points touching at the same time.

Now, if this stage is reached, Mr. Donnell believes that the pressure in all points must be and remain equal. This, however, is not the case, just as the pressure at different points of a bearing is not equal even though it may fit the shaft quite snugly and just as the balls of a ball bearing carry different loads at a certain moment. Now, one should not expect that on account of this difference in pressure the lining will begin once more to touch the drum at different moments. The lining will wear down unequally according to the difference in pressure at different points, but it will wear down so that the outside curvature of the lining remains always true with respect to the durm. A simple geometrical analysis shows that this may very well be the case, and this is just what is implied when the matter is calculated as a statically indeterminate problem.

So, as long as it is assumed that the lining follows Hooke's law and that the shoe and the drum are rigid, the calculation seems to be correct. Whether these assumptions are true or not does not depend upon whether or not the lining is worn-in. It will be approximately correct when the brake is applied slightly, and the error will be greater the more flexible all parts are and the harder the brake is applied. Every flexibility will tend to decrease the pressure at all points in which it is high according to the calculation and will increase it correspondingly in other points giving a more uniform distribution. So, while the author cannot agree with Mr. Donnell as regards the effect of the wear, for other reasons, i.e., on account of the flexibility of the parts, he agrees with Mr. Donnell that the actual conditions will lie somewhere between the uniform distribution of pressure and the distribution which is obtained from the foregoing calculations.

APM-52-17

On the Mechanics of the Plastic State of Metals

By A. NADAI,[1] EAST PITTSBURGH, PA.

This paper is a study of the mechanical conditions under which plastic flow in metals occurs. Rules relative to the stresses and the deformation when the stresses reach the limit of plasticity are discussed. Some cases of plastic flow with rotational symmetry around an axis in cylinders are considered. As examples, the case of a bar in combined tension and torsion and that of a hollow cylinder subjected to a high internal pressure and axial forces are treated. The cases of pure bending of a bar with a rectangular cross-section and of pure torsion of a bar with a circular cross-section are discussed, assuming a general law of deformation. Also application of this theory is made to the hollow cylinder subjected to internal pressure. Methods of finding the residual stresses after severe plastic deformation are discussed, and the results obtained in the study should throw light on the problem of these stresses.

I—Consideration of the Various Types of Plastic Material With Respect to the Mechanical Conditions of Plastic Flow in Metals

AS IS GENERALLY known, the terms "solid" and "fluid" which serve in mechanics, physics, and chemistry to differentiate the states of aggregation of matter are in some cases insufficient, especially if the behavior of matter under high pressures acting from all directions or under elevated temperatures is considered. Cases frequently arise in which a gradual change from the solid state into another state can be observed, and the bodies change their shape permanently and continuously to a considerable amount without appreciable increase in the forces at work. In other cases—a common example is found among the ductile metals—bodies undergo permanent changes in shape at normal temperatures. The ductile metals may be bent, drawn, hammered, rolled, etc. On the other hand, certain substances—for example, bituminous solids—are known which have no definite melting point and which gradually change in shape at temperatures at which they usually are considered as solids. Even very small forces are capable of deforming such substances by considerable amounts. Thus, hard asphalt may flow out of an overturned barrel if sufficient time is allowed for the action of the weight of the mass to accelerate the particles thereof. Thus a solid mass flows like a fluid, which would flow out of the barrel under its own weight.

From these observations it must be concluded that all solid materials—even the strongest metals, like steel, or brittle materials, like marble or sandstone—without exception, may, under favorable mechanical conditions, be brought into a new state—the plastic state—in which permanent deformations of considerable magnitude may be produced in them.

It is an interesting problem of mechanics which has many important applications in science and in engineering practice, to try to describe the principal rules which govern the slow movements of the small particles within solid bodies in the plastic state. In the following an attempt is made to analyze some simple cases of plastic flow, and to describe the deformation and the distribution of internal stress in bars or in bodies of simple geometric form which take place when the material is stressed beyond the limit of plasticity.

The first attempts to establish equations for the plastic state of metals, which must be mentioned here, were those of St. Venant.[2] Using as a basis the tests made by Tresca with ductile metals, St. Venant worked out at the beginning of the seventh decade of the last century a theory of the two-dimensional plastic flow of metals and of the plastic deformations of a twisted bar.[3] Later, Duguet[4] extended the theory of the twisted bar under the assumption of a more general stress-strain relation for pure shear.[5]

While the engineer interested in questions of analysis of stress or deformation relies upon a mathematical theory which has been well established and long ago more or less completely worked out when he confines himself to purely elastic deformation, the case is different where plastic flow occurs in parts of stressed bodies or throughout the entire mass of the material. The main reason for this seems to be that the mechanical properties of our principal materials of construction in the plastic state apparently do not follow so general mechanical rules or laws as do the deformations of an isotropic, perfectly elastic body. A step further in the development of rules for the plastic state seems to be to concede this and to try to work out cases under special assumptions as to the mechanical properties of the substances while they are in a plastic state. The newer development of the mechanics of the plastic state of matter has followed these lines. In recent years H. Hencky,[6] Th. v. Kármán,[7] and L. Prandtl[8] have revived the theory of St. Venant, presenting it in many new aspects and extending it to several new cases,[9] while R. v. Mises[10] has mathematically formulated a theory of the plastic state which has proved very valuable. The mechanics of another group of plastic materials, namely, wet, earthy masses, wet clay, etc., has been especially considered by K. v. Terzaghi.[11]

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors and not those of the Society.

[2] *Comptes Rendus*, Paris, 1870, 1871; also *Jl. de Math.*, vol. 2, no. 16, 1871.

[3] A very comprehensive comparison of the conditions of yielding is given by H. M. Westergaard in *Jl. Franklin Inst.*, 1920.

[4] C. Duguet: "Limite d'élasticité et résistance a la rupture," Paris, 1882, vol. 1, p. 157.

[5] This case has also been dealt with independently by P. Ludwik: "Elemente der technologischen Mechanik," Berlin, 1909: and by L. Prandtl in his lectures.

[6] *Zeit. f. ang. Math. u. Mechanik.*, vol. 3, p. 241, 1923; also Proc. 1, International Congress of Applied Mechanics, Delft, 1924.

[7] Proc. 2, International Congress of Applied Mechanics, Zurich, 1926.

[8] Proc. 1, International Congress of Applied Mechanics, Delft, 1924.

[9] The most complete study of the plastic deformations of single metallic crystals is due to G. T. Taylor (Proc. Roy. Soc., London, vol. 102, Bakerian lecture, 1923) and to his pupils. P. W. Bridgman has made very complete investigations of the physical and chemical properties of single crystals.

[10] Göttingen *Nachrichten*, 1913.

[11] "*Erdbaumechanik*," Leipzig, 1925.

One important point should now be mentioned. The question arises whether a body deformed plastically has to be treated under the assumption that all the forces acting on it form a system which is in static equilibrium, or whether the small particles of the body move in a way similar, for example, to that in which the particles of a viscous fluid do; in other words, whether to treat plastic flow in a body as a question of statics or as one of dynamics. If the latter, it would be necessary to assume that the stresses in the body depend in some way on the rate of change of shape. That is, that the components of stress depend on the velocity with which the corresponding components of unit strain change during the deformation. There is no doubt that in certain cases of plastic flow (an example of the plastic flow of amorphous masses other than asphalt under ordinary temperatures is the flow of glass under elevated temperatures) the movement of the small particles of the mass resembles the flow of the so-called viscous fluids as described in hydrodynamics. Taking the first viewpoint, it would be necessary to assume that the stresses in the small element of a plastic mass are in equilibrium, just as in all problems of the statics of perfectly elastic masses.

Take for example the simple case of a straight, cylindrical bar of a ductile (polycrystalline) metal, for instance, mild steel, stretched under two equal forces acting in opposite directions along its axis. If the longitudinal stresses exceed a certain limit, namely, the yield point of the material, the bar will begin to extend permanently. As tests show, in some kinds of low-carbon steel no appreciable increase in the stress is observed within the first two to three per cent of unit elongation when plastic flow begins to spread throughout the length of the bar.[12] At greater unit elongations, however, the stress in the bar will slowly increase with plastic deformation. The rate of increase of the stress with unit elongation will be the higher the lower the temperature of the metal being stretched. From the first point of view, the stress does not depend on the rate of change of length with time or on the velocity with which the bar is stretched. Under ordinary temperatures, tests show that the yield stress of ductile metals such as mild steel or copper changes but little with the velocity of deformation. But at elevated temperatures, this statement of the facts may not be quite sufficient. The same low stress which may not produce any appreciable permanent elongation during an ordinary tensile test at room temperature, may in a polycrystalline metal produce considerable permanent set or creep when the metal is exposed to sufficiently high temperatures. To observe such creep, it may be necessary to stretch the bar under such conditions during a long period of time, i.e., make long-time tests of the bar. Hence not only is the limit of plasticity lowered with increasing temperatures, but a new fact is introduced, that the stresses now depend on the rate of change of shape.

As R. Becker,[13] M. Polanyi and E. Schmid,[14] and others have shown, two principal modes of plastic flow in solids must be distinguished: the plastic flow of single crystals and that of amorphous (non-crystalline) substances. The mechanism of the plastic flow seems to be different in both cases. In crystalline masses plasticity consists in a certain gliding movement: parts of the crystals glide along certain definite planes or directions with respect to each other, these planes and directions having well-defined positions with respect to the crystalline axis. The velocity of this gliding movement or slip in the crystal planes does not seem to be affected much by the shearing stresses under which plasticity starts within the crystal. Schmid and Polanyi have shown that slip in single metal crystals occurs at the lowest temperatures which could be produced and which were close to the absolute zero of the temperature scale, where the thermal agitation of the atoms in the lattices vanishes. On the other hand, the stresses in amorphous plastic substances such as glass or asphalt are known to depend considerably on the velocity of deformation, especially at high temperatures. Obviously, the mechanism of the plastic flow in such substances is quite different from the mechanism of plasticity in crystalline substances. Within the range of softening temperatures of these substances the stress depends on the rate of change of shape, and the coefficient of viscosity which expresses this fact is a pronounced function of the temperature.

The only conclusion which should be drawn from these remarks is that in a great variety of cases of practical importance it is justifiable to treat the plastic flow of polycrystalline substances as a slow movement during which the forces may be considered to remain in equilibrium. This is especially the case when the temperature is comparatively low, that is, when it is a comparatively small fraction of the melting temperature of the substance. Under these conditions, in many cases, the stresses may be determined by considering the substance to be in equilibrium. On the basis of this assumption, as will be shown in the following section, it will now be necessary to consider, in addition, some special mechanical conditions under which plastic flow occurs in materials.

II—Mechanical Rules Concerning the Slow Stationary Plastic Flow in Metals—The Condition of Plasticity

Some observations relative to the magnitude of the deformation are now in order. Several cases may be distinguished:

a The deformation, or the unit strains during plastic flow, is of the same order of magnitude as the elastic deformation. As the latter in most cases of practical importance remains only small in comparison with the smallest dimension of the stressed piece, this means that we can use for the unit elongations and unit shears, which determine the state of deformation at an arbitrary point of the mass, the same mathematical expressions as those used in the theory of elasticity. If we use rectangular coordinates x, y, z, the state of deformation is determined by the three unit elongations ϵ_x, ϵ_y, ϵ_z parallel, respectively to the axes x, y, z, and the three unit shears γ_{xy}, γ_{yz}, γ_{zx}. These six components of strain are given by the following expressions in which ξ, η, ζ are the components of the displacement of the point x, y, z:

$$\left.\begin{aligned} \epsilon_x = \frac{\partial \xi}{\partial x},\ \epsilon_y = \frac{\partial \eta}{\partial y},\ \epsilon_z = \frac{\partial \zeta}{\partial z} \\ \gamma_{xy} = \frac{\partial \xi}{\partial y} + \frac{\partial \eta}{\partial x},\ \gamma_{yz} = \frac{\partial \eta}{\partial z} + \frac{\partial \zeta}{\partial y},\ \gamma_{zx} = \frac{\partial \zeta}{\partial x} + \frac{\partial \xi}{\partial z} \end{aligned}\right\} \quad \ldots\ldots [1]$$

b A more important case is where, during plastic flow, the unit elongations and shears remain small but may become large if compared to the elastic elongations and shears. Take for example a bent or twisted bar of mild steel in which the permanent unit extension or shear at the most severely stretched fibers may amount to from 2 to 4 per cent. The plastic extension is then from 20 to 40 times as large as the maximum elastic extension which could be produced in the material. In such cases it is justifiable to use for the six components of strain ϵ_x, ϵ_y, ϵ_z, γ_{xy}, γ_{yz}, γ_{zx} the same formula [1] as above, but to neglect, if necessary, the *elastic* parts of the strains. If we can confine the deformation to amounts of this order of magnitude, we can furthermore introduce another important simpli-

[12] As is well known, in certain low-carbon steels the stress necessary to maintain the plastic flow when it has once started is even considerably smaller than the stress under which yield first developed in the bar.

[13] *Zeit. f. Physik*, 1927. Also *Zeit. f. Tech. Physik*, vol. 7, 1926, p. 547.

[14] Several papers in *Zeit. f. Physik* (1928).

fication. We can replace the stress-strain diagram for pure tension or compression or for pure shear for a variety of materials by two straight lines as shown in Fig. 1. A good example of a metal with such a stress-strain diagram would be low-carbon steel; another, a metal such as copper or aluminum which has undergone severe cold work, if stressed again in the same sense. Fig. 1 especially represents the behavior of all materials or ductile metals which have a perfect elastic range and a well-defined yield point at which plastic deformation begins. For some other kinds of steel, as is well known, the stress-strain curve does not always change in direction so abruptly as in Fig. 1. But even for such materials, in many cases it will greatly simplify calculation of the stresses to assume that their stress-strain diagrams are similar to that of Fig. 1.

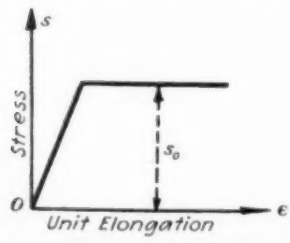


FIG. 1 IDEALIZED STRESS-STRAIN DIAGRAM FOR PLASTIC FLOW

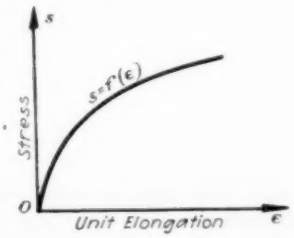


FIG. 2 STRESS-STRAIN DIAGRAM

For such substances it will be possible to distinguish within the stressed body in general an *elastic part* (in which the stresses are below the limit of plasticity), and one or more *plastic parts* where the stresses have reached the yield point.

c A third and more general case would be where the yield stress depends to a considerable extent on the plastic deformation. A well-known example is the stress-strain diagram (for pure tension or compression) for annealed copper or aluminum. It is known that such materials do not possess a limit of plasticity when they are stressed the first time, and that both the elastic and the plastic deformations are at first (under low stresses) of the same magnitude. Such materials show a gradual increase of the stress with deformation (Fig. 2). In such cases the stress in the diagram must be considered as a function of the unit elongation ϵ:

$$s = f(\epsilon) \quad \ldots\ldots [2]$$

which may be determined by means of an ordinary tensile or a compression test or may be assumed in the form of an analytical expression. In this case the deformation may not be limited in amount to some few per cent. It should be noted that if the unit elongations or unit shears exceed certain amounts, the simple relations [1] will no longer be sufficient. These latter will become gradually less exact and the establishment of new expressions will be necessary for the larger deformations. Some examples will be given later in Sections VII and VIII.

While in certain cases of plastic flow it may be advantageous to assume that the yield stress is independent of deformation (as was assumed under *a*), it is necessary to emphasize that this does not mean that no relations exist which would connect the expressions for the components of strain with those for the components of stress. During plastic flow these components depend on each other in a manner which will now be determined. If we use the rectangular coordinates x, y, z, and denote by ϵ_x, ϵ_y, ϵ_z the unit elongation which the material undergoes in the direction parallel to the x, y, and z axes and by γ_{xy}, γ_{yz}, γ_{zx} the corresponding unit shears, then, if we call s_x, s_y, s_z the components of normal stress and s_{xy}, s_{yz}, s_{zx} the three components of shearing stress (the six components of stress which determine the state of stress in a small element at a point with the coordinates x, y, z are shown by the arrows in Fig. 3), the stress components are connected with the strain components in an elastic isotropic material by Hooke's law or by the following six equations:

$$\left.\begin{aligned} \epsilon_x &= \frac{1}{E}(s_x - \nu s_y - \nu s_z); \quad \gamma_{xy} = \frac{s_{xy}}{G} \\ \epsilon_y &= \frac{1}{E}(s_y - \nu s_z - \nu s_x); \quad \gamma_{yz} = \frac{z_{yz}}{G} \\ \epsilon_z &= \frac{1}{E}(s_z - \nu s_x - \nu s_y); \quad \gamma_{zx} = \frac{s_{zx}}{G} \end{aligned}\right\} \quad \ldots\ldots [3]$$

In these equations two independent constants of elasticity enter, namely, Young's modulus of elasticity E and Poisson's ratio ν, while the third constant, the modulus of rigidity G, is given by the former two and is equal to

$$G = \frac{E}{2(1+\nu)}$$

The question arises, by what expressions in the case of plastic flow must these preceding stress-strain relations be replaced?

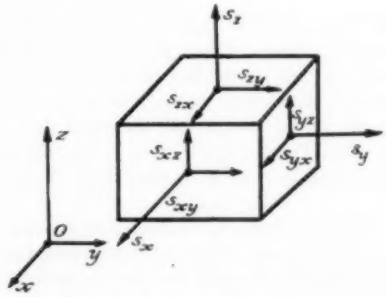


FIG. 3 STRESS COMPONENTS

In general, strain components such, for example, as ϵ_x will have an elastic portion such as ϵ_x' and a plastic (permanent) portion such as ϵ_x'', and the total strain will be equal the sum:

$$\epsilon_x = \epsilon_x' + \epsilon_x'' \quad \ldots\ldots [4]$$

In many cases it may be necessary to deal with these complete expressions for ϵ_x, ϵ_y, ... In other cases, especially for the proposed analysis of stress in a body completely transformed into the plastic state, it will be quite sufficient to neglect the elastic portions of the strains. Under such conditions we may introduce another simplification by neglecting also the small possible elastic changes in volume of the stressed body as well as also some possible small permanent changes therein. Permanent changes of volume have been observed in substances like the metals after plastic flow, but in most of the ductile metals such changes are only perceivable under severe cold working and after great plastic deformation. Under ordinary circumstances (some few per cent of unit elongations) they can be neglected. In other words, we can assume for the sake of simplicity that the volume of the material remains unchanged during or after the plastic deformation. A mathematical expression for this experimental fact is that, for small amounts of deformation, the sum of the unit elongations for the three perpendicular directions x, y, z is zero, or

$$\epsilon_x + \epsilon_y + \epsilon_z = 0 \quad \ldots\ldots [5]$$

In the mechanics of the plastic state of metals it is convenient to use the principal values for strain and stress, denoting them by the subscripts 1, 2, and 3; thus:

ϵ_1, ϵ_2, ϵ_3 are the three principal unit elongations, and
s_1, s_2, s_3 are the corresponding three principal stresses.

In terms of principal elongation, Equation [5] can also be expressed as follows:

$$\epsilon_1 + \epsilon_2 + \epsilon_3 = 0 \ldots\ldots\ldots\ldots\ldots\ldots [5a]$$

A first set of three relations may be obtained for the six quantities ϵ_1, ϵ_2, ϵ_3, s_1, s_2, s_3, by making use of the experimental fact that the three directions of principal strain within a plastic mass

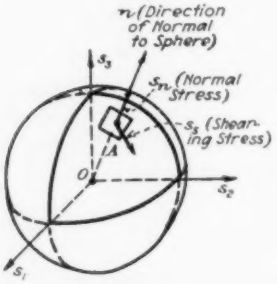


FIG. 4

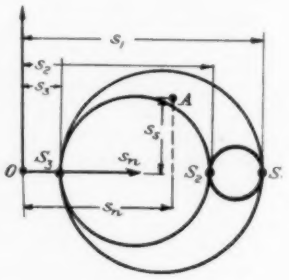


FIG. 5

must always coincide with the corresponding three directions of the principal stresses. The general expressions for these three relations will not be written down here as in general they are fairly complicated (they have been used in the theory of plasticity by M. Lévy), but it will be noted that in the special cases of stress distribution given later, these expressions become simplified considerably. In some important cases they are of a trivial form.

A further relation which, for the sake of brevity, we can perhaps call the *rule of plastic flow*, can be expressed in the simplest way by using Mohr's diagram for the representation of stress and strain.[15]

In general, the position of any plane at the point x, y, z, for which the normal and the shearing stresses s_n and s_s are to be found, can be represented by a plane tangential to a sphere of unit radius or by the normal n thereto (see Fig. 4). The state of stress at the same point of the body can be represented by Mohr's diagram in plotting the normal stress s_n and the shearing stress s_s acting in the plane defined by its normal n in Fig. 4 as the abscissa and the ordinate of a certain point A in Fig. 5. Since to each point A on this sphere (Fig. 4) (that is, to each plane parallel to the corresponding tangent plane of the sphere in the stressed body) there corresponds a certain point A in the stress figure (Fig. 5), to a given curve which may be drawn on the surface of the sphere there will also correspond a certain curve in the plane s_n, s_s. Mohr has shown that to the three principal circles on the sphere, connecting poles for the principal directions, correspond the three principal circles 11, 22, 33. The points of intersection s_1, s_2, s_3 of the three principal circles in Fig. 5 with the s_n-axis determine the principal stresses s_1, s_2, s_3. It should be mentioned that the graphical representation given by Mohr's diagram for stress is also valid for strain. In this case all that is necessary is merely to replace the words "normal stress" and symbol s_n by the words "unit elongation" and symbol ϵ_n, and the words "shearing stress" and symbol s_s by the words "one-half of the unit shear" and symbol $\gamma/2$. We can therefore represent any given state of stress or strain at a point in a plastic mass by the three corresponding principal circles in Mohr's stress or strain diagram.

[15] See S. Timoshenko and J. M. Lessells, "Applied Elasticity," p. 13, 1925; or A. Nadai, "Der bildsame Zustand der Werkstoffe," Springer, Berlin, 1927.

We can now express the "rule of plastic flow" mentioned above as follows:

> During plastic flow, the figure consisting of the three principal stress circles in Mohr's diagram must always remain geometrically similar to the figure consisting of the three principal strain circles.

This is shown by Figs. 6 and 7. From the geometrical proportions of these figures we may obtain the following relations:

$$\mu = \frac{\overline{MS_2}}{\overline{MS_1}}, \quad \nu = \frac{\overline{M'S_2'}}{\overline{M'S_1'}}$$

$$\mu = \nu \quad \text{or} \quad \frac{s_2 - \frac{s_1 + s_3}{2}}{\frac{s_1 - s_3}{2}} = \frac{\epsilon_2 - \frac{\epsilon_1 + \epsilon_3}{2}}{\frac{\epsilon_1 - \epsilon_3}{2}},$$

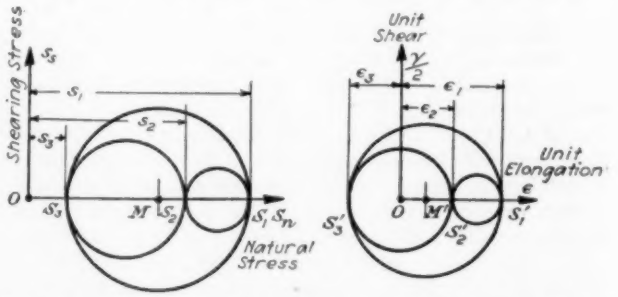
or

$$\frac{s_2 - s_1 + s_2 - s_3}{s_1 - s_3} = \frac{\epsilon_2 - \epsilon_1 + \epsilon_2 - \epsilon_3}{\epsilon_1 - \epsilon_3} \ldots\ldots\ldots [6]$$

which must hold at every point of a plastic mass. It is not difficult to explain the physical meaning of this equation. Neglecting the elastic part of the deformation, compare, for example, the mode of deformation of a small cube cut out from a plastic mass in the following three cases: namely, pure tension, pure compression, and pure shear. It is well known from observation how a small cube will be deformed if subjected to the distributions of stress just mentioned. For example, if a tensile stress acts on two parallel planes of the cube in a direction perpendicular to these planes, the cube will be extended in this direction and will contract in all perpendicular directions by the same amount. Hence the unit contraction in both lateral directions must be equal to half the extension in the direction of the stress, and must be of opposite sign if the volume of the cube is not to be changed. We see that for pure tension $s_1 = s$, $s_2 = s_3 = 0$, $\epsilon_2 = \epsilon_3 = -\epsilon_1/2$, and the quantities defined above, $\mu = -1$, $\nu = -1$, must be equal. In a similar way it can easily be shown that for pure compression we must have $s_1 = -s$, $s_2 = s_3 = 0$, $\epsilon_2 = \epsilon_3 = -\epsilon_1/2$, and $\mu = +1$, $\nu = 1$; and finally for pure shear $s_1 = s$, $s_2 = -s$, $s_3 = 0$, $\epsilon_1 = -\epsilon_2$, $\epsilon_3 = 0$ and $\mu = 0$, $\nu = 0$. We see at once that for all three special cases of stress distribution just mentioned,

$$\mu = \nu \ldots\ldots\ldots\ldots\ldots\ldots [6a]$$

The rule mentioned above assumes now that this condition holds for every kind of stress distribution. Expressed in other words, we can also say that the material yields in such a way that the excess of the intermediate principal stress s_2 over the average value of the two other principal stresses is proportional to the



FIGS. 6 AND 7 MOHR'S CIRCLE DIAGRAMS FOR STRESS AND STRAIN

excess of the intermediate principal strain ϵ_2 over the average value of the two other principal strains.

The three conditions mentioned first, together with those expressed in Equations [5] and [6], that is, a total of five conditions for plastic flow, are equivalent to the six equations which express the stress-strain conditions of an elastic body (Hooke's law). As a total of six quantities must be determined, a sixth equation is necessary. This last equation will express the experimental fact that the stresses have reached the limit of plasticity of the substance. The form for this last condition, which we shall call the "condition of plasticity," has been under investigation for a long time by engineers and physicists in many countries. The more recent tests seem to indicate that in this condition for certain ranges of stress and for certain materials as a first approximation, only the principal stresses enter. O. Mohr's conception that the condition of plasticity would not contain the intermediate principal stress s_2 seems not to have been verified for certain materials, especially the ductile polycrystalline metals. A symmetrical expression of the second degree,

$$(s_1 - s_2)^2 + (s_2 - s_3)^2 + (s_3 - s_1)^2 = 2s_0^2 \quad [7]$$

was found to represent the results of tests in a satisfactory manner. In this equation s_0 is the yield stress in pure tension and, for a certain range of deformation, a constant. This expression was proposed by R. v. Mises. Many tests made by W. Lode[16] and Ros. and Eichinger[17] have verified this condition. In some cases it practically coincides with the expression of an older conception with regard to the yielding of plastic materials, that is, that yield should take place when the maximum shear reaches a certain value (maximum-shear theory); in other cases it differs. These points having been briefly considered, some applications of the foregoing rules will now be discussed.

III—Plastic Flow With Rotational Symmetry Around Axis

Under this heading are comprised all such cases of plastic flow in cylindrical bodies in which the components of stress and of strain are dependent only on a single variable coordinate, namely, the distance r of a point in the body from an axis. As examples may be mentioned the cases of plastic flow in a round bar in pure torsion; in a cylindrical bar of uniform cross-section which has been overstressed through twisting moments; in a round bar subjected to the combined action of tension and torsion; the stress distribution in thick, hollow cylinders or tubes subjected to a high internal (or external) pressure combined with an axial tension or compression or with twist; and the stress distribution in a long cylinder rotating around its axis with a uniform angular speed, if the material has been stressed beyond the yield point.

Taking the axis of the cylinder as the z-axis, in the case of rotational symmetry it is necessary to introduce as the components of normal stress

The radial stress s_r
The tangential stress s_t
The axial stress s_z

Of the three components of the shearing stresses only one has a value other than zero, and this component will be denoted by s_s. The shearing stress s_s acts in the planes perpendicular to the axis of the cylinder in a direction tangential to the circles r = const. The direction of these stress components can be seen in Fig. 8, in which a small element of the cylinder is shown. This element has as boundaries two parallel planes z = const. and $z + dz$ = const., two neighboring concentric cylinders r = const. and $r + dr$ = const., and two planes meeting in the axis of the cylinder and forming a small angle $d\alpha$.

In this case of rotational symmetry the radial normal stress s_r is a principal stress. The two other principal stresses are given by the expressions

$$\left.\begin{aligned} s_1 &= \frac{s_z + s_t}{2} + \sqrt{\frac{(s_z - s_t)^2}{4} + s_s^2} \\ s_2 &= \frac{s_z + s_t}{2} - \sqrt{\frac{(s_z - s_t)^2}{4} + s_s^2} \end{aligned}\right\} \quad [8]$$

It is convenient to use instead of the principal stresses two new stress variables s and s', defined as follows:

$$s = \frac{s_z + s_t}{2} - s_r, \qquad s' = \frac{s_z - s_t}{2} \quad [9]$$

In terms of these stress variables and of the shearing stress s_s the principal stresses can be now expressed as follows:

$$\left.\begin{aligned} s_1 &= s + s_r + \sqrt{s'^2 + s_s^2} \\ s_2 &= s + s_r - \sqrt{s'^2 + s_s^2} \\ s_3 &= s_r \end{aligned}\right\} \quad [10]$$

Hence the stress differences are:

$$\left.\begin{aligned} s_1 - s_2 &= 2\sqrt{s'^2 + s_s^2} \\ s_2 - s_3 &= s - \sqrt{s'^2 + s_s^2} \\ s_1 - s_3 &= s + \sqrt{s'^2 + s_s^2} \end{aligned}\right\} \quad [11]$$

The condition of plasticity (Equation [7])

$$(s_1 - s_2)^2 + (s_2 - s_3)^2 + (s_3 - s_1)^2 = 2s_0^2 \quad [12]$$

takes with these variables the simpler form

$$s^2 + 3(s'^2 + s_s^2) = s_0^2 \quad [13]$$

A set of formulas analogous to [8], [9], and [10] can be written down for the strain components, using the expressions for the principal strains.

Let ϵ_r, ϵ_t, ϵ_z be the unit elongations in the radial, tangential, and axial directions, and γ be the unit shear. Then the three principal strains are given by

$$\left.\begin{aligned} 2\epsilon_1 &= \epsilon_z + \epsilon_t + \sqrt{(\epsilon_z - \epsilon_t)^2 + \gamma^2} \\ 2\epsilon_2 &= \epsilon_z + \epsilon_t - \sqrt{(\epsilon_z - \epsilon_t)^2 + \gamma^2} \\ \epsilon_3 &= \epsilon_r \end{aligned}\right\} \quad [14]$$

Using in analogy to [9] as new strain variables

$$\left.\begin{aligned} \epsilon &= \epsilon_z + \epsilon_t - 2\epsilon_r \\ \epsilon' &= \epsilon_z - \epsilon_t \end{aligned}\right\} \quad [15]$$

and the unit shear γ, the principal strains become

$$\left.\begin{aligned} 2\epsilon_1 &= \epsilon + 2\epsilon_r + \sqrt{\epsilon'^2 + \gamma^2} \\ 2\epsilon_2 &= \epsilon + 2\epsilon_r - \sqrt{\epsilon'^2 + \gamma^2} \\ \epsilon_3 &= \epsilon_r \end{aligned}\right\} \quad [16]$$

[16] "Der Einfluss der mittleren Hauptspannung auf das Fliessen der Metalle." V.D.I. Forschungsarbeiten, Heft 303, 1928.

[17] Proc. 2. International Congress of Applied Mechanics, Zurich, 1926.

and hence the principal strain differences are

$$\left.\begin{aligned} \epsilon_1 - \epsilon_2 &= \sqrt{\epsilon'^2 + \gamma^2} \\ \epsilon_2 - \epsilon_3 &= \frac{\epsilon}{2} - \frac{1}{2}\sqrt{\epsilon'^2 + \gamma^2} \\ \epsilon_1 - \epsilon_3 &= \frac{\epsilon}{2} + \frac{1}{2}\sqrt{\epsilon'^2 + \gamma^2} \end{aligned}\right\} \quad \ldots\ldots\ldots [17]$$

It is now necessary to express the special conditions of plastic flow in a cylinder, which will satisfy the conditions of symmetry. It must be recalled that all the points of the cylinder can move only (1) radially (all points situated on a circle r = const. are displaced by equal amounts), and (2) axially (all points situated in a plane which is perpendicular to the axis of the cylinder are displaced by equal amounts parallel to the axis of the cylinder). To these two deformations must be added the twist around the axis of the cylinder. If ρ is the radial displacement (the change of the radii r in the radial direction), then the tangential and radial unit elongations ϵ_t, ϵ_r are given by

$$\epsilon_t = \frac{\rho}{r}; \qquad \epsilon_r = \frac{d\rho}{dr} \quad \ldots\ldots\ldots [18]$$

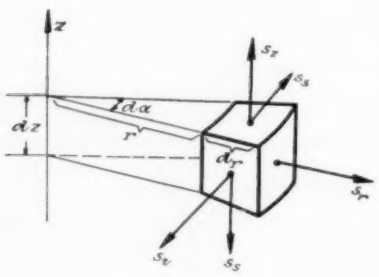


FIG. 8



FIG. 9

The condition of compatibility of these two relations involves

$$\frac{d(r\epsilon_t)}{dr} - \epsilon_r = 0 \quad \ldots\ldots\ldots [19]$$

Taking in account the fact that the volume of the small element (Fig. 8) will not be changed by the plastic flow,

$$\epsilon_r + \epsilon_t + \epsilon_z = 0 \quad \ldots\ldots\ldots [20]$$

In this last equation and in all later relations the axial unit elongation ϵ_z must be considered as a constant, as all cross-sections of the cylinder remain plane. From [19] and [20] we find by integration the expressions of the radial and tangential unit elongations as functions of r:

$$\left.\begin{aligned} \epsilon_r &= -\frac{\epsilon_z}{2} - \frac{c_1}{r^2} \\ \epsilon_t &= -\frac{\epsilon_z}{2} + \frac{c_1}{r^2} \end{aligned}\right\} \quad \ldots\ldots\ldots [21]$$

Here c_1 is a constant of integration. For a hollow cylinder c_1 in general will not be zero, but for a solid cylinder it must be taken equal to zero. Finally we have to express the fact that the cross-sections of the cylinder turn around its axis through a uniform angle of twist. If ϑ is the twist per unit of length, the unit shear at the distance r from the axis will be

$$\gamma = \vartheta r \quad \ldots\ldots\ldots [22]$$

Using the expressions [15], [21], and [22], condition [6] takes the form:

$$\epsilon' = \frac{3\epsilon_z}{2} - \frac{c_1}{r^2}$$

$$\epsilon = \frac{3\epsilon_z}{2} + \frac{3c_1}{r^2}$$

$$\frac{s - 3\sqrt{s'^2 + s_z^2}}{s + \sqrt{s'^2 + s_z^2}} = \frac{\epsilon - 3\sqrt{\epsilon'^2 + \gamma^2}}{\epsilon + \sqrt{\epsilon'^2 + \gamma^2}} \quad \ldots\ldots\ldots [23]$$

But as

$$\frac{A - 3B}{A + B} = \frac{A' - 3B'}{A' + B'}$$

is equivalent to

$$\frac{B}{A + B} = \frac{B'}{A' + B'}$$

[23] can be also expressed as

$$\frac{s^2}{s'^2 + s_z^2} = \frac{\epsilon^2}{\epsilon'^2 + \gamma^2} \quad \ldots\ldots\ldots [24]$$

Finally the coincidence of the principal directions for the stress s, and for the unit elongation ϵ_t must be expressed. This will be the case if

$$\frac{s_z - s_t}{2s_z} = \frac{\epsilon_z - \epsilon_t}{\gamma} \quad \ldots\ldots\ldots [25]$$

or by using s, s' (cf. [9] and [15]) and ϵ_t, ϵ',

$$\frac{s'}{s_z} = \frac{\epsilon'}{\gamma} \quad \ldots\ldots\ldots [26]$$

It is worth while to note that we thus obtain three *finite* equations for the three unknown quantities s,s',s_z, namely, [13], [24], and [26]. To sum up the preceding, we shall write down again these three equations as follows:

$$s^2 + 3(s'^2 + s_z^2) = s_0^2 \quad \ldots\ldots\ldots [27]$$

$$\frac{s^2}{s'^2 + s_z^2} = \frac{\epsilon^2}{\epsilon'^2 + \gamma^2} \quad \ldots\ldots\ldots [28]$$

$$\frac{s'}{s_z} = \frac{\epsilon'}{\gamma} \quad \ldots\ldots\ldots [29]$$

The right-hand members of the last two equations are to be considered as known functions of the independent variable r and given if we use the expressions found in [21] and [22] for $\epsilon_r,\epsilon_t,\gamma$. To solve these three equations we write

$$\delta^2 = \epsilon^2 + 3(\epsilon'^2 + \gamma^2) \quad \ldots\ldots\ldots [30]$$

and find that the three unknown stress variables s,s',s_z are given by the simple expressions:

$$s = \pm s_0 \frac{\epsilon}{\delta}; \qquad s' = \pm s_0 \frac{\epsilon'}{\delta}; \qquad s_z = \pm s_0 \frac{\gamma}{\delta} \quad \ldots [31]$$

Finally the axial and tangential normal stresses s_z,s_t are found:

$$\left.\begin{aligned} s_z &= s + s_r + s' \\ s_t &= s + s_r - s' \end{aligned}\right\} \quad \ldots\ldots\ldots [32]$$

and the radial stress is given by the differential equation

$$\frac{r ds_r}{dr} = s_t - s_r \quad \ldots\ldots\ldots [33]$$

by which the equilibrium of the small element in Fig. 8 is expressed. The last equation leads to

$$\frac{r ds_r}{dr} = s - s' \quad \ldots\ldots [34]$$

which serves for the calculation of the radial stress. In the next section some applications of the preceding formulae to special cases are given.

IV—Plastic Flow in Cylindrical Bar Under Combined Action of Axial Tension and of Torsion (Fig. 9)[17a]

As already mentioned, for a solid cylinder the integration constant c_1 must be taken equal to zero. Hence the radial and the tangential unit elongations in such a bar must be equal to half of the axial unit elongation ϵ_z.

$$\epsilon_r = \epsilon_t = -\frac{\epsilon_z}{2} \quad \ldots\ldots [35]$$

and this latter has to be considered as a constant. The shear is given as before by $\gamma = r\vartheta$. With these expressions the variables ϵ and ϵ' can be found from [15] as

$$\left.\begin{aligned} \epsilon &= \epsilon_z + \epsilon_t - 2\epsilon_r = \frac{3\epsilon_z}{2} \\ \epsilon' &= \epsilon_z - \epsilon_t = \frac{3\epsilon_z}{2} \end{aligned}\right\} \quad \ldots\ldots [36]$$

and the quantity δ as equal to:

$$\delta = 3\epsilon_z \sqrt{1 + \frac{r^2\vartheta^2}{3\epsilon_z^2}} \quad \ldots\ldots [37]$$

As it is not essential the double sign $\pm$ can be cancelled, if only tension in axial direction is to be considered. (cf. [31], etc.). The ratio s'/s_z, taken from Equation [29] is now equal to

$$\frac{s'}{s_z} = \frac{\epsilon'}{\gamma} = \frac{3\epsilon_z}{2\vartheta r} \quad \ldots\ldots [38]$$

Using the expression just found for δ in [37] and [38], the three unknown quantities s, s', s_r are given:

$$s = s' = \frac{s_0}{2} \frac{1}{\sqrt{1 + \frac{\vartheta^2 r^2}{3\epsilon_z^2}}} \quad \ldots\ldots [39]$$

$$s_r = \frac{s_0 \vartheta r}{3\epsilon_z} \frac{1}{\sqrt{1 + \frac{\vartheta^2 r^2}{3\epsilon_z^2}}} \quad \ldots\ldots [40]$$

These expressions can be also written in the following very simple form:

$$\left.\begin{aligned} s = s' &= \frac{s_0}{2} \cos\alpha \\ s_z &= \frac{s_0}{\sqrt{3}} \sin\alpha \end{aligned}\right\} \quad \ldots\ldots [41]$$

or also

$$\left.\begin{aligned} s = s' &= \frac{s_0}{2} \frac{1}{\sqrt{1+u^2}} \\ s_z &= \frac{s_0}{\sqrt{3}} \frac{u}{\sqrt{1+u^2}} \end{aligned}\right\} \quad \ldots\ldots [42]$$

[17a] Dr. W. Lode in Elberfeld, with whom the author has worked during several years, has contributed much to the analysis of this case.

by writing

$$\frac{\vartheta r}{\sqrt{3}\cdot\epsilon_z} = \tan\alpha = u \quad \ldots\ldots [43]$$

As can be seen from [32] and [33], since $s = s'$, both the radial and the tangential stresses s_r, s_t vanish in this case. Hence the distribution of stress in a cylindrical bar subjected to the combined action of an axial load and a twisting moment is given by the formulas:

$$\left.\begin{aligned} &\text{Shearing stress:} \quad s_s = \frac{s_0}{\sqrt{3}} \sin\alpha = \frac{s_0}{\sqrt{3}} \frac{u}{\sqrt{1+u^2}} \\ &\text{Axial normal stress:} \quad s_z = s_0 \cos\alpha = s_0 \frac{1}{\sqrt{1+u^2}} \end{aligned}\right\} \quad [44]$$

These stresses are acting in a bar, in which the material yields from the center to the surface. Here s_0 is the yield stress, ϑ the angle of twist per unit of length, ϵ_z the unit elongation of

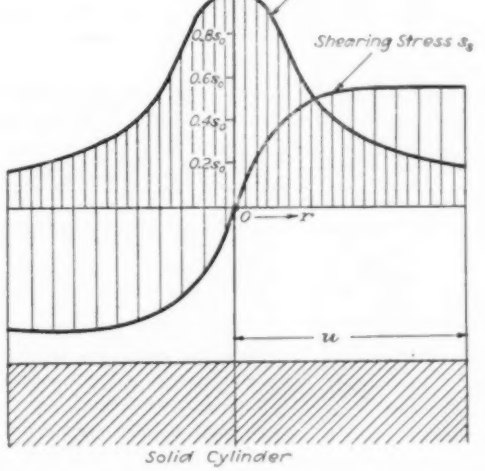


Fig. 10 Stress Distribution Under Plastic Flow in Cylindrical Bar Subjected to the Combined Action of an Axial Pull and a Twisting Moment

the bar in the axial direction, r the distance of a point from the axis of the cylinder, and α and u are given by $\frac{\vartheta r}{\sqrt{3}\epsilon_z} = \tan\alpha = u$. Using a for the radius of the bar and for

$$\frac{\vartheta a}{\sqrt{3}\,\epsilon_z} = \tan\alpha_a = u_a \quad \ldots\ldots [45]$$

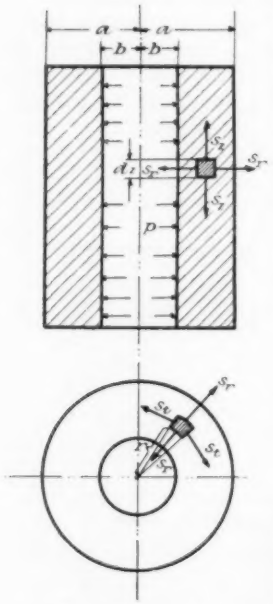
the axial force or the axial load P and the twisting moment M are given as

$$P = 2\pi \int_0^a s_z r dr = 2\pi s_0 a^2 \left(\frac{\sqrt{1+u_a^2} - 1}{u_a^2} \right) \quad \ldots\ldots [46]$$

$$M = 2\pi \int_0^a s_s r^2 dr = \frac{2\pi s_0 a^3}{3\sqrt{3}} \frac{(u_a^2 - 2)\sqrt{1+u_a^2} + 2}{u_a^3} \quad \ldots\ldots [47]$$

In Fig. 10 an example of a stress distribution is given for the value $u_a = 5.25$. As can be seen in the figure, the normal stress becomes a maximum at the center of the cylinder. As at this point the shearing stress vanishes, the normal stress becomes equal to the yield stress in pure tension. To verify

our formulas for P and M, we may derive from them the special cases of pure tension and of pure torsion. In the first case u becomes small, hence the axial stress $s_z = \frac{s_0}{\sqrt{1+u^2}} \sim = s_0$ = const. and $P = \pi a^2 s_0$, and $M = 0$. (The value of P becomes indeterminate for $u_a = 0$: $P = 2\pi a^2 s_0 \frac{\sqrt{1+u_a^2}-1}{u_a^2} = \frac{0}{0}$, but the true value of this fraction is $1/2$, and therefore $P = \pi a^2 s_0$ = yield stress times the area of the cross-section of the cylinder.) In the case of pure torsion, on the contrary, u becomes infinite, hence $s_z = 0$, $s_t = \frac{s_0}{\sqrt{3}}$ and the twisting moment $M = \frac{2\pi s_0 a^3}{3\sqrt{3}} = \frac{2\pi s_t a^3}{3}$. The same moment would have been found by taking the shearing stress s_t as a constant $= s_0/\sqrt{3}$ through the circular cross-section of the bar.



FIGS. 11 AND 12

The formulas derived above for the resultant force P of the axial stresses s_z and for the moment of twist M may be also expressed by using the parameter α_a given in [45] instead of using u_a as

$$P = 2\pi a^2 s_0 \tan \frac{\alpha_a}{2} \quad [48]$$

$$M = \frac{Pa}{3\sqrt{3}} \cot \alpha_a \tan \frac{\alpha_a}{2} \quad [49]$$

V—Thick Hollow Cylinder Subjected to Internal Pressure and Axial Load

If a hollow cylinder with thick walls is submitted to the combined action of high internal pressure and axial load, no shearing stresses s_z will be produced in the planes z = const. In the following case the whole tube will be considered to yield under the forces acting. Let $r = a$ be the outer radius, $r = b$ the inner one, and p the pressure on the cylinder of radius $r = b$ (Figs. 11 and 12).

As the unit elongations ϵ_r and ϵ_t taken from Equation [21] will be

$$\left.\begin{aligned} \epsilon_r &= -\frac{\epsilon_z}{2} - \frac{c_1}{r^2} \\ \epsilon_t &= -\frac{\epsilon_z}{2} + \frac{c_1}{r^2} \end{aligned}\right\} \quad [50]$$

and the unit shear γ is now zero, the quantities ϵ and ϵ' defined in [15] will now equal

$$\epsilon = \frac{3\epsilon_z}{2} + \frac{3c_1}{r^2} \quad \epsilon' = \frac{3\epsilon_z}{2} - \frac{c_1}{r^2} \quad [51]$$

In order to simplify the following expressions, a new independent variable will be used, defined by

$$u = \frac{2c_1}{\epsilon_z r^2 \sqrt{3}} \quad [52]$$

Let the tangential unit elongation ϵ_t at the outer surface of the cylinder $r = a$ be

$$\epsilon_t = \epsilon_a = -\frac{\epsilon_z}{2} + \frac{c_1}{a^2} \quad [53]$$

then

$$u = \frac{1}{\sqrt{3}} \left(1 + \frac{2\epsilon_a}{\epsilon_z}\right) \frac{a^2}{r^2} \quad [54]$$

$$\epsilon = \frac{3\epsilon_z}{2}\left(1 + u\sqrt{3}\right) \qquad \epsilon' = \frac{3\epsilon_z}{2}\left(1 - \frac{u}{\sqrt{3}}\right) \quad [55]$$

The stress variables s and s' in [27] and [29] must satisfy the equations:

$$\frac{s'}{s} = \frac{\epsilon'}{\epsilon} = \frac{\sqrt{3}-u}{\sqrt{3}+3u} \quad [56]$$

$$s^2 + 3s'^2 = s_0^2 \quad [57]$$

From these equations s and s' are found:

$$s = \frac{s_0}{2\sqrt{3}} \frac{\sqrt{3}+3u}{\sqrt{1+u^2}}, \quad s' = \frac{s_0}{2\sqrt{3}} \frac{\sqrt{3}-u}{\sqrt{1+u^2}} \quad [58]$$

Replacing now s and s' by their expressions given in [9], the three components of stress are found:

$$\left.\begin{aligned} &\text{Axial stress:} && s_z = s + s' + s_r \\ &\text{Tangential stress:} && s_t = s - s' + s_r \\ &\text{Radial stress:} && s_r = \int \frac{(s-s')}{r}\, dr + c_2 \\ & && = \pm \frac{s_0}{\sqrt{3}} \int \frac{du}{\sqrt{1+u^2}} + c_2 \end{aligned}\right\} \quad [59]$$

Integrating the last expression and taking, for $r = a$,

$$u = u_a = \frac{1}{\sqrt{3}} \left(1 + \frac{2\epsilon_a}{\epsilon_z}\right) \quad [60]$$

and for $r = a$ the stress $s_r = 0$, it is seen that the radial stress is

$$s_r = \frac{s_0}{\sqrt{3}} \log_e \frac{u_a + \sqrt{1+u_a^2}}{u + \sqrt{1+u^2}} \quad [61]$$

This equation now permits us to find an expression for the pressure p under which the whole thick tube will yield. At the inner surface of the tube, for $r = b$, $s = -p_r$. If $r = b$, the variable u becomes

$$u = u_b = \frac{a^2}{b^2} u_a \quad [62]$$

Hence the pressure p is

$$p = \frac{s_0}{\sqrt{3}} \log_e \frac{u_b + \sqrt{1+u_b^2}}{u_a + \sqrt{1+u_a^2}} \quad [63]$$

As u varies inversely as the square of r, if $b < r < a$ the variable u is $u_a < u < u_b$. Both u_a and u_b depend on the ratio ϵ_a/ϵ_z. The meaning of the integration constant u_a has already been explained by Equation [60]. The distribution of stress depends mainly on two parameters: the ratio of the outer to the inner radius a/b, and the ratio of the tangential unit elongation ϵ_a at the outer surface $r = a$ to the axial unit elongation ϵ_z. In a tube with a given ratio a/b, different stress distributions will be obtained for different ratios ϵ_a/ϵ_z. The pressure p necessary to produce yield in the whole tube will also depend on ϵ_a/ϵ_z.

One special case which must be included in Equation [59] is that of the yielding of a hollow cylinder under axial pull. We know that for the simple case of pure tension, the lateral plastic contraction of the tube must be half of the longitudinal expansion if the volume of the material is not to be changed. Hence $\epsilon_a = -\epsilon_z/2$, and in this case, u and u_a must vanish. Equations [59] show that for pure tension the stresses $s_r = s_t = 0$, and the axial normal stress s_z becomes

$$s_z = \frac{\sqrt{3}}{\sqrt{3}} s_0 = s_0 = \text{const.}$$

An interesting special case is that of a thick tube under high internal pressure p which is not allowed to expand or contract in axial direction. In this case the unit elongation parallel to the axis of the tube must be $\epsilon_z = 0$ and the integration constants u_a and u_b become both infinite, but their ratio $u_b/u_a = a^2/b^2$ remains finite. Here a and b designate the outer and the

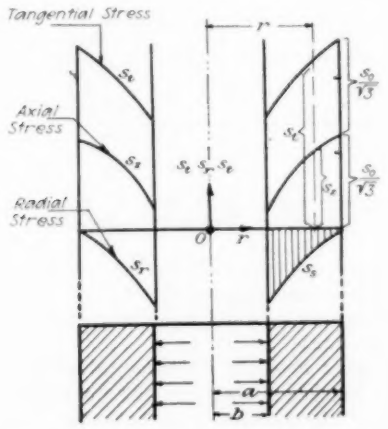


FIG. 13 PLASTIC FLOW IN THICK HOLLOW CYLINDER: STRESS DISTRIBUTION WHEN AXIAL ELONGATION $\epsilon = 0$

inner radius of the hollow cylinder. Equation [63] for the internal pressure p simplifies in this case and leads to the formula

$$p = \frac{2s_0}{\sqrt{3}} \log_e \frac{a}{b} \quad \text{[64]}$$

Equations [59] and [61] now take the simpler and symmetrical forms:

$$\left.\begin{aligned} &\text{Radial stress:} && s_r = -\frac{2s_0}{\sqrt{3}} \log_e \frac{a}{r} \\ &\text{Tangential stress:} && s_t = \frac{2s_0}{\sqrt{3}}\left(1 - \log_e \frac{a}{r}\right) \\ &\text{Axial stress:} && s_z = \frac{s_0}{\sqrt{3}}\left(1 - 2\log_e \frac{a}{r}\right) \end{aligned}\right\} \quad \text{[65]}$$

The resultant of the axial stresses s_z can now be easily calculated and is

$$P = \frac{\pi s_0}{\sqrt{3}}\left[\frac{3(a^2 - b^2)}{2} - b^2 \log_e \frac{a}{b}\right] \quad \text{[66]}$$

As the expression can never vanish for any tube with a *finite* thickness $a - b$, we see that if a thick tube exposed to an internal pressure p is not allowed to expand or to contract in the axial direction during the plastic flow, normal stresses s_z in the axial direction of considerable amount must act, and these will prevent axial contraction. The axial normal stress is at every point of the tube equal to the average value of the radial and the tangential normal stresses. The three normal stresses are given by three equidistant logarithmic curves. The curves for the axial and the tangential stress s_z and s_t are obtained from the curve of the radial stress s_r by shifting the latter in the vertical direction (see Fig. 13) by the amounts $\frac{s_0}{\sqrt{3}}$ and $\frac{2s_0}{\sqrt{3}}$.

VI—ROTATING CYLINDER

In this case, it is necessary to modify the equilibrium condition. Let ω be the angular speed of the cylinder, λ the specific gravity of the material, g the acceleration of gravity, and a the radius of the cylinder. In the condition of equilibrium [33] an additional term must be introduced with regard to the action of the centrifugal forces:

$$\frac{ds_r}{dr} = \frac{s_t - s_r}{r} - \frac{\lambda\omega^2}{g} \cdot r \quad \text{[67]}$$

In a solid cylinder again the constant c_1 in Equation [50] must be taken equal to zero; hence the radial and the tangential unit elongations in the cylinder will become again equal to

$$\epsilon_r = \epsilon_t = -\frac{\epsilon_z}{2} \quad \text{[68]}$$

A consequence of this fact will be that in an infinitely long, solid rotating cylinder, both the radial and the tangential stresses become equal, or

$$s_r = s_t \quad \text{[69]}$$

Hence the equation of equilibrium takes the form

$$\frac{ds_r}{dr} = -\frac{\lambda\omega^2}{g} \cdot r$$

whence by integration

$$s_r = \frac{\lambda\omega^2}{2g}(a^2 - r^2) \quad \text{[70]}$$

The condition of plasticity [7] is now

$$s_t - s_z = \pm s_0 \quad \text{[71]}$$

where s_0 designates the yield stress for pure tension. Two distinct cases are obtained, depending on whether the upper or the lower sign is taken. Taking the upper sign, the axial stress is given by

$$s_z = s_t - s_0 \quad \text{[72]}$$

From the condition that the resultant of the axial stresses

$$2\pi \int_0^a s_z r dr = 0 \quad \text{[73]}$$

should become zero, the peripheral speed $a\omega$ is found under which the entire cylinder will yield, or

$$a\omega = 2\sqrt{\frac{s_0 g}{\lambda}} \quad \text{[74]}$$

It may be of interest to compare this speed with the velocity at which first yield occurs in a rotating cylinder. As shown in the theory of elasticity, in a rotating solid cylinder of radius a, yield starts at the center if the peripheral velocity is equal to

$$u\omega' = 2\sqrt{\frac{2(1-\nu)\, s_0 g}{(3-4\nu)\lambda}} \quad \text{[75]}$$

where ν is Poisson's ratio. In this formula it is assumed that the resultant of the axial stresses $s_z = 0$. Assuming that $\nu = 1/3$, this velocity is found to be

$$a\omega' = 1.764\sqrt{\frac{s_0 g}{\lambda}} \quad [76]$$

Comparing [74] with [76] it is seen that with an increase of speed above the velocity given in [76] of about 13 per cent, the entire cylinder will be brought into the plastic state. The distribution of stress under which the whole cylinder becomes plastic is as follows:

$$\left.\begin{aligned} &\text{Radial and tangential stresses: } s_r = s_t = 2s_0\left(1 - \frac{r^2}{a^2}\right) \\ &\text{Axial stress: } \qquad s_z = s_0\left(1 - 2\frac{r^2}{a^2}\right) \end{aligned}\right\} \quad [77]$$

where s_0 is the yield stress in pure tension.

The central part of the cylinder receives tensile, and the peripheral parts compressive stresses in the axial direction, but the action of the tensile stresses balances that of the compressive, so that no resultant axial force is acting.

VII—Pure Bending of Bars With Arbitrary Law of Deformation

In this and the following sections some applications of the theory will be shown for plastic or inelastic materials which do not follow the simple rules for deformation as stated and applied in the preceding sections. The theory of bending of a slender cylinder or prismatic bar, or of torsion of a cylindrical bar when the deformation does not obey Hooke's law, may be taken as examples. The calculation is based on certain assumptions with regard to the deformation, which are also made in the ordinary theory of elastic bending or torsion. The validity of these assumptions for bending in the case of non-elastic materials such as cast iron or plastic materials such as steel has been carefully studied, especially by C. v. Bach,[18] E. Meyer,[19] and Herbert,[20] who have proved by their tests that the assumptions are well founded.

The theory of bending of a narrow bar with an arbitrary form of the cross-section when the deformation obeys a general law as stated under (*c*), Section II, may be worked out in a different, but relatively simple manner.[21]

In the following, only the case of pure bending of a bar with rectangular cross-section will be treated, as this is the simplest case to which these calculations may be applied. It is assumed that the bar is bent by the action of two equal bending moments M acting at the ends of the bar in opposite directions and in one of the principal planes of inertia of the cross-section.

Two problems, the direct and the inverse problem, are here of interest. In the former, given the stress-strain diagram of the material for pure tension and compression, the stress s may be taken as a known function $f(\epsilon)$ of the unit elongation ϵ, or,

$$s = f(\epsilon) \quad [78]$$

The problem then is to determine how the bending moment M of the bar will increase with the deformation of the bar, or how it will depend on the deflection y or on the slope β of the tangent to the elastic line of the bar for a certain given gage length l.

The relation which gives the bending moment M as a function of the slope β of the tangent to the elastic line will be denoted by $M = F(\beta)$.

In the inverse problem, the bending moment M in a bar of rectangular cross-section of a given material is to be considered as a given function:

$$M = F(\beta) \quad [79]$$

as it depends on the deformation of the bar. This function is found by observation. The problem here is to find or construct the stress-strain diagram

$$s = f(\epsilon)$$

in pure tension and compression for this material. As will be seen, the direct problem involves an integration and the inverse problem a differentiation, both of which may be effected by graphic methods.

The assumptions on which the calculation of the bending

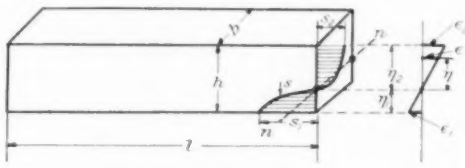


Fig. 14 Bending Stresses in Bar of Inelastic or Plastic Material Under Pure Bending

stresses is based are the same as those made in the ordinary theory of bending, namely, that the cross-sections remain plane during bending and that the unit elongations ϵ remain small compared with unity.

(*a*) *The Direct Problem.* In order to determine the distribution of the bending stress s over the cross-section, the stress-strain curve of the material must be known from tests. We can assume that it is given by an analytical expression or graphically by the function

$$s = f(\epsilon)$$

of the unit elongation ϵ.

Referring to Fig. 14, let us take $n - n$ as the position of the neutral axis in the cross-section in which the bending stress s vanishes and η as the distance of a point in the cross-section from the neutral axis where the bending stress is s. An element dx at the distance η from the neutral axis undergoes an extension and the unit strain ϵ at this distance will be

$$\epsilon = \frac{\eta}{\rho} \quad [80]$$

In this ρ is the radius of curvature of the bent bar. If h is the height and b the width of the cross-section, and if η_1 and η_2 are the distances of the lower and upper fibers of the bar, where the greatest tensile and compression stresses s_1 and s_2 may act, then

$$\eta_1 + \eta_2 = h \quad [81]$$

and

$$\epsilon_1 + \epsilon_2 = \frac{h}{\rho} \quad [82]$$

If ρ and ϵ_1 are given, then ϵ_2 is determined by Equation [82].

The equilibrium of the forces acting in a cross-section is expressed by the two equations:

$$\int s\,d\eta = 0 \quad [83]$$

[18] C. v. Bach und R. Baumann: "Elastizität und Festigkeit, Berlin, 1924. 9th ed., p. 259.

[19] *Zeitschrift des Vereines deutscher Ingenieure*, vol. 52 (1908), p. 167.

[20] Mitteilungen Forschungsarbeiten V.D.I., Heft 89, 1910.

[21] A. Nadai: "Der bildsame Zustand der Werkstoffe," Berlin, 1927, p. 111.

$$M = b \int s\eta d\eta \quad \ldots\ldots [84]$$

where the integrals must be taken between the limits $\eta = -\eta_1$ and $\eta = \eta_2$. Taking s from [78] and replacing the variable η by ϵ Equation [83] leads to the condition

$$\int_{-\epsilon_1}^{\epsilon_2} f(\epsilon) d\epsilon = 0 \quad \ldots\ldots [85]$$

Let A_1 and A_2 be the absolute values of the area in the stress-strain diagram of Fig. 15.

$$A_1 = -\int_{-\epsilon}^{0} f(\epsilon) d\epsilon; \quad A_2 = \int_{0}^{\epsilon_2} f(\epsilon) d\epsilon \quad \ldots\ldots [86]$$

Then condition [85], which may also be expressed as

$$A_1 = A_2 \quad \ldots\ldots [87]$$

determines for any given stress-strain curve $s = f(\epsilon)$ to any arbitrarily chosen value of ϵ_1 a certain value $\epsilon_2 = \varphi(\epsilon_1)$. Equation [82] then determines the corresponding value of the radius of curvature ρ of the bent bar, and from Equation [84] the bending moment M is found:

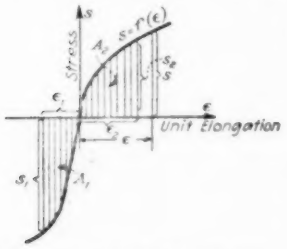


FIG. 15 STRESS-STRAIN DIAGRAM IN TENSION AND COMPRESSION

$$M = b\rho^2 \int_{-\epsilon_1}^{\epsilon_2} f(\epsilon)\, \epsilon d\epsilon \quad [88]$$

The slope β of the tangent to the elastic line of the bent bar is, for the gage length l,

$$\beta = \frac{l}{2\rho} = \frac{l(\epsilon_1 + \epsilon_2)}{2\rho} \quad \ldots [89]$$

The deflection y of the bar for the length l is (Fig. 16).

$$y = \rho \frac{\beta^2}{2} = \frac{l^2}{8\rho} \quad \ldots\ldots [90]$$

Equations [88] and [89] determine the unknown relation

$$M = F(\beta) \quad \ldots\ldots [91]$$

This provides a convenient experimental proof of the assumptions made in the calculations, as the bending moment M and the slope β can be determined on the one hand by [91] and on the other hand by observation from a bending test.

(*b*) *The Inverse Problem.* In order to obtain the stress-strain diagram $s = f(\epsilon)$ in pure tension and compression *from a bending test*, it must be assumed that in a bending test with a straight bar with rectangular cross-section it has been observed how the bending moment M increases with the slope at the ends of the gage length l of the bent middle line of the bar. Then M is known as a function of β, or

$$M = F(\beta) \quad \ldots\ldots [92]$$

In order to obtain the stress-strain diagram

$$s = f(\epsilon) \quad \ldots\ldots [93]$$

from the bending test, a start is made with Equation [88], in which ρ is replaced by the angle β using [89], where $\rho = l/2\beta$. Multiplying both sides of [88] by β^2 we obtain:

$$M\beta^2 = \frac{bl^2}{4} \int_{-\epsilon_1}^{\epsilon_2} f(\epsilon)\, \epsilon d\epsilon \quad \ldots\ldots [94]$$

The left-hand member of [94] has now to be considered as a known function of β since the moment is known to be $M = F(\beta)$. Hence the right-hand member must be a function of β. Differentiating [94] with respect to the variable β,

$$\frac{d}{d\beta}(M\beta^2) = \frac{bl^2}{4}\left[f(\epsilon_2)\epsilon_2 \frac{d\epsilon_2}{d\beta} - f(-\epsilon_1)(-\epsilon_1)\frac{d(-\epsilon_1)}{d\beta}\right] \quad [95]$$

Now

$$f(\epsilon_2) = s_2; \quad f(-\epsilon_1) = -s_1 \quad \ldots\ldots [96]$$

are the tensile and compression stresses at the extreme fibers $\eta = \eta_2$ and $\eta = -\eta_1$ of the bar, and from the equilibrium conditions [86] or [87] it is seen that the small increments of the area A_1 and A_2, that is

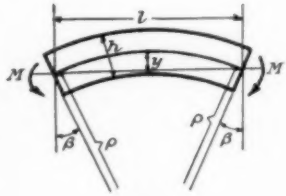


FIG. 16

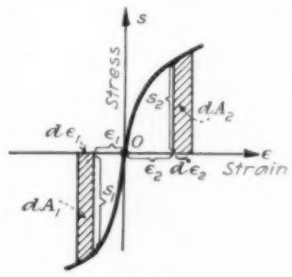


FIG. 17

$$dA_1 = dA_2 \quad \text{or} \quad s_1 d\epsilon_1 = s_2 d\epsilon_2 \quad \ldots\ldots [97]$$

must always be equal. (Fig. 17.) Differentiating [82],

$$d\epsilon_1 + d\epsilon_2 = \frac{2h}{l} d\beta \quad \ldots\ldots [98]$$

From [97] and [98],

$$\left.\begin{aligned} d\epsilon_1 &= \frac{2hs_1}{l(s_1 + s_2)} d\beta \\ d\epsilon_2 &= \frac{2hs_2}{l(s_1 + s_2)} d\beta \end{aligned}\right\} \quad \ldots\ldots [99]$$

Using [96]–[99] it is found that [95] can also be written as

$$d(M\beta^2) = \frac{bl^2}{4}(s_2\epsilon_2 d\epsilon_2 + s_1\epsilon_1 d\epsilon_1) \quad \ldots\ldots [100]$$

$$\frac{1}{\beta}\frac{d}{d\beta}(M\beta^2) = bhl \cdot \frac{s_1 s_2}{s_1 + s_2} \quad \ldots\ldots [101]$$

The left-hand side of this last equation is a known function of β. Equation [101] and the condition [97] that

$$\frac{d\epsilon_1}{d\epsilon_2} = \frac{s_2}{s_1} \quad \ldots\ldots [102]$$

permit us to calculate the unknown stresses s_1 and s_2 at the extreme fibers of the bar for different values of the unit elongations ϵ_1 and ϵ_2; that is, to construct from the bending test the stress-strain diagram $s = f(\epsilon)$ of the material for pure tension and compression.

VIII—TORSION OF BAR OF CIRCULAR CROSS-SECTION AND HAVING AN ARBITRARY LAW OF DEFORMATION

In a similar way as for bending, the theory of torsion can be extended to the case of a cylindrical bar and having a general

law of deformation.[22] If a solid or hollow cylinder is submitted to the action of two pure twisting moments, it will twist uniformly. If the material is stressed beyond the limit of plasticity, the angle of twist per unit of length may become permanent. But as the cross-sections after twist will remain plane and will simply turn around the axis of the cylinder (we assume that there is no perceptible change in the volume of the coaxial cylindrical shells with plastic deformation), coaxial cylindrical shells are deformed with the increasing twisting moment in proportion to their radii. As they do not exert stresses in a radial direction on one another, we can assume that the unit shear γ at a distance r from the center of the circular cross-sections is proportional to the radius r, or

$$\gamma = \vartheta r \ldots\ldots\ldots\ldots\ldots\ldots [103]$$

The constant of proportionality ϑ is the angle of twist per unit of length.

A consequence of the manner in which the small elements of a cylinder are deformed by pure shear is that the internal

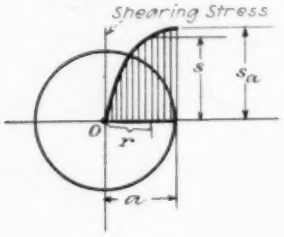


FIG. 18

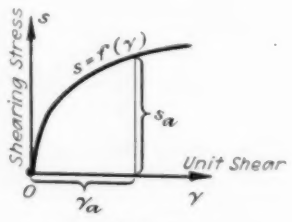


FIG. 19

stresses acting in the plane cross-sections of the bar are only shearing stresses which are perpendicular to the radii r. In general, the shearing stress s will depend on the amount of deformation, that is, it will be a function of the unit shear γ

$$s = f(\gamma) \ldots\ldots\ldots\ldots\ldots\ldots [104]$$

This relation may be called the "stress-strain diagram of the material in pure shear," and can be considered as given analytically or graphically (see Fig. 19).

(*a*) *Direct Problem.* If the stress-strain diagram of the material in pure shear is given it is easy to find the magnitude of the twisting moment M which corresponds to a given angle of twist ϑ. The moment M for a solid cylinder is

$$M = 2\pi \int_0^a sr^2 dr \ldots\ldots\ldots\ldots\ldots [105]$$

The integration variable r being, for a given angle of twist ϑ, proportional to the unit shear γ, can be replaced by γ/ϑ and s by $f(\gamma)$. Hence:

$$M = \frac{2\pi}{\vartheta^3} \int_0^{\gamma_a} f(\gamma)\gamma^2 d\gamma \ldots\ldots\ldots\ldots\ldots [106]$$

The upper limit of this integral is $\gamma_a = \vartheta a$, where a is the radius of the cylinder. Expressed in words:

> For a given unit angular twist the corresponding twisting moment M of a solid cylinder subjected to torsion beyond the plastic limit is equal to the moment of inertia of the area under the stress-strain diagram $s = f(\gamma)$ taken about the s-axis (see Fig. 19) and extending to the unit shear $\gamma_a = \vartheta a$, this being multiplied by $2\pi/\vartheta^3$.

(*b*) *The Inverse Problem.* Conversely, if the moment twist curve $M = F(\vartheta)$ is known from an ordinary torsion test, it is possible to determine from it the stress-strain diagram of the material in pure shear. Multiply [106] by ϑ^3. Then the left-hand member side of [106] is a known function of the variable ϑ. Hence the right-hand member of [107] must be a function of ϑ. The upper limit γ_a contains the angle ϑ, as

$$\gamma_a = \vartheta a \ldots\ldots\ldots\ldots\ldots\ldots [107]$$

Differentiating both sides of [106],

$$\frac{d}{d\vartheta}(M\vartheta^3) = 2\pi f(\gamma_a)\,\gamma_a^2 \frac{d\gamma_a}{d\vartheta} \ldots\ldots\ldots [108]$$

Here we can replace γ_a by ϑa. $f(\gamma_a)$ is the shearing stress s_a at

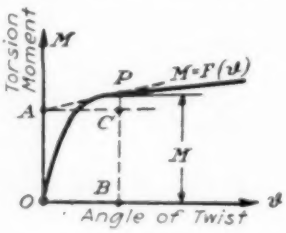


FIG. 20

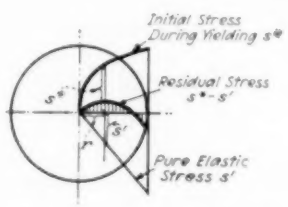


FIG. 21 RESIDUAL STRESSES IN TWISTED CYLINDRICAL BAR AFTER YIELDING

the extreme fiber $r = a$. Thus, solving [108] for s_a we find the shearing stress for $r = a$ in the twisted bar equal to

$$s_a = \frac{1}{2\pi a^3 \vartheta^2} \frac{d}{d\vartheta}(M\vartheta^3) = \frac{1}{2\pi a^3}\left[\vartheta \frac{dM}{d\vartheta} + 3M\right] \ldots [109]$$

The shearing stress s_a corresponding to a unit shear $\gamma_a = a\vartheta$ can be obtained by a geometrical construction from the moment-twist diagram $M = F(\vartheta)$ taken from a torsion test as follows: Referring to Fig. 20, project on the ordinate PB the tangent AP to the moment-twist curve $M = F(\vartheta)$. Add to this length PC three times the value of the moment M and divide this sum by $2\pi a^3$. The form of the stress-strain diagram for pure shear and the stress distribution in cylindrical bars under the combined action of an axial load and a twisting moment have been studied carefully for ductile metals by P. Ludwik[23] and Seigle.[24]

IX—PLASTIC FLOW IN A THICK-WALLED TUBE UNDER THE ACTION OF INTERNAL PRESSURE FOR AN ARBITRARY LAW OF DEFORMATION

The case of a thick-walled tube can also be treated in a similar manner as the preceding two cases by assuming a general law of deformation if the cylinder expands only in a radial direction, that is, if the tube does not change in dimension parallel to its axis. Assuming that the axial unit elongation $\epsilon_z = 0$, the other two unit elongations ϵ_r and ϵ_t must be equal and have opposite signs, or

$$\epsilon_r = -\epsilon_t \ldots\ldots\ldots\ldots\ldots\ldots [110]$$

Let a be the outer and b the inner radius of the tube and ρ the

[22] The theory seems to have been established the first time by C. Duguet in his book, previously referred to, which appeared in 1883. P. Ludwik ("Elemente der technologischen Mechanik," Berlin, 1909) and L. Prandtl (lectures) have independently derived essentially the same rules. Tests in torsion with plastic metals which support the assumptions have been made by W. Bader (Dissertation, University of Göttingen, 1927).

[23] "Elemente der technische Mechanik," Berlin 1909. See also Ludwik and Scheu, *Stahl und Eisen*, vol. 45 (1925), p. 373.

[24] Comptes rendus, Paris, 1925. Page 98, also Genie Civil, Paris, years 1925, 1926, also Revue de metallurgie, 1925, p. 374.

radial displacement of a point at a distance r from the axis of the tube. As $\epsilon_r = \frac{d\rho}{dr}$ and $\epsilon_t = \frac{\rho}{r}$, the radial displacement ρ must satisfy the condition that

$$\frac{d\rho}{dr} = -\frac{\rho}{r} \quad \text{[111]}$$

and

$$\rho = \rho_a \frac{a}{r} \quad \text{[111}a\text{]}$$

Obviously the integration constant ρ_a is one-half of the total expansion of the external diameter $2a$ of the tube. As $\epsilon_t = 0$ and $\epsilon_r = -\epsilon_t$, the rule of plastic flow requires that the axial normal stress shall become equal to the average value of the radial and tangential stresses:

$$s_z = \frac{s_r + s_t}{2} \quad \text{[112]}$$

(otherwise the figure of the three principal circles in Mohr's stress-diagram would not be similar to the figure of the three principal circles of strain). In both diagrams the two smaller circles must have equal diameters. Under these circumstances the state of strain in every element of the tube is a pure shear. At a distance r from the axis of the tube this unit shear must be

$$\gamma = \epsilon_t - \epsilon_r = 2\epsilon_t = 2\rho_a \frac{a}{r^2} \quad \text{[113]}$$

Evidently all the small elements of the tube are deformed in a similar manner as are the elements in a solid cylinder under the action of a twisting moment as described in Section VIII. The only difference is that the shear in the tube under internal pressure has now a different direction from that in the cylinder under torsion. It is easy now to see that the stresses in the tube can be determined if the stress-strain diagram of the material in pure shear is known. This latter could, for example, be determined by an ordinary torsion test with a bar of circular cross-section as shown in Section VIII. Let the law of plastic deformation in pure shear be given by

$$s = f(\gamma) \quad \text{[114]}$$

where s is the maximum shearing stress and γ the unit shear. For a hollow cylinder stressed by an internal pressure p, the maximum shearing stress is given by

$$s = \frac{s_t - s_r}{2} \quad \text{[115]}$$

and the unit shear γ given by

$$\gamma = \epsilon_t - \epsilon_r = 2\rho_a \frac{a}{r^2} \quad \text{[116]}$$

These two expressions have then to be combined with the condition of equilibrium of the stresses, or

$$\frac{ds_r}{dr} = \frac{s_t - s_r}{r} \quad \text{[117]}$$

Integrating this expression between the limits r and a and taking into consideration that the radial stress s_r must vanish for $r = a$, the radial stress is found to be

$$s_r = \int_r^a \frac{(s_t - s_r)}{r}\, dr \quad \text{[118]}$$

Replacing here the variable r by the variable γ, using [116],

$$\frac{dr}{r} = -\frac{d\gamma}{2\gamma}; \quad s_t - s_r = 2s = 2f(\gamma)$$

the radial and the tangential stresses are found to be

$$\left.\begin{aligned} s_r &= -\int_{\gamma_a}^{\gamma} f(\gamma)\,\frac{d\gamma}{\gamma} \\ s_t &= 2s + s_r = 2f(\gamma) - \int_{\gamma_a}^{\gamma} f(\gamma)\,\frac{d\gamma}{\gamma} \end{aligned}\right\} \quad \text{[119]}$$

Since for $r = b$ the radial stress $s_r = -p$ from the last equation for s_r, the pressure p can be calculated under which the whole tube will yield. It is given by the integral:

$$p = \int_{\gamma_b}^{\gamma_a} f(\gamma)\,\frac{d\gamma}{\gamma} \quad \text{[120]}$$

The limits of this integral are

$$\gamma_a = \frac{2\rho_a}{a}, \qquad \gamma_b = \frac{2\rho_a a}{b^2} = \gamma_a \frac{a^2}{b^2} \quad \text{[121]}$$

So far none of the above expressions contain any assumptions with regard to the special form of the stress-strain diagram $s = f(\gamma)$ in pure shear, and they can therefore be used for any material. We can, however, try to employ some analytical expression for this function and to calculate the corresponding stresses. Taking, for example, as the law of deformation a power function

$$s = s_0 \left(\frac{\gamma}{\gamma_0}\right)^{\frac{m}{2}} \quad \text{[122]}$$

the components of stress will be given by functions containing $(a/r)^m$. In [122] the constants s_0 and γ_0 are two arbitrarily chosen, corresponding values of stress and unit shear in the stress-strain diagram for pure shear.[23] The exponent m must be taken $0 < m < 2$. Using the above relation for the stress diagram, the radial and tangential stresses can be calculated from [119]. They are:

$$\text{Radial stress:} \quad s_r = -\frac{pb^m}{a^m - b^m}\left(\frac{a^m}{r^m} - 1\right)$$

$$\text{Tangential stress:} \quad s_t = \frac{d}{dr}(rs_r) = \frac{pb^m}{a^m - b^m}\left[(m-1)\frac{a^m}{r^m} + 1\right] \quad \text{[123]}$$

They determine the distribution of stress in a thick-walled tube yielding under an internal pressure p if (1) the tube does not expand or contract in the axial direction and if (2) the stress-strain relation can be expressed by a power function.

The limiting case, that the exponent $m = 2$, may be mentioned here as in this case

$$\left.\begin{aligned} s_r &= -p\,\frac{b^2}{a^2 - b^2}\left(\frac{a^2}{r^2} - 1\right) \\ s_t &= \frac{pb^2}{a^2 - b^2}\left(\frac{a^2}{r^2} + 1\right) \end{aligned}\right\} \quad \text{[124]}$$

[23] It should be noted that a power function cannot express quite accurately the behavior of the stresses for low values of stresses for such plastic materials as the ductile metals. If $m < 2$ the angle of the tangent of the stress-strain curve at the origin $s = 0$ would become equal $\pi/2$, while all diagrams for ductile metals have a slope smaller than 90 deg. An exception is the case where $m = 2$, in which case the diagram is correct also for $s = 0$.

These are exactly the same expressions for the radial stress s_r and the tangential stress s_t which would have been found by assuming Hooke's law or a linear function for the stress-strain diagram. Thus we see that the deformation of a perfectly elastic tube is included as a special case in [119].

The stresses are given at the inner surface (for $r = b$):

$$s_r = -p; \qquad s_t = p \cdot \frac{(m-1)a^m + b^m}{a^m - b^m}$$

and at the outer surface (for $r = a$):

$$s_r = 0; \qquad s_t = \frac{mb^m p}{a^m - b^m}$$

X—Residual Stresses

As is well known, in an isotropic elastic material internal stresses can be produced without having an external load acting on the surface of the stressed piece of material. In a ring of elastic material, for example, spring steel, internal stresses will be set up if the ring is slotted radially and then the ends are brought together and welded or fastened with a bolt. In an elastic material, internal stresses of considerable amount may result also from non-uniform heating. In such cases one speaks of "thermal stresses." These will result, for instance, in the case of a thick-walled tube cooled on the outside and conducting a hot fluid. On the other hand, if in a severely stressed material one portion is permanently strained and the external forces are removed, in general, in the neighborhood of the permanently strained region, stresses will remain. With suitable means or methods these permanent or "residual stresses" can be demonstrated, as was proved by the late E. Heyn in cold-rolled or drawn bars of ductile metals. If, for example, a hardened steel ball is pressed into a piece of mild steel, the material in the neighborhood of the indentation after removal of the load cannot remain unstressed. While the ball was sinking into the soft metal and the material was yielding under the ball, very considerable compressive stresses must have acted not only in a direction parallel to the load but also in a direction perpendicular to it. The portion plastically distorted under the ball behaves like a wedge driven into a soft mass, and the neighboring parts must be stressed like a tree trunk into which a wedge has been driven. Thus after the removal of the external pressure the lateral stresses will not disappear and will form a system of residual stresses. Such internal stresses and severe residual stresses present after plastic deformation are familar to steel men and engineers on account of their deleterious action and destructive consequences in the case of pressed, forged, or rolled bars. Sometimes fractures are caused by the action of these stresses if the forged or rolled pieces are not annealed before they are further deformed. Methods of estimating these residual stresses may therefore have several practical applications.

The theory of residual stresses can be based on assumptions analogous to those on which is based the theory of plastic deformations, which are the causes of the existence of these stresses. As in the case of the fundamental assumptions with regard to the mechanical conditions of plastic flow, also for the residual stresses, several cases may be distinguished, according to the magnitude of the deformations during the removal of the load or to the character of the stress-strain relation during unloading. In the following, only two cases will be discussed.[26]

[26] For methods relative to the calculation of internal stresses, there may be mentioned papers of E. Heyn: (Martens-Heyn: Materialkunde, 2d ed., p. 280 (1912). Festschrft Kaiser Wilhelm Ges., p. 121, 1921, and *Stahl und Eisen*, 1917.) G. Masing: (several papers in Veröffentlichungen des Siemenskonzerns, Berlin 1923–1926). Ch. Duguet (loc. cit.), and A. u. L. Föppl "Drang u. Zwang," vol. 2, p. 297, 1920, Munich.

(*a*) *Linear Law for Unloading.* In metals having a definite yield point, the residual stresses after unloading may as a first approximation be obtained by assuming that after plastic deformation and subsequent removal of the load the metal behaves like a perfectly elastic material. If a bar of soft steel, previously stressed somewhat above the yield point in tension, be unloaded and then loaded again in *tension*, it will be observed that as long as the stresses remain below a certain value, it behaves as an elastic body. Thus the elastic modulus will be found of practically the same value as that observed before permanent elongation. Therefore, to obtain in some degree

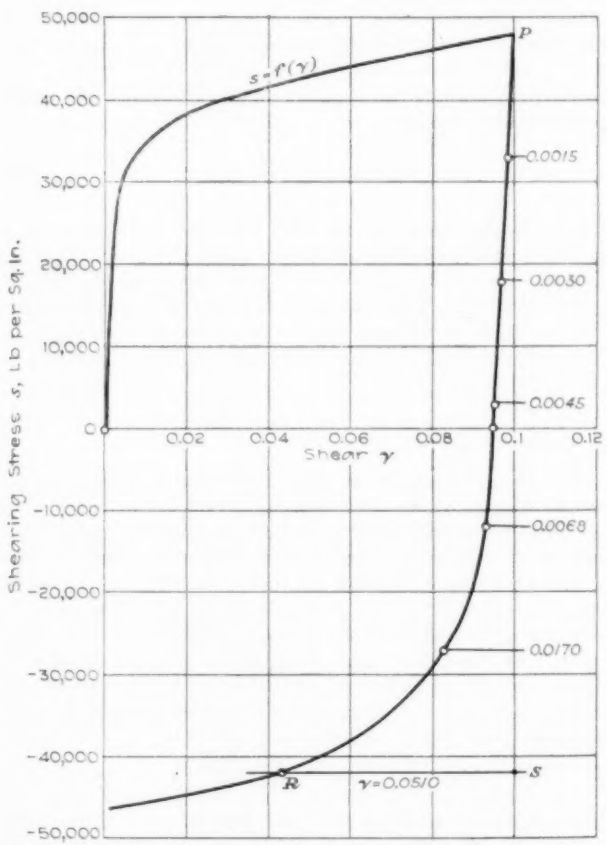


Fig. 22 Stress-Strain Diagram for Pure Shear for Direct and Reversed Sense of Shear

a picture of the residual stresses in soft iron after plastic deformation, it will be necessary only to subtract from the stresses s_x^*, s_y^*, ... occurring when the external forces and moments begin to diminish, a certain ideal system of internal stresses s_x', s_y', ..., assuming purely elastic deformations. The latter are so chosen that after unloading the original values of the resultant forces and external moments will disappear. Hence this second system of stresses s_x', s_y', ... possible in a perfectly elastic body must lead to the same *resultant* forces and resultant moments as the first system, but must balance the action thereof. In using this method, of course, such effects on residual stress as that of "after flow," the after effects of previous stresses, and the phenomenon known as the "Bauschinger effect" (decreasing of the stress at which the stress-strain curves leaves Hooke's straight line when the stresses change sign) are naturally not considered.

(*b*) *General Law for Deformation for Unloading.* From tests with subsequent change of sign in stress, for example, from a

test in torsion with thin-walled tubes with subsequent twisting of the test piece in both directions to the plastic limit, it is known that the shape of the curve for pure shear tends to have a more or less curved portion where the stresses change their sign. The shape of a stress-strain diagram in pure shear containing also the portion of the curve for the decreasing of the torsion moment and reverse twisting is shown in Fig. 22 for mild steel.[27] The shape of the curve for twisting in a reversed sense seems to change with the increasing amounts of plastic deformation (of cold work); the first section of the curves, *PQ* (Fig. 22), however, seems to differ but little from a straight line.

With these considerations it is possible to improve the first method [as mentioned under (*a*)] to find the residual stresses by using now a curved stress-strain diagram also for the unloading (obtained from tests), such as the line *PQR* in Fig. 22. If the shape of the curve *PQR* for the unloading does not change with the amount of cold work preceding the unloading, this curve can be used at all distances r from the center of the bar.

As an example, the case of the twisting and untwisting of a solid, cylindrical bar should be taken. Let the shape of the stress-strain diagram in pure shear for direct twist be represented by the curve *OP* in Fig. 22. With the help of the curve $s = f(\gamma)$ the true distribution of shearing stress s in the cross-section of the bar can be obtained in the manner described under Section VIII. Let this distribution be called s^*. Then s^* are the shearing stresses, acting at the distance r from the center of the bar at the moment when untwisting begins, and γ_a^* is the corresponding unit shear at the extreme fiber $r = a$ of the bar. If now the torsion moment is decreased and the bar begins to untwist, the shearing stresses will begin to decrease. Let the amount of untwist be given by the unit shear γ_a' at the extreme fiber $r = a$. This amount γ_a' may be represented in the diagram (Fig. 22) by the horizontal distance *RS*. To find the distribution of stress in the cross-section of the bar corresponding to a unit shear $\gamma_a^* - \gamma_a'$, we have to subtract at every point in the distance r from s^* a certain stress distribution s'. This latter is obtained by distributing a system of shearing stresses s' as given in the portion *PR* of the curve *PQR* in Fig. 22, over the radii r of the circular cross-section, and by plotting the differences $s^* - s'$ over the radii r. In this manner the different stress curves in Fig. 23 have been obtained for a cylindrical bar of mild steel with a diameter of 2 in. ($a = 1$ in.). These curves show the distribution of shearing stress over the radial distances r for different amounts of untwisting. The latter is indicated by the figures giving the final angle of twist per unit of length, and these are written above the corresponding stress curves. Thus, for example, in Fig. 23 the lowest curve shows the distribution of shearing stress in the bar for a total angle of twist per unit of length $\vartheta = -4.30$ deg. The highest curve shows the original distribution of stress corresponding to an angle of twist of $\vartheta = 5.7$ deg. For intermediate angles of twist several other curves have been constructed in Fig. 23 which show the corresponding stress distributions during the unloading of the bar and the subsequent twisting in the opposite direction. Under these intermediate curves might be drawn one to which a torsion moment equal to zero would correspond. This curve would give the residual stresses which must remain after the complete removal of the torsion moment. (This curve is not shown in Fig. 23.)

In a similar manner the residual stresses in a thick-walled tube after plastic flow could be obtained. If, for example, the stress-strain curve of the material could be expressed by a power function as assumed in Section IX, the radial and the tangential stresses at the moment when the plastic flow is interrupted would be given by the expressions:

[27] Relative to such tests see dissertation of W. Bader (Univ. of Göttingen, 1927).

$$s_r^* = -p\,\frac{br^m}{a^m - b^m}\left(\frac{a^m}{r^m} - 1\right)$$

$$s_t^* = p\,\frac{b^m}{a^m - b^m}\left[(m-1)\,\frac{a^m}{r^m} + 1\right]$$

Here a and b are the outer and inner radii of the tube, and p is the internal pressure. It is assumed that the tube does not change its dimensions in the axial direction.

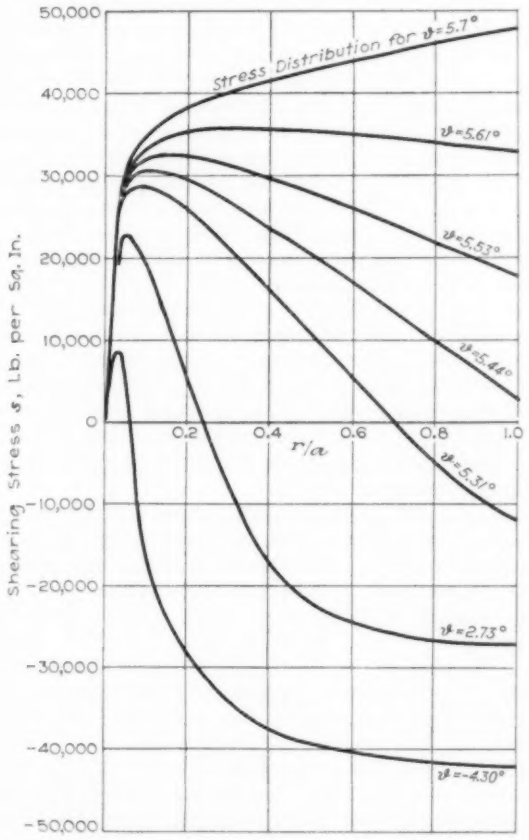


FIG. 23 RESIDUAL STRESSES IN TWISTED CYLINDRICAL BAR FOR DIFFERENT ANGLES OF TWIST

($2a$ = diam. = 2 in.; ϑ = angle of twist per inch; r = distance of point from axis of cylinder.)

Assuming that during the subsequent decreasing of the pressure p a linear stress-strain law would hold, the second stress system (denoted by the prime mark) would be given by

$$s_r' = -p\,\frac{b^2}{a^2 - b^2}\left(\frac{a^2}{r^2} - 1\right)$$

$$s_t' = p\,\frac{b^2}{a^2 - b^2}\left(\frac{a^2}{r^2} + 1\right)$$

Hence the residual stresses acting in the tube after the internal pressure p has been removed will be given by the stress differences

$$s_r = s_r^* - s_r'; \quad s_t = s_t^* - s_t'$$

It should be remembered, however, that these expressions will only hold when at no place within the tube is the limit of plasticity reached during the removal of the pressure a second time.

Discussion

ARTHUR MCCUTCHAN.[28] The paper is a highly interesting excursion into the land where stress is no longer directly proportional to strain and where deformation depends on time and temperature as well as on stress. The paper should help to clarify present rather obscure conceptions of just how one should distribute materials advantageously in equipment operating at temperatures which place the metal in the plastic condition.

It is noted in the analysis of a thick hollow cylinder, on the eighth page and Fig. 13, that the maximum tangential stress occurs at the outer surface of the tube. As this is exactly contrary to the result obtained from elastic consideration,[29] it is pertinent to inquire if the effect of heat flow from the outside to the inside of a tube in the plastic state would tend to reduce the stress rather than augment it as is found from the elastic theory. This effect of heat flow on the stress in a tube wall is an important factor in present-day design of heat-absorption equipment. Hence, if it can be shown that heat flowing through a tube wall from the outside does not increase the stress in the tube wall, one of the restrictions on the design of such apparatus for high-temperature service will be removed.

It is to be hoped that the author will indicate how to coordinate the findings of this paper with available test data on the behavior of materials at high temperature. Stress-strain curves in the plastic region are seldom reported, attention being directed toward creep phenomena. It would appear that the majority of investigators are more concerned with the behavior of metals in the temperature range in which rate of change of shape is a dominating factor.

In designing equipment in which the stress set up by thermal expansion is a determining factor, the lack of a satisfactory equivalent for Young's modulus for materials operating at or near the plastic condition is evident. It is not clear just how such problems can be handled by the methods outlined in this paper.

J. J. KANTER.[30] During recent years, use of metals under severe stresses at elevated temperatures has given impetus to the study of plastic flow under such conditions. As the author has intimated in introducing his theories, tests have shown that steels and other metals at temperatures of 750 deg. fahr. possess stress-strain diagrams of the type shown in his Fig. 2. Furthermore, the creep or continuous deformation under load becomes a considerable factor in many service applications.

While it is not the intention in any manner to deprecate the excellent and scholarly analysis that the paper presents, it appears that the theories advanced could well be extended to embrace creep conditions. From the standpoint of testing materials, plastic flow of metals at temperatures permitting stress-strain diagrams as shown in the author's Fig. 1 is of great importance and may be applied to the interpretation of cold plastic deformation. When creep is a factor, however, as is the case in most high-temperature applications, the variable of time must be taken into account in the stress-elongation and stress-shear functions.

It would appear that in the theories developed by the author where he uses

$$s = f(\epsilon) \quad \ldots\ldots [2]$$

$$s = f(\gamma) \quad \ldots\ldots [114]$$

to represent equilibrium conditions of axial deformation and shear, respectively, inverse experimental functions also containing the variable of time could be applied to include the more general case, as

$$\epsilon = f(s, t)$$

or

$$\gamma = f(s, t)$$

where t is time under stress.

Many technical data on the creep of various materials are being accumulated from a number of laboratories. The author's theories, particularly in the case of tubes under internal pressure, should be of value in applying this information if revised to take account of deformation as a function of both stress and time. Inasmuch as an arbitrary law is resorted to for the tube case, it would seem that the formulation could just as well be altered to include plastic-flow conditions at elevated temperatures.

FRANKLIN L. EVERETT.[31] An investigation of the behavior in the plastic region of ductile materials subjected to twist is being conducted at the University of Michigan. This is in the nature of an academic experimental problem and is related to the character of the analytical work presented in the paper. The validity of the maximum-shear theory is to be studied at elevated temperatures. The work until the present time has been at normal temperatures. Annealed medium-carbon-steel tubing with thin walls is used.

Tests commonly described as "creep" will be made in torsion to produce pure shear, and the results compared with tension tests performed under similar conditions of loading and temperature. A special apparatus has been constructed with the desired features. Angular deformation is accurately measured by projecting beams of light on a distant scale.

The present interest is in discovering a suitable design of specimen to provide a pure shear fracture. Buckling offers some complexities which are being overcome.

Some of the earlier experiments were concerned with obtaining a general knowledge of the characteristics of the material. In several cases specimens were carefully loaded to cause yielding, and allowed to remain in this stressed condition for twelve hours. Upon further loading without having removed the load, an elastic region was again observed, followed by a second yielding. The inference is that during the period of rest there occurred some internal readjustment (possibly an "aging" effect) which restored the elastic properties. The second elastic range was of the order of 4 per cent of the former.

EARLE B. NORRIS.[32] The last few years have seen a great development in the use of cold-worked steel, in which material in sheet or bar form is permanently shaped by the application of stresses beyond the elastic limit of the material. It is known that such cold-working produces an increasing resistance of the material to further deformation. In many cases it is necessary to anneal the material between the various steps of cold-working, in order to restore its ductility.

It is known in a general way that deforming a piece of metal beyond its elastic limit produces a new elastic limit equal to the stress which has been applied and reduces the ductility by the amount of unit deformation which has taken place. And it is known that annealing will restore the metal practically to its initial physical properties. For many applications of cold-working such general information is sufficient. In other cases, however, direct use is made of the altered properties. In the design of a machine part which is to be made by cold-working, one may,

[28] Engineering Division, The Detroit Edison Company, Detroit, Mich.

[29] "Applied Elasticity," Timoshenko and Lessells, p. 252.

[30] Crane Company, Chicago, Ill.

[31] Research Engineer, Department of Engineering Research, University of Michigan, Ann Arbor, Mich. Jun. A.S.M.E.

[32] Dean of Engineering, Virginia Polytechnic Institute, Blacksburg, Va. Mem. A.S.M.E.

and should, make use of the higher elastic limit produced in the process of shaping. Even where the altered physical properties are not used in design, one may be concerned with the amount of energy required to perform the operations of cold-forming against a progressively increasing resistance. In wire mills cold-drawing produces an increase in elastic limit and ultimate strength of the wire, and dependence is placed on these higher physical properties in use of the wire; as, for example, in the cables of suspension bridges. The wire mills have certain rule-of-thumb equations which are used as an indication of the physical properties to be expected by any given amount of wire drawing.

Referring to Fig. 24, it is known that when an ordinary tensile specimen has been stretched to the point A on a stress-strain curve, if the load is removed the recovery will be elastic, and that when the load is again applied the material will be elastic until the point A is again reached, when the original curve of the material will be followed if the load is continued to rupture. There never has been, however, any equation that represented accurately the stress-strain curve, and, due to the irregularities of shape, it is doubtful if any equation could be derived of general application. The French use a parabolic curve starting with the

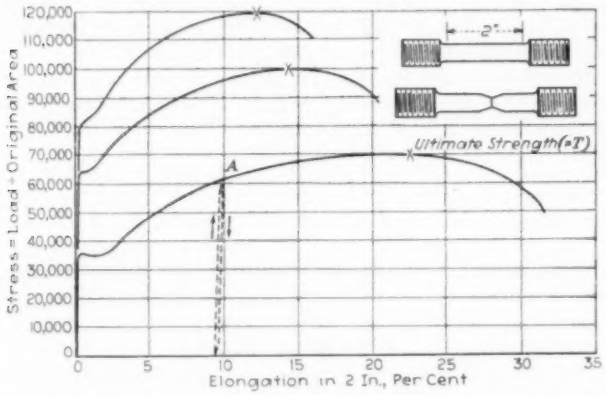


Fig. 24 Conventional Stress-Strain Curves, Elongations in 2 In. as Abscissas, Stresses on Original Area as Ordinates

original elastic limit and increasing the elastic limit according to the square root of the elongation. This is only a rough approximation and certainly could not apply beyond the ultimate strength of the material, where the actual curves begin to curve downward to rupture. Furthermore, it is an open question whether the shape of the curve between the elastic limit and the ultimate strength is a function of the elastic limit or of the ultimate strength, or possibly of both.

It is known that in these ordinary stress-strain curves there are used as abscissas the mean elongation of the specimen in a standard gage length. Actually, one part of the specimen is receiving more elongation than the rest, and is being gradually drawn to a small diameter until rupture finally occurs at this minimum section. Obviously the resistance of the material as shown by the ordinate of the curve is the resistance of this minimum section, while the elongation which is plotted is not the elongation of this section, but is the average for the entire gage length of the specimen. The curves are therefore not a true indication of the relationship of resistance to flow.

In looking for a possible method of plotting physical test data which might give a better indication of a reliable law, the writer used first a practice adopted from the English of plotting "true stress." By "true stress" they mean the load upon the specimen divided by the actual area at the minimum section corresponding to each load. This gives a constantly rising curve instead of one which begins to fall off at the time the ultimate strength is reached. This use of "the true stress" seems more logical, because in the design of any cold-worked article one is concerned with the finished dimensions and not with the original dimensions of the piece from which it is made.

As a measure of true deformation there are a number of terms which might be used. One might plot as abscissas the reduction in area at the minimum section. He might plot the reduction of diameter, which, of course, is merely a function of reduction of area; or he might plot the true elongation, meaning the elonga-

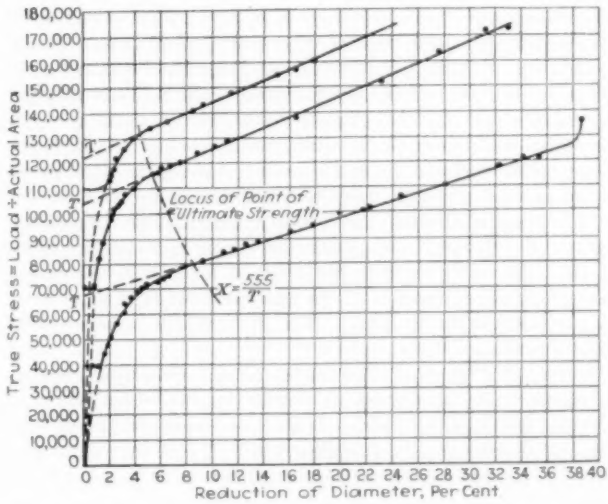


Fig. 25 Curves Obtained by Plotting Reductions of Diameter as Abscissas and True Stresses as Ordinates

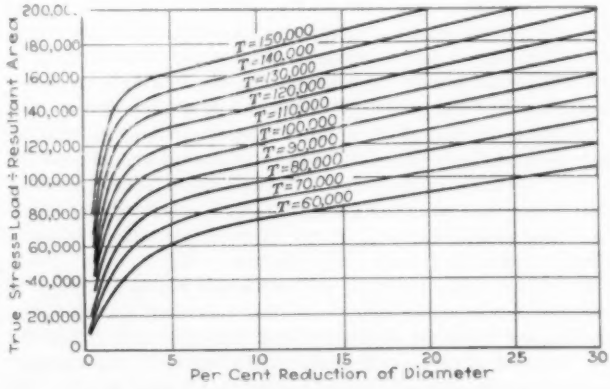


Fig. 26 Stress-Strain Curves From Formula

tion which takes place right at the minimum section and which would represent the elongation of the specimen if it had all been drawn down to the minimum diameter. Whichever one of these is used seems immaterial, because any of them may be computed from any of the others. After trial of these three possibilities for abscissas the writer finds the most promising at present to be to plot reduction of diameter as abscissas and true stress as ordinates. Fig. 25 shows a series of curves obtained in this way from which many interesting deductions may be made. To get these curves from the ordinary tensile test requires only the addition to the equipment of a micrometer with ball points. As each load is observed, the corresponding diameter is determined with the micrometer, from which the per cent reduction of the diameter may be calculated. Each load is divided by the actual area supporting it, as determined by the micrometer readings,

thus giving the ordinates of "true stress," or load per unit of actual area.

Fig. 25 shows such curves for three specimens of radically different physical properties. One is immediately struck by the straight-line characteristic of the greater part of these curves. This discovery was not original with the writer, but so far as he knows was first made by Mr. John Stead, an English metallurgist, and was presented in a paper given before the Iron and Steel Institute in about 1924. Mr. Stead tested 39 steel specimens of widely varying physical properties, and all of them showed this characteristic. He overlooked, however, what seems to be the most significant feature of this method of plotting. If the straight-line section of the curve is prolonged to its intersection with the y-axis, this intersection proves to be the ordinary ultimate tensile strength based on original area. The ultimate strength therefore becomes one of the constants of the equation which can be set up to represent the entire curve. In fact, the equation as the writer has deduced it contains no other constant than the ordinary tensile strength, and therefore indicates that the elastic limit of the material once passed, the elastic limit has no effect on the behavior of the material afterward, but that the plastic flow is entirely a function of the ultimate strength and of some measure of true deformation. This deduction is believed to be great significance.

The point of tangency on these curves corresponds to the point at which the ultimate strength is reached on the ordinary curve of a tensile test, but the ordinate is the true stress at this point instead of the stress on the original area. Up to that point this new curve proves to be logarithmic and beyond that is a straight line.

To deduce an equation representing this combination of a curve with a straight line is not as complicated as it might appear. The straight line, of course, is the ordinary equation $Y = a + mx$, in which a is the y-intercept ($= T$) and m is the slope of the line. From this equation there must be subtracted a third term on the right which has a maximum value equal to a when x is 0, and which vanishes at the point of tangency. This term therefore takes the form $-T^{(1-bx)}$, in which the value of b is such that bx is equal to 1 at the point of tangency. This subtractive term therefore has a value of T when x is 0 and has a value of T^0 or 1 at the point of tangency. In comparison with stresses of several thousand pounds a value of unity is practically infinitesimal. For greater values of x, the value of this term becomes a fraction, less than 1, and also negligible.

The writer finds that the stronger steels reach the point of tangency much earlier than the softer and more ductile steels. The locus of this point appears to be a function of x such that the x at the point of tangency is equal to 555 divided by the ultimate strength of the steel in thousands of pounds. The term b in the writer's equation therefore becomes $0.0018T$.

In examining a great many of these curves, both Mr. Stead's and his own, the writer finds that in general the slope of the straight line increases for the stronger steels. This slope may be approximated as $^2/_{10}$ of the square root of T, where T is the ultimate strength in thousands of pounds. This study thus gives a general equation for the true stress upon a specimen at a reduction of diameter of x per cent:

$$S = T + 0.2\sqrt{T}\,x - T^{(1\,-\,0.0018\,Tx)}$$

As it stands, this equation is clearly an equation made up of two sections, one of which is exponential or logarithmic in general form, and the other of which is a straight line.

In plotting his curves the writer finds that austenitic steels, such as very high-tungsten or nickel steels, give radically different slopes to the line from all other varieties. Probably this is due to a different crystalline structure. It is a question whether alloys of 15 to 30 per cent tungsten or nickel should be classed as steels or as alloys. In any event, their behavior requires different constants from those in the equations proposed. These equations are, however, applicable to the ordinary carbon steels, to nickel steels, and chrome-nickel steels up to 4 per cent nickel and 1.6 per cent chromium, and to a manganese content up to 0.75 per cent. Whether these limits may be extended cannot be said, but in all probability the limits established will cover the full range of steels which are likely to be cold-worked under any methods known at present. The austenitic steels display merely a change of slope of the line and require changing the slope coefficient from $0.2\sqrt{T}$ to some other value. Otherwise the curves are the same.

It will further be noticed that these curves, if prolonged in a logical direction, seem to converge at the intersection of the two axes. This would indicate that there are two laws of behavior of steel, one of which is Hooke's law, which prevails up to the proportional limit. The other law, which the writer will call the "law of plastic flow," apparently also starts at zero load, but is not apparent until Hooke's law has been broken down, when it comes into action and controls the plastic behavior of the metal.

A rather complicated mathematical study of this equation for plastic flow indicates that the curved section is purely logarithmic in form, which is the form that might be rationally expected for a progressive building-up of specific resistance.

If each successive increment of deformation dx produced an increment $K\,S\,dx$ in the specific resistance of the metal, there would be the relation $dS = K\,S\,dx$, whence by integration one would have

$$\log_e S = K\,x + C$$

or

$$S = e^C\,e^{Kx}$$

In comparison with this, the logarithmic part of the equation may be written

$$S = e^C\left(1 - \frac{1}{e^{Kx}}\right)$$

There is an interesting similarity here, but the writer so far has been unable to evolve a rational explanation of the difference between the two forms.

The straight-line section corresponds to a fundamental assumption that $ds/dx = m$, whence $S = a + mx$.

The physical significance of this is that for this part of the flow curve the rate of true stress increment with respect to reduction of diameter is a constant.

In Fig. 26 the equation has been plotted for steels of different tensile strengths. The value of S from these curves, for a steel of any given tensile strength, represents the resistance per unit of final area which the steel will exert at the end of cold working to a given reduction of diameter of x per cent. From the ordinary tensile tests it is known that a specimen which has been stressed to a value S beyond its original yield point becomes elastic up to this same stress S when load is reapplied; in other words, S becomes the new yield point. Consequently, Fig. 26 should indicate the approximate yield point of steel which has been cold-worked. In the case of drawn wire, these values could be used only where the diameter has been reduced without intermediate annealing, since the anneal removes much of the effect of previous wiredrawing.

It would seem, however, that the necking of a specimen after the ultimate strength has been reached has a tendency to produce stresses which are not uniform over the section, and that therefore the simple tensile test cannot be used to determine the effect of cold-working beyond that point. This discussion is therefore offered merely to indicate the direction in which the writer has been working and to indicate the value of and the need

for further research in this field if full use is to be made of the changes in physical properties of metals produced by plastic working.

THOMAS MCLEAN JASPER.[33] This paper is exceedingly timely and is on a subject which is of great interest to all designers of heavy equipment. The writer has been particularly aware in the past two years of the different action of cylinders of thin section as compared to cylinders of thick section. In the thin cylinder, adjustment to shape may occur which allows it to a greater degree to carry the full load represented by the strength of the metal, while in the thick cylinder a slight out-of-roundness is not adjusted for at the yield-point stress of the metal, with the result that the load carried at failure is less than that anticipated. An endeavor to calculate for this, using the stress-strain diagram of the metal, has not answered all of the questions involved. The development of plastic mechanics which can be superimposed upon the elastic mechanics will be of great benefit, and it is anticipated will materially help the designer of thick sections to appreciate the problems involved.

In thick cylinders where there is a slight out-of-roundness, it is believed that the plane section at no stress does not remain plane under stress. The action of forming the cylinder walls by rolling does not allow it to act under stress according to the stress-strain diagram obtained on a tensile- or compression-test specimen, and therefore there are introduced considerable difficulties in the application of strains to thick-walled sections.

In the forming of all thick cylinders, no matter by what method made, the general principle just suggested will apply to a greater or lesser degree. It is not believed that a heating of the cylinder will adjust this difficulty completely. It will, it is believed, materially help the situation.

What applies to the problem of thick cylinders also applies to all heavy sections carrying load.

The value of this paper lies in the attempt to outline a rational system of plastic mechanics. Its need first attracted the attention of the writer when the effects of a hole in ductile steel under fatigue indicated that the stress was really twice the average instead of around three times as indicated by elastic mechanics.

To have a man of Dr. Nadai's qualifications attempt to rationalize this problem means that it will get much farther than if attempted by one of less experience or with a less appreciation of the use of mathematical tools.

G. M. EATON.[34] The writer has comparatively recently come to the clear realization that internal or residual stress is the direct result of plastic flow. If one permits himself to imagine a piece of steel in which residual stress is absolutely non-existent and puts this piece of steel through any heat cycle with the sole limitation that no plastic flow occurs at any stage of this heat cycle, then at the end of the cycle, after the metal has returned to the original temperature, residual stress will still be non-existent.

It is on account of the fact that plastic flow plants the seeds for the future building up of residual stress that one recognizes the vital importance of learning all that he can about plastic flow. The type of plastic-flow phenomena in which the writer is now most vitally interested is that plastic flow which occurs when steel is quenched. The conditions surrounding the passage of the steel through the recalescence point make a problem of rare complication. The major variable is the volumetric change occurring in the steel during this cycle, and the local distress in weak spots of the steel is, among other things, a function of the rate at which this volumetric change takes place. A great deal of work has been done purporting to evaluate this change of volume, but logical criticism can be directed at every attempt which has come to attention to give even a close approximation of this rate. The writer is convinced that the rate of change of volume is more rapid, and he believes far more rapid, than any of the tests referred to would appear to indicate.

It is hoped that eventually the work of the Division in connection with plasticity will shed some light on this difficult subject.

S. TIMOSHENKO.[35] This paper contains not only the original work of Dr. Nadai, but also an introduction describing the general theory of plastic flow. Such an introduction is necessary because the paper is the first work on plastic flow published in this country. Having this in mind, the writer feels that it is desirable to have some portions of the general theory discussed more in detail. For instance, he would like to have a more detailed discussion of the *rule of plastic flow*, especially its physical side, and the experimental evidence supporting the theory.

The problems of plastic flow with rotational symmetry are of more theoretical than practical interest, but they are necessary for illustrating the general theory. It seems to the writer that the discussion of the first example can be simplified. Since shear does not produce change in volume, it can be concluded that S_r and S_t are zero. Hence the problem is reduced to pure shear combined with tension. The problem on plastic flow in bending is of practical importance. Here the author discusses not only the *direct problem*, but also the *inverse problem*, and shows that having the relation between the bending moment and the deflection, it is possible to obtain the strain-stress curve of the material. There are no indications regarding the accuracy of this method, and some experimental development in this direction would be of practical interest.

The theory of plastic flow in a twisted circular shaft is discussed very completely. Solutions are given for both the direct and the inverse problem. The assumption taken as the basis is that, during twist beyond the elastic limit, the radii of cross-sections remain straight; hence the shearing strain is proportional to the distance from the axis of the shaft. This assumption is quite natural, and it would be very important to have it proved by direct tests.

The last portion of the paper contains a discussion of *residual stresses*. This subject is of utmost practical importance. In many practical cases there are residual stresses due to plastic flow. Sometimes this plastic flow is produced intentionally, so as to raise the yield point of the material and to develop residual stresses, which, combined with stresses produced by external loads, give a more favorable stress distribution. It is common practice, for instance, to submit the chains of hoisting machines to a certain amount of overstrain to eliminate undesirable stretching of these parts in service. The cylinders of hydraulic presses are sometimes submitted to an initial internal pressure sufficient to produce a permanent deformation in the walls. The strain hardening and the residual stresses produced by this method prevent any permanent set in service. The overstraining of metal is sometimes used in the manufacture of guns.[36] By stretching the metal in the wall of a gun beyond the initial yield point and afterward submitting it to a mild heat treatment, the elastic properties of the material are improved; at the same time initial stresses are produced, which combined with the stresses produced by the explosion, give a more favorable stress distribution. Tur-

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[34] Commercial Engineer, Molybdenum Corporation of America, Pittsburgh, Pa. Mem. A.S.M.E.

[35] Professor of Mechanical Engineering, University of Michigan, Ann Arbor, Mich. Mem. A.S.M.E.

[36] See paper by S. J. Brown, "A Method of Gun Construction by Radial Expansion," United States Naval Institute Proceedings, vol. 46, p. 1941, 1920.

bine disks and rotors are sometimes given an analogous treatment. By running these parts at *overspeed*, a permanent set is obtained around the central hole, which raises the yield point of the material and produces initial stresses, which are in a favorable direction. This problem has been discussed in detail by A. Nadai and L. H. Donnell.[37] Die-cast aluminum fans are sometimes submitted to overstrain at the bore to prevent any possibility of their loosening on the shaft in service.

In using overstrain in this manner, it is necessary to keep in mind (1) that the hardening disappears if the structure is submitted to annealing temperatures, and (2) that the stretching of metal in a certain direction, while making it stronger with respect to tension in this direction, does not proportionately improve the mechanical properties with respect to compression in the same direction.

Considerable residual stresses are produced in the process of drawing or rolling metals. In drawing a bar through a die, for

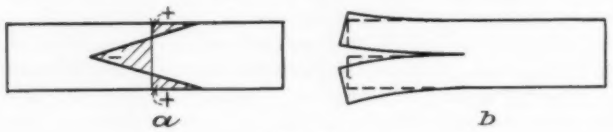


FIG. 27

instance, the metal at the outside is stretched more than the metal at the middle. Hence, drawn bars have considerable residual stresses in tension at the surface and compression at the middle. In drawn copper bars of narrow rectangular cross-section, the distribution of these stresses at a distance from the ends is as shown in Fig. 27-*a*. If the bar is cut lengthwise, there will be bending such as shown in Fig. 27-*b*. Measurement of this bending shows that the maximum residual stresses produced in drawing the copper bars are of the same order as the yield point of the material. These kinds of stresses are of great practical importance. They cause undesirable warping in the process of machining, and to them must be attributed also *season cracking* in various copper alloys which have been cold-worked and not properly annealed afterward.

C. A. NORMAN.[38] It is very much to be appreciated that the author now has developed methods for at least approximately estimating the stress and strain conditions arising in machine parts when the elastic limit has been passed, at least locally.

It has been well known for a long time that such situations might exist, for instance, at the edges of holes in highly stressed parts, at sharp reentrant corners, etc., but just how serious and how widespread these situations were could not be quantitatively stated, except perhaps on the basis of rather complicated tests with polarized light or other roundabout methods.

It is to be hoped that the paper will form the first step toward a great number of mathematical investigations of this character, and particularly it is to be hoped that men carrying out such investigations will be able to use methods as clear and comparatively simple as those introduced by Dr. Nadai.

As for the formation of a committee to coordinate research along these lines, there are perhaps arguments both for and against. It is certain that no serious progress can be made unless certain individuals of outstanding ability are willing to sink themselves more or less wholeheartedly for a while into these problems. Persons capable of such work are often irritated or paralyzed rather than helped by being made wheels of an organization, and they can usually be depended upon to have a fair knowledge of work that has been done in the same field before.

Nevertheless, if a committee can secure financial support for individual workers, give them either leisure or experimental equipment; or if the committee can secure the help of routine workers to produce a bibliography, etc., it may be of considerable assistance to the men on whom real progress must depend. Also, such a committee may of course act as a source of information to persons not very familiar with the field and yet in need of information.

W. TRINKS.[39] This excellent paper refers, it is believed, to the plastic state of metals at infinitely low velocity of deformation. From a statement in the paper it appears that the author is of the opinion that internal friction of the metal plays a part in plastic deformation only at elevated temperatures of the metal and not at room temperature. As far as the writer knows, the viscosity or internal friction does play an important part even at room temperature. The diagram (Fig. 28) was copied from original data furnished by the Ellwood City works of the National Tube Company. In that particular plant it was desired to speed up the cold-drawing of tubes. The drawing force was furnished by hydraulic pressure acting upon a piston. Higher speed required higher pressure in the cylinder. The cylinder area multiplied by the pressure gives the straight line shown in the diagram. It is quite remarkable how well all the data furnished by the National Tube Company plot on a straight line.

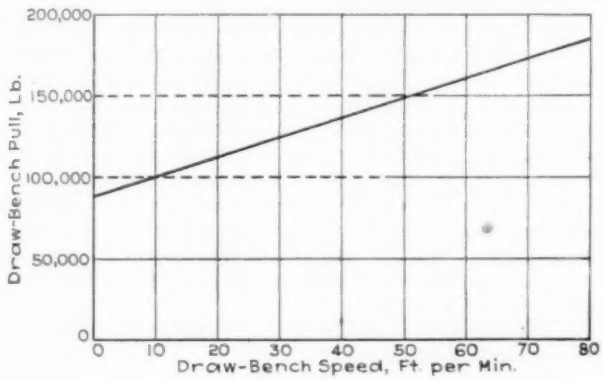


FIG. 28

Since the tubes have a chance to cool down to room temperature, it is quite evident that the increase of resisting force is a function of speed of deformation even at room temperature, because it cannot be assumed that the material would be heated to a red heat in the die. Neither can it be assumed that friction in the operating mechanism grows with the speed, because the friction of the hydraulic cylinder does not vary perceptibly with the speed.

AUTHOR'S CLOSURE

In reviewing the different discussions, the author first wishes to recapitulate briefly one or two points mentioned in the introductory part of his paper which were perhaps not worked out with sufficient clearness. It appears from all that is known about the laws under which plastic deformations in ductile metals in general start to develop, that these deformations seem not to follow as general rules, as do, for example, the elastic deformations of isotropic and perfectly elastic bodies. While it seems necessary—and is inevitable if any mechanical methods of com-

[37] Trans. A.S.M.E., Applied Mechanics Division, 1928.

[38] Professor of Machine Design, Mechanical Engineering Department, Ohio State University, Columbus, Ohio. Mem. A.S.M.E.

[39] Professor of Mechanical Engineering, Carnegie Institute of Technology, Pittsburgh, Pa. Mem. A.S.M.E.

parative simplicity shall be applied to analyze distributions of stress or of strain in the plastic state—to restrict the considerations to some few assumptions and to describe plastic flow under certain idealized conditions, it should be remembered that the results of computations should not be generalized beyond the limitations under which they were obtained. The author endeavored, however, to show in some few examples that the methods appear to be flexible enough to allow more general assumptions to be considered also in cases of plastic flow in which more complicated stress-strain relations may be assumed or to include other experimentally known facts or effects, such as work-hardening, elasticity, a non-linear stress-strain relation for loading different from that for unloading, or other observable facts.

In the paper, cases were considered where the entire body was assumed to be in the plastic state. In many cases in the practical applications, however, it is important to study how plastic flow gradually spreads through a given body under an increasing load. If the material has a well-defined yield point, in such cases in general an elastic portion of the stressed body can be distinguished from a plastic part. In the first, the stresses do not reach the limit of plasticity and all the strains remain purely elastic. In the second, the stresses will satisfy the condition of plasticity.

As an example, the case of a thick-walled tube under internal pressure and without axial extension may be mentioned, consisting of a material having a well-defined yield point. Using the same notation as before for the thick-walled cylinder, (see Figs. 11 and 12) there will be a plastic part $b \leqq r \leqq c$, and an elastic part $c \leqq r \leqq a$ in the tube. In the first, the stresses can be expressed by formulas similar to Equation [65]. Taking in account that for $b < r < c$ the condition of equilibrium is:

$$\frac{d(rs_r)}{dr} - s_t = 0$$

and the condition of plasticity:

$$s_t - s_r = \frac{2s_0}{\sqrt{3}} = \text{const}$$

the stresses s_r and s_t are already given, whence

$$s_r = -\frac{2s_0}{\sqrt{3}} \log_e \frac{c}{r}$$

$$s_t = \frac{2s_0}{\sqrt{3}}\left(1 - \log_e \frac{c}{r}\right)$$

$$s_z = \frac{s_0}{\sqrt{3}}\left(1 - 2\log_e \frac{c}{r}\right)$$

where c is a constant of integration.

In the elastic portion $c < r < a$ of the tube, the stresses satisfy the conditions of an elastic material and are given by the well-known formulas

$$s_r' = c_1 + \frac{c_2}{r_2}$$

$$s_t' = c_1 - \frac{c_2}{r^2}$$

$$s_z' = \nu(s_r + s_t) = 2\nu c_1$$

where ν is Poisson's ratio. To determine the three constants c, c_1, and c_2 we have the conditions:

$$r = a_1,\ s_r' = 0$$

$$r = c_1,\ s_r' = s_r$$

$$r = c_1,\ s_t' = s_t$$

The pressure p under which the plastic part of the tube will reach to a radius $r = c$ is given by

$$p = \frac{2s_0}{\sqrt{3}}\left[\log_e \frac{c}{b} + \frac{1}{2}\left(1 - \frac{c^2}{a^2}\right)\right]$$

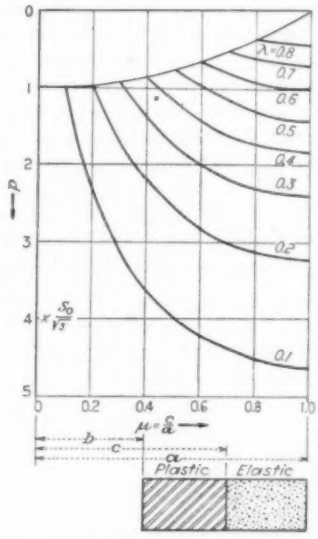


FIG. 29 PRESSURE P PRODUCING YIELDING PLOTTED AGAINST RATIO $\mu = c/\alpha$
(c = radius of surface to which plastic region extends.)

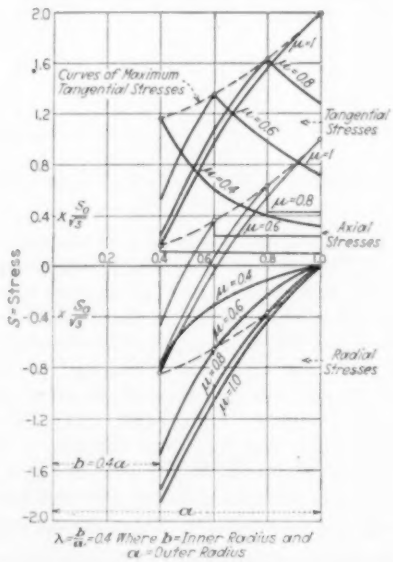


FIG. 30 CURVES SHOWING STRESSES IN THICK-WALLED TUBES

where s_0 is the yield stress in pure tension or compression, and a and b are the outer and inner radius of the tube. Denoting by

$$\rho = \frac{r}{a}, \qquad \lambda = \frac{b}{a}, \qquad \mu = \frac{c}{a}$$

in Fig. 29, the pressures p for a thick-walled tube are plotted against the ratio $\mu = c/\alpha$, thus indicating how the pressure p will increase in a tube which yields to a distance $r = c$ in the radial direction. In Fig. 29, ten curves are shown, each for a given

value of $\lambda = b/a$ of the ratio of inner to outer radius of the tube.[40]

An example showing the stress distribution is given in Fig. 28 for a tube with $\lambda = b/a = 0.4$ and $\mu = c/a = 0.4, 0.6, 0.8,$ and 1.0.

It is perhaps instructive to compare now the shape of the curves showing the distribution of the radial, tangential, and axial stress in a thick-walled tube subjected to high internal pressure p and yielding partially or completely under the assumption of *an idealized stress-strain diagram*, such as Fig. 1 and as used in the preceding formulas or in Figs. 29 and 30 with results obtained by *assuming a general curve for stress-strain diagram* such as Fig. 2. Such a comparison is given in that which follows, by using a stress-strain diagram which was actually obtained in a tensile test with a die-cast aluminum. The stress-strain curve for this die-cast aluminum is shown in Fig. 31 for a unit elongation not exceeding $\epsilon = 0.02$ or 2 per cent. It may be remarked that this diagram shows a case where the material does not exhibit a yield point: the stresses increase quite gradually and there

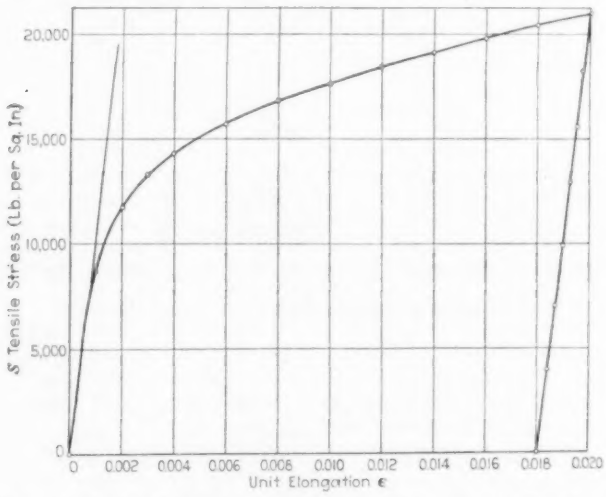


FIG. 31 STRESS-STRAIN CURVE IN PURE TENSION FOR DIE-CAST ALUMINUM

is no "elastic range" which could be first distinguished from a "plastic region." To obtain from the tensile diagram the "stress-strain curve for pure shear" which is necessary to compute the stresses according to what was said in the paper under section IX, it was assumed that the elastic parts of the strains could be first neglected if compared with their permanent or plastic parts. Under these assumptions, the stress-strain curve *in pure shear* for the die-cast aluminum $s = f(\gamma)$ (Eq. [114]) can be obtained in first approximation from the tensile curve $s_0 = \varphi(\epsilon)$ as given by Fig. 31 by multiplying all the abscissas of Fig. 31 by $^3/_2$ and by plotting against the thus obtained new abscissas γ the new ordinates s which are equal to $s_0/\sqrt{3}$ at the corresponding points. By means of this transformation: $\gamma = 3\epsilon/2$ and $s = s_0/\sqrt{3}$, the shear diagram Fig. 32 was constructed. In this figure, an "idealized diagram," with thin lines also, consisting of two straight lines was plotted and the stresses were computed according to what was said above and according to section IX, by using these two stress-strain curves. The tube was assumed to have a ratio of inner to outer radius equal to $\lambda = b/a = 0.4$. In Fig. 33, the distribution of the radial, tangential, and axial stresses is shown as obtained by using the actual curved stress-strain diagram as well as the idealized straight-line diagram. A comparison of the two sets of stress curves shows that a greater difference between the actual and the ideal stresses can only be observed at the tangential stresses s_t. The tangential stresses s_t in the outermost parts of the tube are smaller according to the actual stress-strain curve than those derived by the ideal diagram. The values of the two other stresses s_r and s_z are, however, not considerably different.

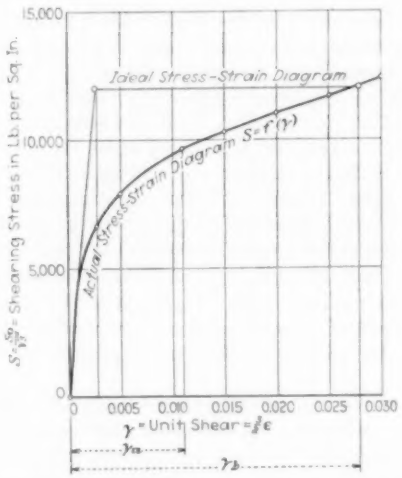


FIG. 32 STRESS-STRAIN DIAGRAM IN PURE SHEAR FOR DIE-CAST ALUMINUM

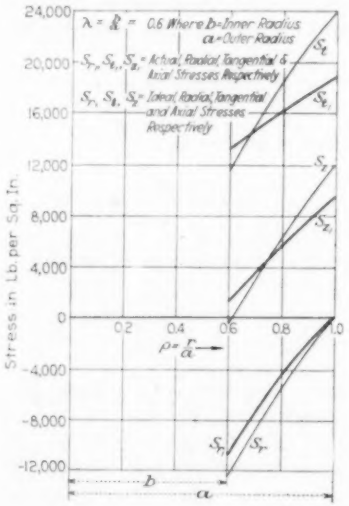


FIG. 33 COMPARISON OF ACTUAL STRESS WITH IDEAL STRESS IN A THICK-WALLED TUBE OF DIE-CAST ALUMINUM

Regarding the possible extension of these methods to the case of creep, that is, where the plastic deformation of metals at elevated temperature must be considered, the author feels that such an extension of the methods would be of practical value. In this paper, however, he restricted himself first to the simplest case of a slow plastic flow, such as occurs during a tensile, compression, or torsion test with a ductile metal and at a comparatively low temperature, and what was perhaps not clearly enough said, also assuming comparatively *small velocities* of deformation. If the temperature is raised to temperatures where recrystallization or creep begins, the laws become different and it will be necessary to introduce in addition to the one independent variable which was the unit extension in a tensile test, a second independent variable, namely, the rate of flow or the unit elongation

[40] The author is much indebted to Mr. H. Friedman for having constructed the diagrams shown in Figs. 29–32, including those for the die-cast aluminum tube.

per unit of time. Thus the velocity of deformation has to be considered, which was not necessary at slow deformations and under ordinary temperatures.

In this connection, the remarks of Professor Trinks might be answered by referring to some old results of tests made by P. Ludwik with tin wires and by Cassebaum with mild steel. The latter showed that a metal such as mild steel even at room temperature exhibits an effect of the velocity of deformation on the stress-strain diagram. It is a well-known fact to steel producers that the stress-strain diagram can be altered slightly by the speed with which, for example, a tensile-test bar is stretched while it yields. The higher the speed, the higher the tensile-stress curve proceeds. But it was shown by both Ludwik and Cassebaum that the law appears to be a logarithmic one, i.e., that if the ordinates of the stress-strain diagram are increased by equally small amounts, the corresponding velocities must be increased by much more and they follow in first approximation a geometric progression. This means that under low velocities applied in ordinary testing, stress may practically be considered as independent o velocity of deformation and the whole effect can be considered as a secondary one. Under the high velocities of an impact test, during quick cold rolling, or the drawing manipulation of a metal, the velocities are of a different order of magnitude and their effect of speed might become appreciable, as appears to be the case in Professor Trinks' tests. This effect also has been observed, for example, in impact tests.

Referring to the remarks of Professor Norris, the author thinks that here again a case of plastic flow is mentioned, which appears to be more complicated than the cases to which he wished to call attention. In the paper, only small deformations were considered, while the diagrams to which Professor Norris refers include comparatively large deformations, which in general involve a more complicated analysis. The diagrams (Figs. 25 and 26) extend to 30 and 40 per cent reduction of diameter, respectively. Denoting by l the length and by a the diameter of a tensile bar at the instant it carries a stress s and by the subscript zero the initial values of these lengths, the unit elongation ϵ and the reduction of diameter designated hereafter are given by

$$\epsilon = \frac{l - l_0}{l_0}, \qquad \delta = \frac{a_0 - a}{a_0}$$

Considering now that the volume of the bar remains practically unchanged until necking starts, we have

$$\pi a^2 l = \pi a_0^2 l_0$$

which equation permits us to express ϵ by means of the variable δ:

$$1 + \epsilon = \frac{1}{(1 - \delta)^2}$$

and this relation might be used as a first approximation also to express the mean axial extension in the smallest cross-section of the neck of a tensile-test piece. From this it is seen that an axial unit elongation ϵ corresponds to a reduction of diameter of 30 per cent or $\delta = 0.3$, which is nearly equal to unity or $\epsilon = 100$ per cent.

The stress-strain curve of a tensile test ordinarily expressed by

$$s = f(\epsilon)$$

may, according to Professor Norris, be transformed into a function

$$s = F(\delta)$$

using δ as the independent variable, where δ and ϵ satisfy the equation

$$(1 + \epsilon)(1 - \delta)^2 = 1$$

In the former form $s = f(\epsilon)$, the point on the stress-strain curve which corresponds to "the ultimate strength" (denoted hereafter by S_0 if based on the original cross-section A_0) is found in taking the load P or

$$P = A_0 s_0 = A \frac{s}{1 + \epsilon}$$

a maximum, where A designates the area of cross-section of the bar. The condition that P shall become a maximum leads here to the equivalent condition that

$$\frac{ds}{d\epsilon} = \frac{s}{1 + \epsilon}$$

which equation suggests a very simple geometrical construction of the tangent of the tensile stress-strain curve $s = f(\epsilon)$ at the point where the stress s_0 computed by the original cross-section A_0 is just equal to the ultimate strength S_0. This contruction, according to Rejtö, consists of drawing the tangent to the true stress-strain diagram $s = f(\epsilon)$ from a point situated at a distance = unity from the origin of the $s_1\epsilon$-axis system on the negative ϵ-axis. If one wishes, the same condition can be expressed by using the variable δ of Professor Norris as

$$\frac{ds}{d\delta} = \frac{2s}{1 - \delta}$$

and by means of this last equation it is now possible to compute the length of the ordinate on the stress axis s comprised between the origin and the intersection of the tangent to the stress-strain curve $s = F(\delta)$ at the point which corresponds to the ultimate strength S_0 on it. This length, which should, according to Professor Norris, be equal to the ultimate strength S_0, is found equal to

$$\frac{1 - 3\delta'}{(1 - \delta')^3} \cdot S_0$$

where δ' designates the reduction of diameter at the moment the necking starts. From the last formula it is seen that the rule Professor Norris mentions is not exactly verified. Only when δ' is a small quantity (compared to unity) does the fraction which precedes the term S_0 in the last equation equal unity.

Regarding the formula proposed by Professor Norris to express the curve $s = F(\delta)$ by an analytic law, the author would prefer an expression for the stress-strain curve which would not depend on the dimensions of the quantities entering in it, and which would not contain two distinct analytical laws in the two respective intervals of the independent variable quoted. There is no physical reason apparent which would make it necessary to assume that in the point where the ultimate strength is reached in a tensile bar, a new law of deformation comes into action. All that actually occurs is that a certain resultant of the stresses taken over the cross-section of the bar, namely, the tensile load P, becomes an analytical maximum. A consequence of this fact is that the deformation ceases to spread uniformly over the length of the bar and is further localized, this causing the well-known effect of necking, and finally rupture.

With respect to the question of residual stresses, which has been referred to by Professor Timoshenko as one of practical interest and which has only been touched in the paper, the author feels that much more has to be done to discuss the conditions under which these stresses are formed after partial plastic flow or in connection with temperature stresses as in the case of quenched-steel specimens mentioned by Mr. Eaton, where also an essential part of the question seems to be the sudden change in volume occurring during quenching. Referring to the question of Mr.

McCutchan, it seems to the author that the cases of a plastic flow in a thick-walled tube might be treated first independently of the question of temperature stresses, and that perhaps temperature stresses might be superimposed later on the first distribution, if plastic flow had ceased to spread and provided that by the addition of the new stresses the plastic limit was not reached again.

The author agrees with Professor Timoshenko that it would have been possible to treat the case of the pure combination of an axial tensile stress with a twisting couple by assuming at once that the stresses s_t and s_r are zero. It was, however, the intention to derive all the simpler cases, which are included in the case of rotational symmetry, as special cases from the more general case treated in the paper. The question whether these cases have more theoretical than practical interest seems to the author not so clear, because it could be conceived perhaps that one might analyze with similar methods the equilibrium of plastic bodies pressed between rigid plates or during such manipulations as the production of a seamless tube (combined tension and twist) or in other cases of pressing hollow or solid cylinders. It is true that in these cases of practical interest the conditions have probably been more complicated than could be treated until now, and as was already pointed out during the discussion by Messrs. McCutchan, Kanter, and Trinks, but the author is nevertheless of the opinion that in attempting to learn more regarding the distributions of stress in the plastic state, it will be perhaps also useful to discuss first some of the simpler cases under restricting or idealizing assumptions.

APM-52-18

Further Research on Helical Springs of Round and Square Wire

By A. M. WAHL,[1] EAST PITTSBURGH, PA.

A method of calculating the stress in helical springs of square or rectangular wire is proposed. The method is to use an approximate formula, based on St. Venant's results for torsion of rectangular wire, and multiply by a correction factor based on the spring index. The method is quantitatively checked by strain measurements both on semi-coils of actual square-wire springs and on complete springs loaded in compression. It is suggested that this new formula replace those commonly given in handbooks, which may lead to considerable error in certain cases.

Deflection measurements between coils of helical round-wire tension springs made from the same bar but having spring indices varying from 2.7 to 9.5 indicate that the ordinary helical-spring formula for round wire gives results quite close to the actual deflection.

A simple approximate formula, based on St. Venant's results, for deflection of rectangular-bar springs is proposed to replace those ordinarily used and commonly given in handbooks. These ordinary deflection formulas may also lead to considerable error in certain cases.

THE fact that helical springs of round wire may be stressed much higher than ordinary spring tables or formulas indicate has been shown both analytically and experimentally in a previous paper.[2] This paper showed that where the spring index—i.e., the ratio of the mean coil diameter to the wire diameter—was 3, the maximum stress was approximately 60 per cent greater than that computed by the use of ordinary spring formulas or tables. Furthermore, it was shown that the stress at the inside of the coil at *a* (Fig. 1) was around $2^1/_2$ times that on the outside of the coil at *b*.

In the discussion of this paper it was asked whether or not this effect would exist in springs made from square wire, which because of the coiling operation assume the trapezoidal form shown in Fig. 2. Because of the increase in width at the inside edge at *a* (Fig. 2), it was suggested that the stress might even be reduced. The experiments described in this paper, however, do not indicate any such reduction, but on the other hand indicate quite clearly that the same thing happens in the case of square wire as in the case of round wire, i.e., there is a stress at the inside of the coil about 2 or $2^1/_2$ times greater than that at the outside of the coil and around 40 to 60 per cent greater than that calculated by St. Venant's torsion theory.

In this discussion it was also suggested that the deflections of helical springs having small indices might deviate somewhat from the results calculated by the ordinary spring-deflection formulas or tables. The experiments described in this paper indicate that this deviation is quite small and of the order of a few per cent only, the experimental results checking those calculated by the ordinary deflection formula with a considerable degree of accuracy.

I—THE STRESS IN HELICAL SPRINGS OF SQUARE OR RECTANGULAR WIRE

In order to calculate the stress in helical springs of square wire, it is proposed to use the following method. From St. Venant's results for the torsion of a square bar,[3] the maximum shearing stress is given by

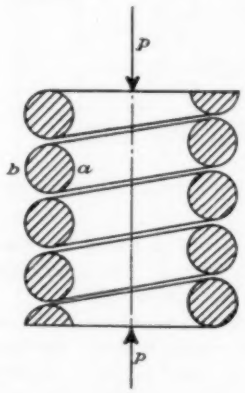


FIG. 1 HELICAL SPRING OF ROUND WIRE, AXIALLY LOADED

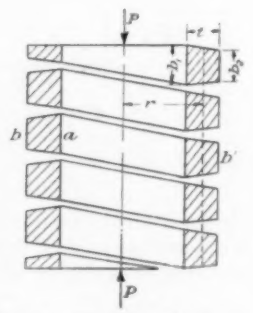


FIG. 2 HELICAL SPRING OF SQUARE WIRE, AXIALLY LOADED

(Note that the wire becomes somewhat trapezoidal due to the coiling operation.)

$$S = \frac{M}{0.208\, b^3} \qquad [1]$$

where S = maximum shearing stress, lb. per sq. in.
b = side of square cross-section, in.
M = applied torsional moment, in-lb.

For a helical spring of fairly small index, the fibers along the inside of the coil are shorter than those along the outside; this fact causes an increase in stress on the inside. To take into account this increase, it is proposed to use the same stress multiplication factor K as that given in the author's previous paper[2] (see Equation [2] of that paper), namely,

$$K = \frac{4c-1}{4c-4} + \frac{0.615}{c} \qquad [2]$$

where c = spring index. For a spring of square wire we take

[1] Research Engineer, Westinghouse Electric & Manufacturing Co. Assoc-Mem. A.S.M.E. Mr. Wahl was born at Churdan, Iowa, in 1901. He attended Grinnell College and Iowa State College in Iowa, receiving the degree of Bachelor of Science in Mechanical Engineering from the latter in 1925. Two years later he received his M.S. degree from the University of Pittsburgh. In 1925 he entered the graduate student course of the Westinghouse Company, and in 1926 he was transferred to the Research Department, where he has been located since.

[2] "Stresses in Heavy, Closely Coiled Helical Springs," A. M. Wahl. Trans. A.S.M.E. (1929), Paper No. APM-51-17.

Contributed by the A.S.M.E. Special Research Committee on Mechanical Springs and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] St. Venant's results for square and rectangular bars are given in "A History of the Theory of Elasticity," by Todhunter and Pearson, vol. II, part 1. See also "Applied Elasticity," by Timoshenko and Lessells; and Edgerton, Trans. A.S.M.E., vol. 47, p. 717.

$c = 2r/t$ (see Fig. 2). The value of K may be obtained graphically from the curve of Fig. 3.

To get the true maximum stress in a spring of square wire, it is therefore proposed to use Equation [1] and multiply by K as taken from Fig. 3, thus obtaining

$$S = K \frac{Pr}{0.208\, b^3} \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

where P = axial load on spring, lb. (Fig. 2), and
r = mean radius of coil, in.

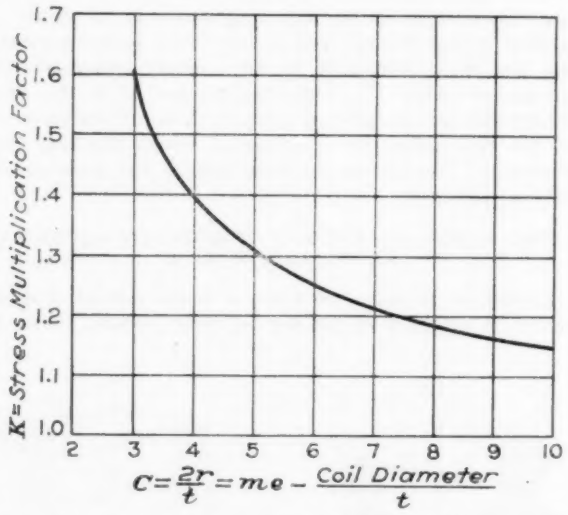


FIG. 3 CURVE FOR DETERMINING THE STRESS MULTIPLICATION FACTOR K

In determining b, an approximation will be had if b is taken as the average side of the trapezoid (Fig. 2). In determining the spring index c, however, the ratio $2r/t$ is to be taken.

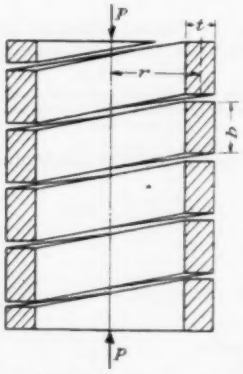


FIG. 4 HELICAL SPRING OF RECTANGULAR WIRE, AXIALLY LOADED

In calculating the stress on the outside of the coil we multiply the value of stress as given by [1] by a factor K', where K' is given by Equation [3] of the above-mentioned paper:

$$K' = \frac{4c + 1}{4c + 4} - \frac{0.615}{c} \quad \ldots [4]$$

We thus obtain the stress on the outside of the coil $S_{min.}$ equal to

$$S_{min.} = K' \frac{Pr}{0.208\, b^3} \quad \ldots [5]$$

where K' is given by [4].

Tests made as described below indicate that this method of calculation will give fairly accurate results.

STRESS CALCULATION IN HELICAL SPRINGS OF RECTANGULAR WIRE, AXIALLY LOADED

For calculating the stress in a helical spring of rectangular wire, axially loaded, and having the longer side of the rectangle parallel to the spring axis (Fig. 4), the following formula is proposed:[4]

$$S = K \frac{Pr(3b + 1.8t)}{b^2 t^2} \quad \ldots [6]$$

where S = maximum shearing stress, lb. per sq. in.
b = length of side of rectangular cross-section (Fig. 4)
t = width of rectangular cross-section, in.
r = mean radius of coil, in.
K = stress multiplication factor (to be computed from Equation [2] or taken from the curve of Fig. 3).

The approximate formula as given by Equation [6] will, if K is unity, give the stress in a bar of rectangular cross-section acted on[5] by a twisting moment Pr. The results calculated by this formula for $K = 1$ agree with those calculated by St. Venant's more exact theory within about 4 per cent in all cases from $b/t = 1$ to $b/t = \infty$. Since authorities on the theory of elasticity generally agree that St. Venant's torsion theory is correct, it

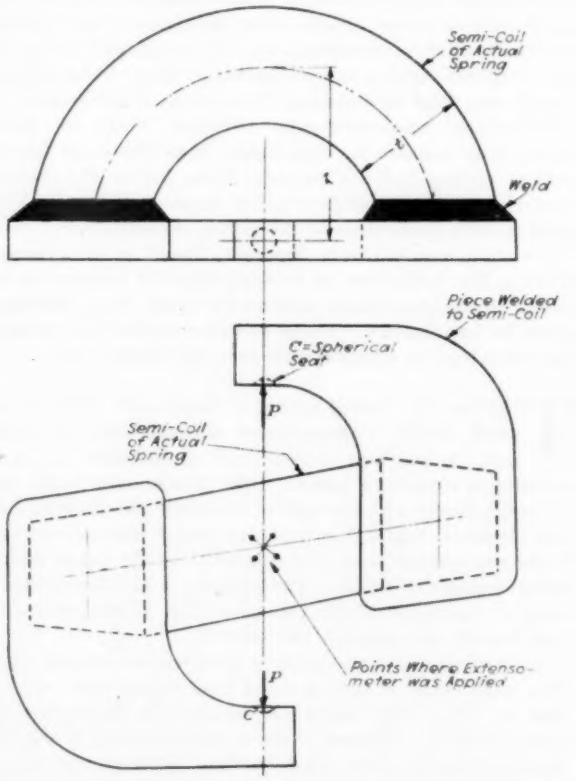


FIG. 5 SQUARE-WIRE SEMI-COIL, TEST ARRANGEMENT

seems to the author that any formula for rectangular-bar springs should be based on this theory. Some of the approximate formulas given in handbooks for springs of rectangular wire may deviate considerably from St. Venant's results when the ratio b/t is 2 or more. It appears that some simple formula such as Equation [6], which is approximately correct throughout the whole range from $b/t = 1$ to $b/t = \infty$, should be used in spring tables and handbooks rather than the formulas which are ordinarily given. These latter are in error for two reasons: first, because they do not agree with St. Venant's results, and, second, because they take no account of the effect of change in fiber length as the spring index is decreased.

Where $b < t$ in Fig. 4, i.e., where the longer side of the rectangle is perpendicular to the spring axis, the author does not believe that the above method of calculation will be accurate. In the author's opinion, further research will probably be necessary on this point.

[4] It should be noted that this formula holds only where the longer side of the rectangle is parallel to the spring axis as in Fig. 4.

[5] See "Applied Elasticity," p. 33.

Experiments to Check Proposed Method of Calculation

In order to confirm the above-described method of calculation, strain measurements were made on helical springs of square

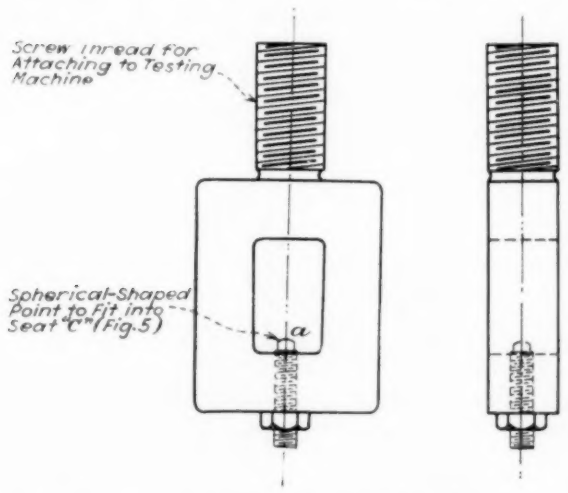


Fig. 6 Eyebolt for Applying Load to Semi-Coil of Spring

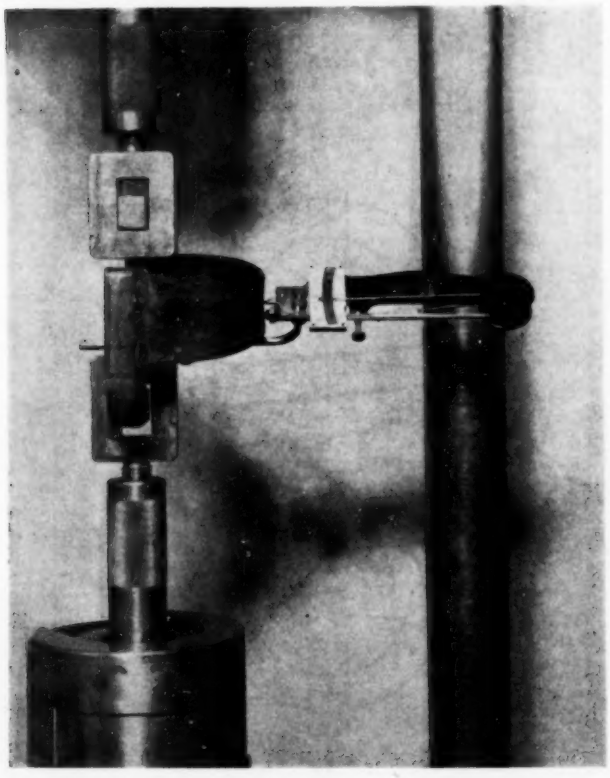


Fig. 7 Test With Extensometer on Outside of Coil

wire which, because of the forming operation, became somewhat trapezoidal in cross-section (see Fig. 2). The method of doing this was exactly the same as that described in the author's previous paper,[2] except that in this case all springs tested were made of square wire. In brief, the method was as follows: Semi-coils were cut from actual helical springs made of ordinary carbon spring steel, heat treated. Two arms were welded on to each semi-coil to form the arrangement shown in Fig. 5. The points of a special extensometer were placed as shown in Fig. 5 so that the strains at 45 deg. to the axis of the spring could be measured. The load was applied to the spherical seat by means of a testing machine, using the eyebolt shown in Fig. 6. By means of this arrangement it was possible to measure the maximum stress at the inside of the coil while the spring was subjected to an axial load as in the complete spring. A photograph of the test arrangement in the testing machine with the extensometer on the outside of the coil is shown in Fig. 7, while in Fig. 8 is

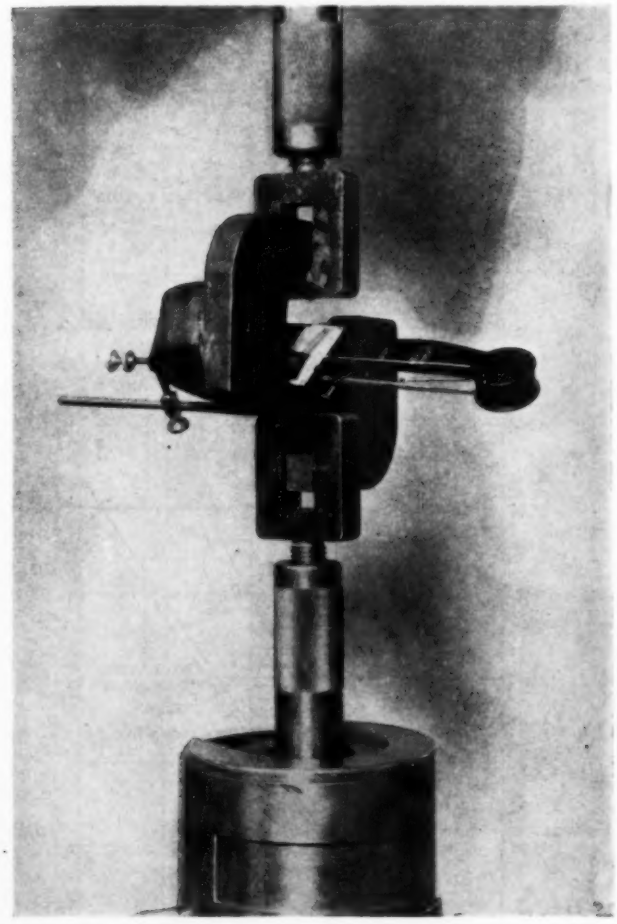


Fig. 8 Test With Extensometer on Inside of Coil

shown the test arrangement with the extensometer on the inside of the coil. The method of loading the semi-coil and of measuring the stress are clearly shown in the photographs.

The method of determining the stresses from the measured strains is fully described in the paper mentioned above[2] and will not be discussed here, since it merely makes use of the fundamental equations of elasticity. In working up the test results, a modulus of elasticity in tension $E = 30 \times 10^6$ lb. per sq.in. and a value of Poisson's ratio equal to 0.3 were used. The equivalent value of the shear modulus is 11.5×10^6 lb. per sq. in. Tests on the spring material (about 1 per cent carbon steel) indicate values very close to these.

Two semi-coils were made and tested, one having an index $c = 2r/t = 3.07$ and the other an index $c = 4.14$. The test results for semi-coil A are shown by the full lines on Fig. 9, one full line representing the stress on the inside of the coil, and

the other that on the outside of the coil. The broken line midway between the two test curves gives the result calculated by St. Venant's torsion formula (Equation [1]). The broken line representing the stress on the outside of the coil was calculated from Equation [5] while that representing the stress on the inside of the coil was calculated from Equation [3], the method of

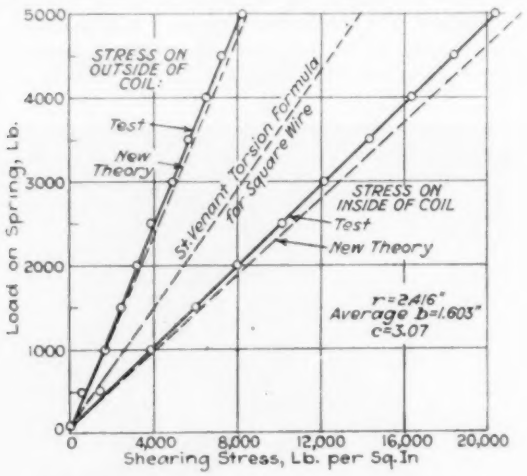


FIG. 9 LOAD-STRESS CURVES FOR SEMI-COIL A

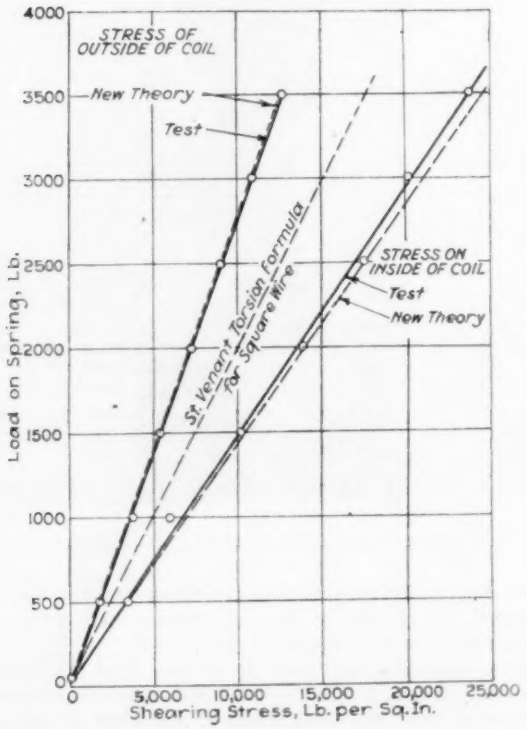


FIG. 10 LOAD-STRESS CURVES FOR SEMI-COIL B

calculation as outlined above being used. It may be seen that while the new method gives fairly accurate results, St. Venant's torsion theory is considerably in error in such cases. The stress on the inside of the coil is seen to be around $2^1/_2$ times that on the outside of the coil, and around 50 per cent greater than that figured by St. Venant's theory. These general results are about the same as those previously obtained by the author for springs of circular wire.

The test results on semi-coil B which had an index of 4.14 are shown in Fig. 10. The method of calculating the theoretical curves is exactly the same as that used for semi-coil A. In this case, however, since the spring index was not as small as was the case for semi-coil A, the stress on the inside of the coil was only about 1.9 times that on the outside and about 35 per cent greater

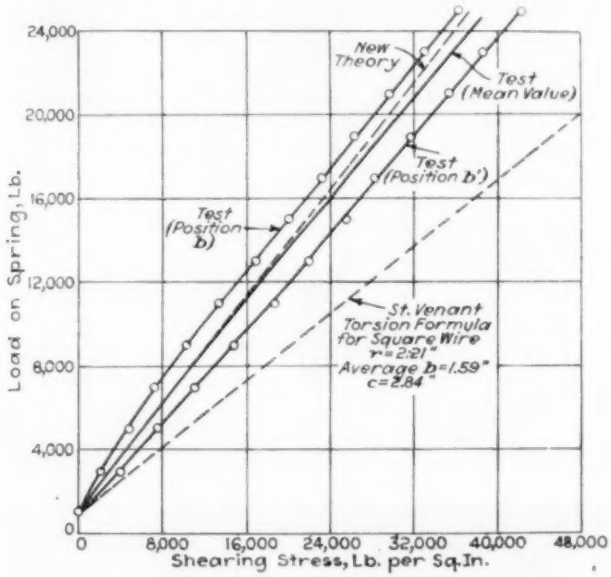


FIG. 11 LOAD-STRESS CURVES FOR COMPLETE SPRING

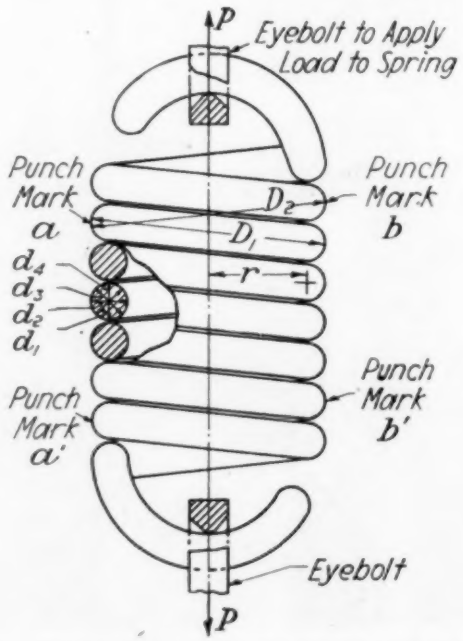


FIG. 12 METHOD OF TESTING SPRING FOR DEFLECTION

than that figured from St. Venant's theory. It is seen that, in this case, very good agreement with the new method of calculation is obtained.

In order to show that these test results were representative of complete springs axially loaded, a test was made on a helical spring of square wire loaded in compression as shown in Fig. 2. Stresses were measured by applying the extensometer at positions

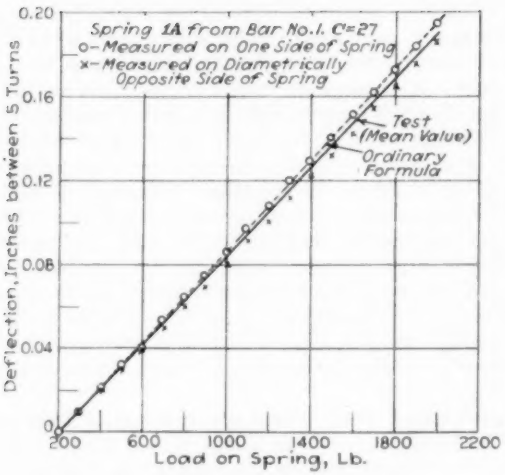


FIG. 13 DEFLECTION TEST OF SPRING 1 A

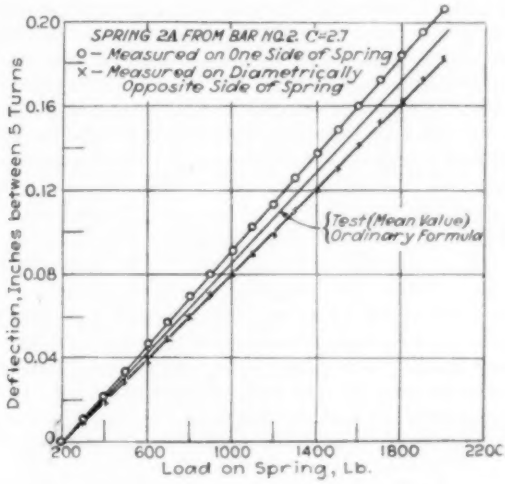


FIG. 16 DEFLECTION TEST OF SPRING 2 A

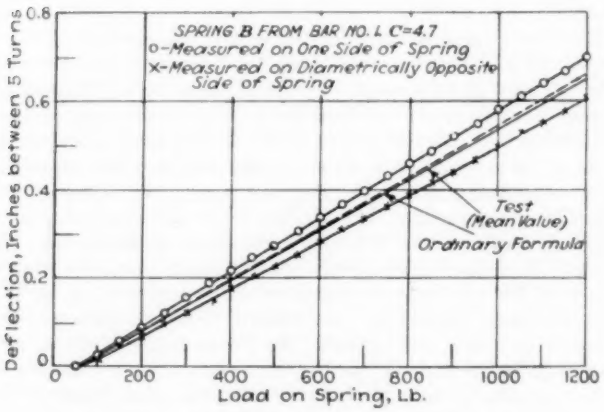


FIG. 14 DEFLECTION TEST OF SPRING 1 B

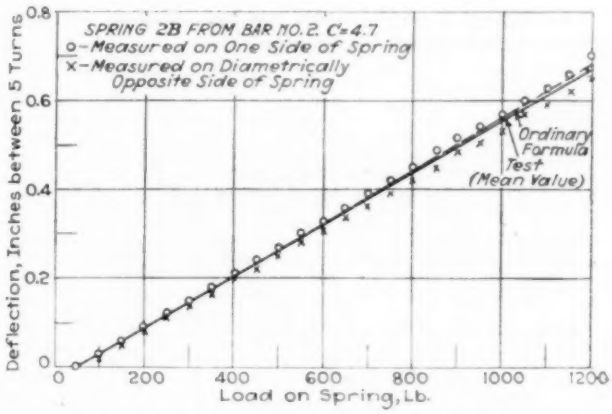


FIG. 17 DEFLECTION TEST OF SPRING 2 B

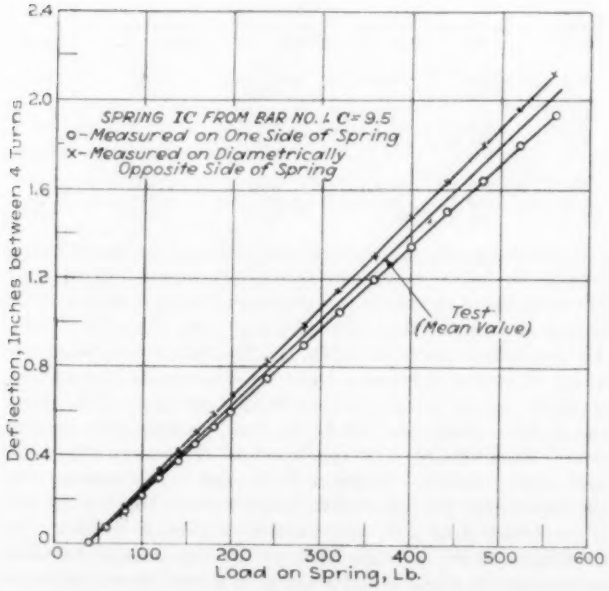


FIG. 15 DEFLECTION TEST OF SPRING 1 C

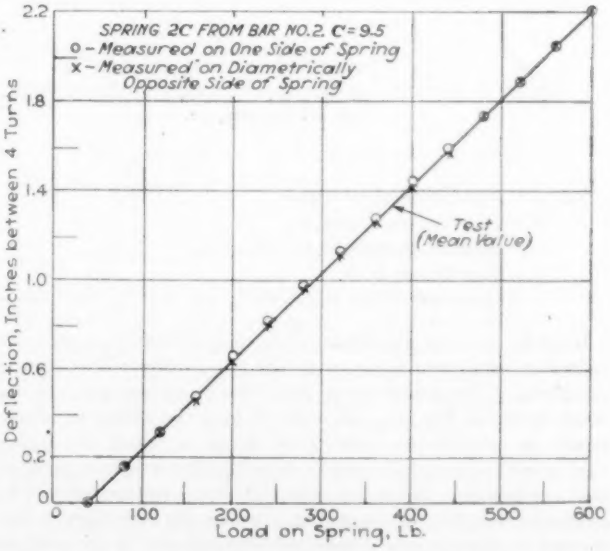


FIG. 18 DEFLECTION TEST OF SPRING 2 C

b and b' on diametrically opposite points of the spring on the outside of the coil. This was done to eliminate the effect of eccentricity of load. The results of this test are shown in Fig. 11. It will be noticed that the stress at position b' (Fig. 2) is somewhat greater than that at position b because of the eccentricity of the load. To get the stress due only to the axial load we take a mean value as shown by the full line midway between the experimental curves. The theoretical curve, computed from Equation [5], is shown as a broken line. This curve agrees very well with the mean experimental value. For comparison, the curve computed by St. Venant's theory (Equation [1]) is also given. Moreover, the mean experimental curve of Fig. 11 agrees very well with that representing the stress on the outside of semi-coil A (Fig. 9), which had about the same index. This indicates that the semi-coil test arrangement does simulate the axial loading of helical spring. Therefore the results indicate that the above-described method of calculation gives sufficiently accurate results for practical work.

II—DEFLECTION OF HELICAL SPRINGS WHEN AXIALLY LOADED

Experiments to Check Ordinary Deflection Formula for Round-Wire Springs

The commonly used formula for the deflection of helical springs of circular wire, axially loaded, is:

$$\delta = \frac{64\ Pr^3n}{Gd^4} \qquad [7]$$

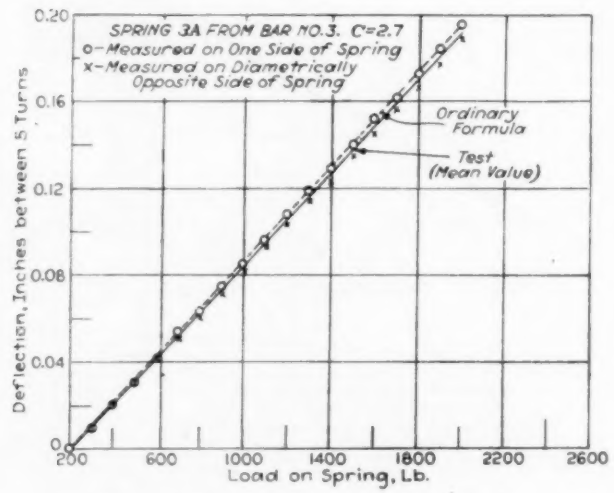


Fig. 19 Deflection Test of Spring 3 A

where n = number of active turns
P = axial load on spring, lb.
G = modulus of rigidity, lb. per sq. in.
d = wire diameter, in.
r = mean coil radius, in.

In order to check up the accuracy of this formula, a series of deflection tests were made on actual round-wire helical springs as follows. The springs tested were tension springs made to the shape shown in Fig. 12. In order to load the spring as nearly axially as possible the eyebolts by means of which the spring was loaded had rounded edges as shown, giving a definite point of load application. Since the object of this investigation was to determine the effect of the spring index on the deflection, it was decided to measure only deflections between coils in the body of the spring in order to eliminate the disturbing effect of the end coils. Therefore deflections were measured between the punch marks a and a' and b and b' which were a given whole number of turns apart as shown on Fig. 12. The reason for putting these marks on diametrically opposite sides of the spring was to eliminate the effect of the almost unavoidable eccentricity of the load. For the smallest springs ($2^1/_4$ in. O.D.) a cathetometer was used to measure the deflections, which were rather small; for the larger springs having larger deflections, a steel scale was used with dividers, this giving sufficient accuracy. Since an eccentric load may be resolved into an axial load plus a pure couple, and since the pure couple does not produce axial extension of the spring but only an increase in length on one side of the spring and a corresponding decrease in length on the other, the deflection due only to the axial load will therefore be obtained by taking the average deflection on diametrically opposite sides of the spring.

A total of nine springs were made from three different bars of $^5/_8$-in. diameter wire by the Union Spring & Manufacturing Company. The material for these springs was ordinary carbon spring steel of about 1 per cent carbon, heat treated. The bars were labeled No. 1, No. 2, and No. 3. From each bar, one spring of $2^1/_4$ in. O. D., one of about $3^1/_2$ in. O. D., and one of about $6^1/_2$ in. O. D., were made, giving a total of 9 springs. The reason for making springs of widely different indices from each bar was that presumably the modulus of rigidity would be the same in all springs cut from the same bar. The springs were then tested for deflection in the manner described above. After the test, the dimensions of the springs were measured to determine accurately the mean coil diameter and wire diameter for computation purposes. For each spring the diameters D_1 and D_2 were measured with a micrometer caliper at two points on each coil between the punch marks as shown in Fig. 12. Then the spring was cut along a plane through its axis by means of a cutting torch and the diameters of the wire d_1, d_2, d_3, and d_4 in the axial, transverse, and oblique directions measured as shown on each coil between the punch marks. Using the average values of d_1, d_2, d_3, and d_4 and the average value of d for the coils between the punch marks, the mean diameter $2r$ could be determined. The average dimensions for each spring, together with other information, are given in Table 1.

TABLE 1 DIMENSIONS OF SPRINGS TESTED

Spring no.	Cut from no.	Nominal O.D., in.	Measured avg. wire diam., in.	Measured avg. mean coil diam., in.	Approximate index
1 A	Bar 1	$2^1/_4$	0.6192	1.657	2.7
1 B	Bar 1	$3^1/_2$	0.6201	2.906	4.7
1 C	Bar 1	$6^1/_2$	0.6208	5.923	9.5
2 A	Bar 2	$2^1/_4$	0.6190	1.654	2.7
2 B	Bar 2	$3^1/_2$	0.6160	2.902	4.7
2 C	Bar 2	$6^1/_2$	0.6202	5.931	9.5
3 A	Bar 3	$2^1/_4$	0.6200	1.664	2.7
3 B	Bar 3	$3^1/_2$	0.6179	2.910	4.7
3 C	Bar 3	$6^1/_2$	0.6207	5.900	9.5

In working up the results the ordinary spring-deflection formula for helical round-wire springs (Equation [7]) was used. The modulus of rigidity G was determined from the tests on the springs of largest index from each bar using Equation [7] and the dimensions given in Table 1. The reason for taking the spring of largest index as a basis for determining the modulus of rigidity is that in this case the effects of curvature of the spring and of direct shear load will be the least; consequently we may expect the ordinary deflection formula (Equation [7]) to be most nearly correct. Values of G obtained in this manner from the slope of the mean load-deflection curves of Figs. 15, 18, and 21 on springs 1 C, 2 C, and 3 C are as given in Table 2. As mentioned before, it is necessary to use the average deflection between points a and a' and b and b' as a basis, to eliminate the effect of eccentricity of load.

TABLE 2 VALUES OF MODULUS OF RIGIDITY FOR BARS NOS. 1, 2, AND 3

Bar No.	1	2	3
G, lb. per sq. in.	11.45×10^6	11.46×10^6	11.50×10^6

This table indicates that the modulus of rigidity was practically the same for all three bars, which fact indicates a uniform material.

The test results on the springs of smaller index are given on the curves of Figs. 13, 14, 16, 17, 19, and 20. In these curves the ordinates represent deflections measured between the punch marks a, a' and b, b' while the abscissas represent load on the spring. The circles represent experimental points obtained on one side of the spring, and the crosses experimental points obtained on a diametrically opposite side of the spring. In most of the springs a small but definite eccentricity of the load is evident. Where possible, full lines were drawn through these points to represent deflection on opposite sides of the springs; in all cases a full line representing the average deflection between the two sides was drawn. The broken lines marked "Ordinary Formula" represent the load-deflection curve as calculated from the ordinary helical-spring deflection formula (Equation [7]). In this computation, a modulus G was used for each spring corresponding to the bar from which the spring was made (Table 2), while the dimensions given in Table 1 were also used. It may be seen that the theoretical curves as represented by the broken lines in all cases come very close to the mean experimental curve, in some cases actually coinciding with it. This indicates that the ordinary spring deflection formula gives results quite close to the truth.

Discussion of Results

For the present let us assume that the ordinary helical-spring formula is correct for determining the deflection due only to torsion of the wire section, and let us add to this deflection the deflection due to direct shear caused by the axial load. Assuming the shearing stress due to the direct axial load as uniformly distributed over the cross-section, we obtain the shearing deflection due only to the direct axial load:

$$\delta_s = \frac{\text{shearing stress} \times \text{length of wire}}{G}$$

$$\delta_s = \frac{P}{(\pi/4)d^2} \times \frac{2\pi nr}{G} = \frac{8\,Pnr}{Gd^2} \quad \ldots\ldots\ldots\ldots [8]$$

Actually the shearing stress is not uniform, and the value given by [8] must therefore be multiplied by a factor usually slightly greater than unity; however, this factor may be taken equal to unity for our present purpose.

Adding δ_s as given by [8] to δ as given by [7] we obtain an approximate solution which takes into account the additional deflection due to direct shear.

This may be written as the ordinary deflection formula [7] multiplied by a factor K_f, thus:

$$\delta_s = K_f \frac{64\,Pr^3n}{Gd^4} \quad \ldots\ldots\ldots\ldots\ldots [10]$$

where

$$K_f = 1 + \frac{0.5}{c^2} \quad \ldots\ldots\ldots\ldots\ldots [10a]$$

It may be seen that for most practical springs having indices greater than 3, K_f differs but little from unity. Thus for $c = 3$, $Kf = 1.055$, i.e., according to this formula the deflection is about $5^1/_2$ per cent greater than that figured by the ordinary formula.

Since the smallest springs tested had indices of about 2.7, the factor K_f from Equation [10a] would be about 1.07, i.e., the deflections as measured should have been about 7 per cent greater than those calculated by the ordinary formula on the basis of this equation. In the case of the largest springs 1 C, 2 C, and 3 C having indices of 9.5, the error as given by [10a] was only about $^1/_2$ per cent, and hence it seems justifiable to use the values obtained from these springs for use in determining G.

J. K. Wood, in his discussion of the author's previous paper,[6] has also derived an expression for K_f, namely,

$$K_f = \frac{2c^2 + c - 1}{2c^2} \quad \ldots\ldots\ldots\ldots\ldots [11]$$

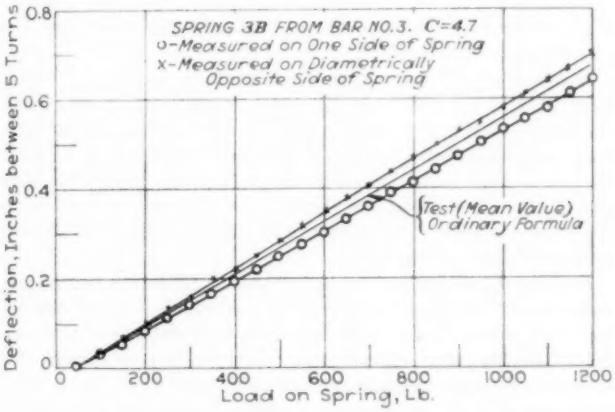


FIG. 20 DEFLECTION TEST OF SPRING 3 B

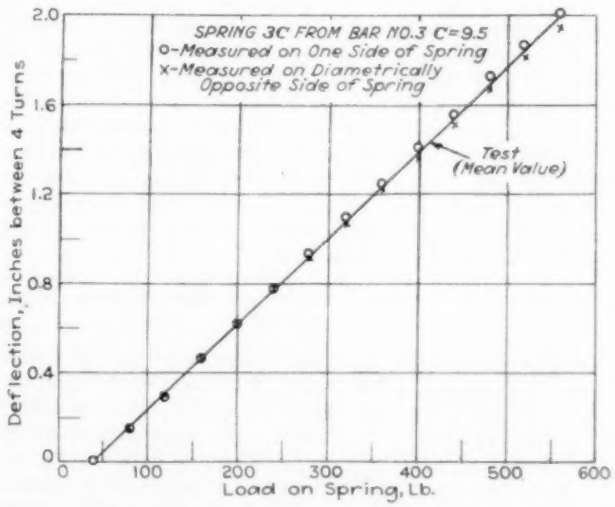


FIG. 21 DEFLECTION TEST OF SPRING 3 C

For an index of 9.5, this gives $K_f = 1.047$ and for an index of 2.7 it gives $K_f = 1.116$. Since the author assumed $K_f = 1$ in determining the modulus G for the largest springs having an index of 9.5, the deflection as measured on the smallest springs having indices 2.7 should on the basis of this formula be about $1.116/1.047 = 1.068$ times the value computed by the author's method. This gives a value of the discrepancy about the same as that estimated by the author using Equation [10a], i.e., about 7 per cent.

As may be seen by referring to the test curves (Figs. 13 to 21),

[6] Trans. A.S.M.E. (1929), APM-51-17, p. 197.

in no case was the test deflection greater than that computed from the ordinary formula, and much less was it 7 per cent greater. In some cases the mean test and the theoretical curve determined from the ordinary formula coincide; in others the mean test deflection is actually slightly less than that computed by the ordinary formula, the difference in no case being more than about 3 per cent. A cumulative error of 1 per cent in measuring the wire diameter, combined with one of 1 per cent in measuring the coil diameter, might account for a 7 per cent difference between test and theory, but it is difficult to see how such a large error could be made in the measurements using a micrometer reading to thousandths of an inch. Furthermore, it is difficult to see how such an error could be made on the three springs of small index simultaneously. The testing machine was calibrated by means of dead weights after the test and was found to be within 1 per cent in accuracy at all loads over one quarter the scale deflection.

It is not certain that the material of the spring made from a given bar of material and having a small index would have the same modulus of rigidity as that of the spring having a large index and made from the same bar. However, in view of the uniformity of the modulus obtained from the three bars (see Table 2), it seems very probable that springs made from the same bar should have practically the same modulus, as was assumed in the calculations.

Another possibility is the error arising from measuring the deflections. For the small springs, the accuracy of the cathetometer was within about 0.001 in. for a total deflection of about 0.2 in., or less than 0.5 per cent. For the largest springs the error in steel-scale measurement was around 0.01 in. for a total deflection of 2 in., or an error of about $^1/_2$ per cent. None of these errors appear to be large enough to make a 7 per cent difference in results. For the present, it may be said that the tests checked the ordinary deflection formula within 3 per cent in all cases, while they did not show an additional deflection as predicted from the above formulas (Equations [10] and [11]), but if anything, a slightly less deflection. It is therefore the author's opinion that the ordinary deflection formula will give results sufficiently accurate for most practical purposes in calculating the deflections in the body of the spring, provided the actual dimensions of the spring and the modulus of rigidity of the material are both accurately known. The effect of the end coils is, however, not considered here.

Deflection of Helical Springs of Rectangular Wire, Axially Loaded

Various formulas are given in handbooks for the calculation of the deflection of helical springs of rectangular wire, axially loaded. None of these formulas give results which agree closely with those obtained by St. Venant, while some of them are in error by as much as 30 per cent for certain ratios of b/t (Fig. 4). For these reasons the author recommends the following approximate formula[7] which will give results agreeing within 4 per cent with St. Venant's for all cases from the ratio $b/t = 1$ to $b/c = \infty$. The formula is accurate within about 1 per cent in the practical range from $b/t = 1$ to $b/t = 3$. The formula is

$$\delta = \frac{19.6\ Pr^3n}{Gt^3(b - 0.56t)} \qquad [12]$$

where P = axial load on spring, lb.
r = mean coil radius, in.
G = modulus of rigidity, lb. per sq. in.
t = thickness of rectangular section (Fig. 4)
b = width of rectangular section (Fig. 4)
n = number of coils.

[7] This formula only applies where the longer side of the rectangle is parallel to the spring axis as in Fig. 4. Where $b < t$, an approximation will be had by interchanging b and t in Equation [12].

Conclusions

The strain measurements which have been described indicate that the stress in square- or rectangular-wire springs may be calculated with sufficient accuracy by the method proposed in this paper. In this connection, the author believes that the formulas for rectangular wire commonly given in handbooks should be replaced by more exact formulas, both for stress and deflection, such as those given in this paper.

The deflection tests on helical springs of round wire, having different indices but made from the same bar of material, indicate that the ordinary helical-spring deflection formula is very close to the truth.

Acknowledgments

For assisting in the work covered by this paper, acknowledgment should be made to C. E. Crang and K. R. Waugh of the Research Laboratories for carrying out the deflection tests on the round-wire springs, to C. McGregor for making most of the measurements of the spring dimensions, to T. C. Rathbone of the Turbine Engineering Department for suggesting the research on square-wire springs, and to Drs. A. Nadai and S. Timoshenko for various suggestions. Credit is also due the Union Spring & Manufacturing Company for furnishing the springs used in the deflection tests.

Discussion

W. M. Austin.[8] The first part of the author's present paper is a companion to his previous paper describing experiments with springs of round wire, and we should expect similar conclusions. His experiments that show a slightly less deflection than that given by the formulas agree with an analytical investigation made by the writer on round-wire springs which showed a correction factor of 0.974 for a spring with an index 3 and of 0.994 for an index of 6.

The writer's analysis of the deflection eliminates the direct shear entirely from the problem, by considering the moment of the load about the torsional axis as resisted by equal moment (not a couple), due to the stresses in the wire, in the opposite direction.

The usual formula for deflection for round-wire springs assumes that the wire is twisted about the center of the section and stressed uniformly in every direction from this center. It also assumes that the lever arms of the load is the mean radius of the spring. Neither of these assumptions is true. The center about which the wire is twisted is distant from the axis of the spring approximately $r + {}^1/_4a^2/r$ where r is the mean radius of the helix and a the radius of the wire. This has the effect of increasing the effective diameter of the spring by $^1/_2a^2/r$. This would tend to make the deflection larger than that given by the formula. It is more than offset, however, by the increased torsional moment of resistance, for a given deflection, about the eccentric torsional axis of the wire.

One would expect, therefore, that with wires of prismatic or composite shape similar conditions would be found; although analytical proof is much more difficult.

In the writer's analytical determination of the deflection he found the correction factor for an index of 9.5 to be between 0.997 and 0.998, so if we neglect errors in observation the author's determination of G from the spring of largest diameter should be correct to $^1/_4$ per cent.

[8] Engineer, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

The writer believes that the modulus of rigidity G should be considered as a property of the material of the spring and independent of the relative spring dimensions. He also thinks that the proper way to ascertain G for helical springs is to take torque and angular-deflection measurements on relatively long straight cylindrical bars that have the same heat treatments that the springs are to receive.

The method of fastening the arms to the semi-coil shown in the paper and also in the author's previous paper on round-wire springs is easy and convenient. The writer remembers that in the previous paper it was stated that the stresses were not carried anywhere near the elastic limit for fear of the weld's yielding. Would not it have been possible for a good blacksmith to have forged the ends from the same piece of steel as the semi-coil? If this could have been done, extensometer readings nearly twice as great as those taken could have been secured and the degree of accuracy increased.

As pointed out by Edgerton in the paper mentioned by the author, relatively few helical springs of square or rectangular wire are made as compared to round wire. One reason is that springs of square or rectangular wire are not as economical in the use of material or space as are springs of round wire. The primary function of a spring is to store energy, and more energy can be stored in the annular cylinder occupied by a spring of round wire than by a square-wire spring of the same material occupying the same space when solid and subjected to the same maximum stress.

To show this, let us calculate two springs, one made from round wire of diameter d and mean radius r, and the other from square wire of short diameter d and mean radius r.

For round wire

$$P = \frac{\pi d^3 S}{16rK} \text{ and } F = \frac{64Pr^2}{Gd^4} = \frac{4\pi r^2 S}{GdK} \text{ and the energy stored per turn} = \frac{1}{2}PF = \frac{\pi^2 S^2 r d^3}{8GK^2} = 1.234\frac{S^2 r d^3}{GK^2} \text{ in inch-pounds.}$$

For square wire

$$P^1 = \frac{0.208\, d^3 S}{Kr} \text{ and } F^1 = \frac{44.5\, Pr^2}{Gd^4} = \frac{9.256\, r^2 S}{GdK} \text{ and the energy stored per turn equals } \frac{1}{2}PF = \frac{0.996\, rd^3 S}{GK^2} \text{ in inch-pounds. The ratio of energies} = \frac{1.234}{0.996} = 1.24.$$

The ratio of weights of material is $\frac{1.24}{0.7854} = 1.57$.

The above figures hold for springs having a square-sectioned wire in the finished spring. If the wire is square before winding the length occupied by one turn will be greater than for round wire. This will cause the space factor 1.24 to be increased a little for springs of large index and much more for springs of small index.

The space ratio of 1.24 and the material-weight ratio of 1.57 in favor of the round-wire spring hold only for springs that are free from stress when unloaded. If, as is often the case, the springs are surged to a lower free height after tempering, these ratios may be somewhat reduced.

If we compare the energies that can be stored by round- and square-wire springs according to the annular cylindrical spaces occupied by the springs in their free state, the ratio 1.24 in favor of the round-wire spring for the solid loaded condition will be found to be much reduced for the free condition and may even be changed to a ratio in favor of the square-wire spring if the springs have a very large index.

One argument in favor of the use of helical compression springs of square wire is that they are somewhat less liable to buckle than springs of round wire having the same outside and inside dimensions. Another is that they can be compressed solid and thereafter act as a column with much less danger of damaging the springs than if the springs were made of round wire. In order to fully realize this last advantage it is necessary in springs of small index to wind them from trapezoidal-shaped wire having dimensions such that the wire is square after the spring is wound.

Successful attempts have been made to make springs having some of the characteristics of both round- and square-wire springs. A notable case is the recoil springs in the 14-in. U. S. Naval gun carriage. These springs are wound from a cylindrical wire that is flattened on opposite sides, the elements of these flat sides becoming perpendicular to the axis of the finished spring (see Fig. 22). In accordance with St. Venant's theory, there

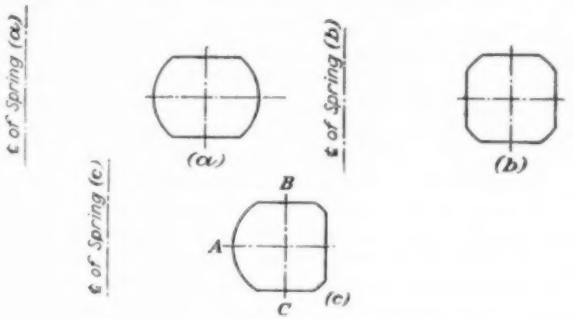


FIG. 22

is a stress maximum in the middle of the flat side. But if the proper amount of flattening is done the stress in the middle of the flat side can be made the same as the stress on the inside diameter of the spring. It is evident that economy in space occupied over round-wire springs is secured by this design.

The writer once had the idea of making a spring the wire of which was to be of circular section except to be flattened on four sides, or it might be described as a square wire with the corners cut off by circular arcs having the same center as the square—see Fig. 22 (*b*). He now believes that a spring made from a wire of circular section flattened on three sides and wound with the unflattened side on the inside of the spring is preferable. It appears that the greatest economy would be had if the wire were flattened more on the outside of the spring than on the sides at right angles to the outside. This follows from the fact that in the flat-sided straight wire the greatest stress is found in the middle of the widest flat side, and from the further condition that in the wound spring having the four sides flattened equally the greatest stress is in the middle of the inside flat. There is a somewhat lower stress in the middle of the radial flat side and a lower stress yet in the middle of the outside flat. Thus by proper proportioning the stresses at *A*, *B*, and *C*, Fig. 22 (*c*), can be made the same.

A little greater economy might still be had if the curvature on the inside of the springs were made elliptical with the focus of the ellipse in the axis of torsion and the directrix of the ellipse were the axis of the spring.

One serious objection to the use of these composite sections is that we have no accurate method of calculating the stress. Unquestionably there is tensile stress at the corners and a maximum shear at or near the middle of the flat sides, which causes a plane section when the spring is free to be curved when the spring is loaded. It is very probable that even in springs of round wire a plane section when the spring is free is not a plane when the spring is loaded, and that the distortion of the plane is greater

the smaller the index of the spring. The distortion of the plane section in the case of circular-wire springs is very small compared to that in the case of flat-sided wires, and, the writer believes, has only a very small influence on the stress.

W. G. Brombacher.[9] Although the formulas of Röver were published in 1913, until the author's work their significance has been but little appreciated by the practical spring designer. The author has made a distinct contribution in calling attention to the fact that the ordinarily accepted and used formulas for the maximum stresses in helical springs are in many cases seriously in error. He has brought this out vividly in his two papers, in which each case has been carefully considered both experimentally and theoretically.

In the past the error in using the commonly given formulas for the maximum fiber stress of helical springs has been compensated by the experience of the spring maker. The breaking load of the springs of various sizes was determined by tests. These test data were used to determine the maximum stress in terms of the common formula, which finally gave rise to tables of permissible stresses for springs of typical design. The author's results now permit a better coordination of the data available on the strength of spring materials, and the performance which may be expected of springs of such materials.

The paper shows that the formulas commonly given for the deflection of a helical spring in tension or compression are independent of the spring index in all cases with but small error. It should be pointed out that the experimental verification was carried out on springs of only one diameter of wire, and further, that the modulus of rigidity was determined by means of data obtained on the helical springs. It is common experience that the value of the modulus of rigidity must be modified when used in designing helical springs by an amount which depends on the diameter of the spring wire and perhaps on the spring index, although the author's data apparently indicate that the value does not depend on this index. It would appear worth while to extend the experimental data to include some on springs of two or three additional wire diameters in order to eliminate definitely all doubts as to the value of the modulus. The discrepancy in the modulus of rigidity of the material and of that used in the deflection formula of helical springs is real and its theoretical explanation is still lacking. This further experimental work may lead well to a deflection formula which is on a logical basis.

The policy of the Westinghouse Electric and Manufacturing Company in releasing the author's paper should be commended. It is to be hoped that other manufacturers who are carrying on research on springs and spring materials will follow their example.

T. McLean Jasper.[10] The question of evaluating spring formulas is a difficult one. The writer would like to see the tests on springs sufficiently extended so that the calculated stress at the elastic limit of the spring would be compared with tests on the elastic limit of the steel from which the spring is made. This involves a very careful set-up with reference to the heat treatment of the finished test bars to bring them to the same general elastic limit.

The writer is in complete agreement with the author that springs in general are not used to a sufficiently high stress, and that for this reason springs do not have the real cushioning effect which they could be designed to give.

There is no question but that the spring problem is closely associated with that of fatigue, and this question also should be considered in determining the maximum allowable stress values to use.

Most problems involving the use of springs can be solved so as to avoid overstressing the springs when unusual conditions are imposed upon the equipment.

The author's work is very much appreciated by the writer.

Jos. B. Reynolds.[11] In discussing the degree of accuracy to be expected from the method of this paper, it seems to the writer that in deducing Equation [13] of the author's previous paper [APM-51-17] from Equation [12], instead of putting $M = Pr$ as was there done, we should set $M = P(r - \gamma)$, giving

$$S''_{a'} = \frac{16P}{\pi d^3}(r - \gamma)\frac{4c - 1}{4c - 4}$$

Now by Equation [7] of the same paper, $\gamma = d^2/16r = r/4c^2$, since $c = 2r/d$ hence

$$S''_{a'} = \frac{16Pr}{\pi d^3}(1 - 1/4c^2)\frac{4c - 1}{4c - 4}$$

If c is as low as 3, this is seen to make a difference of about 3 per cent.

Again, since the moment M is about a horizontal line tangent to the cylinder of the helix, its component about the tangent to the helix is $M \cos \alpha$, where α is the angle of the helical spring. For a closely wound spring $\tan \alpha = d/2\pi r = 1/\pi c$; hence $\cos \alpha = (1 - 1/\pi^2 c^2)^{1/2} = 1 - 1/2\pi^2 c^2$; neglecting powers of $1/c$ higher than the squares. This would give

$$S''_{a'} = \frac{16Pr}{\pi d^3}(1 - 1/4c^2 - 1/2\pi^2 c^2)\frac{4c - 1}{4c - 4}$$

This second correction does not amount to 1 per cent for $c = 3$, but its value increases as the spring deflects.

If this reasoning is correct, it would seem that a theoretical formula containing the coefficient 0.615 gives an appearance of accuracy that the theory does not substantiate.

The composite sections proposed by Mr. Austin are interesting, but, as he mentions, it is difficult to evaluate the worth of these sections because the lack of accurate methods for calculating the stresses set up in such complicated shapes.

Author's Closure

Mr. Austin remarks that the value of G as found from the tests on the springs of largest index should be correct to within about $1/4$ per cent. This coincides with the author's opinion also that the values of G found experimentally are quite near the truth. Tensile tests on this spring material in the heat-treated condition showed values of modulus of elasticity in tension E varying from 39.6×10^6 to 31.1×10^6 lb. per sq. in. the average being $E = 29.9 \times 10^6$ lb. per sq. in. Taking Poisson's ratio equal to 0.3 (the commonly accepted value for steel), this would be equivalent to a modulus of rigidity $G = 29.9 \times 10^6/(2 \times 1.3) = 11.5 \times 10^6$ lb. per sq. in., using the fundamental formula of elasticity connecting E and G. It will be noted that this value is very close to the values found from the tests on the springs of largest index and given in Table 2 of the paper. Frequently, however, spring manufacturers recommend values of the modulus G as low as 10×10^6 lb. per sq. in. for use in design. It is the author's opinion that this low modulus is possibly used to compensate for the fact the wire of the spring may be slightly undersize, or perhaps to compensate for inaccuracy in estimating the effect of the end turns. A modulus of rigidity of 10×10^6 lb. per sq. in. would (taking Poisson's ratio = 0.3) correspond to a modulus of elasticity in tension of

[9] Bureau of Standards, Washington, D. C.

[10] Director of Research, A. O. Smith Corporation, Milwaukee, Wis. Mem. A.S.M.E.

[11] Lehigh University, Bethlehem, Pa.

only 26×10^6 lb. per sq. in., which seems to be an abnormally low value for spring steel. However, the author agrees with Mr. Brombacher that further tests on springs of several different wire sizes would be very desirable. No doubt the most nearly correct way to ascertain G would be to take torque and angular-deflection measurements on a heat-treated straight cylindrical bar, as Mr. Austin suggests, but the author does not believe that a great difference would be found between this method and that of measuring the deflections inside the coils of a spring of large index, as described in the paper.

The author also believes that additional tests as suggested by Mr. Jasper would be desirable. Fatigue tests on actual springs loaded as in service would also be of value. By comparing the results of such fatigue tests with those obtained in ordinary torsion fatigue tests, the effect of surface imperfections in the springs would be clearly brought out.

Regarding Professor Reynolds' discussion, the author cannot agree that more accuracy would be gained by putting $M = P(r - \gamma)$ instead of Pr. It is well known that a force acting at a distance from the center of the wire may be replaced by a couple M equal to the load P times the distance of the resultant shearing force from the axis, plus an additional shearing force P. Professor Reynolds' suggestion is equivalent to placing this shearing force P at a distance γ from the center of the wire section. Since the shearing force is placed eccentrically with respect to the wire section, it will *itself* produce an *additional* twisting moment acting on the section. This additional twisting moment would be equal to $P\gamma$ and when added to the couple $P(r - \gamma)$ would give a couple Pr, as the author used in his derivation. However, the author agrees with Professor Reynolds that the method of derivation given in his previous paper [APM-51-17] is, as mentioned therein, not exact. Nevertheless he believes that the formula thus derived is sufficiently accurate for most practical purposes. It should also be noted that this formula agrees within less than 1 per cent with a more exact value derived by Röver, using the equations of the theory of elasticity for the case of a spring with an index of 4. For this reason, it is the author's opinion that this formula, although derived by approximate methods, is quite close to the truth, particularly since it agrees well with test results.

The forging of a semi-coil from spring steel and heat-treating it to simulate the condition in an actual spring, as suggested by Mr. Austin, appear to be very difficult, although if this could be done more accuracy would be obtained. From a practical standpoint, however, the author believes that the method described in his papers gives sufficiently good results.

The author takes this opportunity to express his indebtedness to those who have contributed to the discussion.

Progress in Fuel Utilization in 1929

Contributed by the Fuels Division

Executive Committee: Victor J. Azbe, *Chairman*, A. D. Blake, *Secretary*, John Van Brunt, C. P. Tolman, S. B. Ely, and W. J. Wohlenberg

ALTHOUGH during the past year there have been no revolutionary developments in the field of fuel utilization, there has been a slow but consistent progression in the direction of greater efficiency and lower operating, maintenance, and capital charges. Of particular importance is the gradual adoption of the more advanced methods by those industries which have until recently shown only a slight amount of interest in fuel economy.

However gratifying may be the progress which has been made, there is still ample room for improvement, particularly in the more general application of existing knowledge and methods. It has been demonstrated, for example, that steam may be produced consistently at efficiencies of 90 per cent and above, but the average efficiency of all steam-generating stations is far below this figure. It is believed that the greatest strides in the future will be made not so much in the improvement of the present best methods, but in the adoption of these methods by a larger portion of the fuel-using industries.

It is obviously impossible in a report of this length to refer to all the detailed improvements which have been made. It is the intention merely to summarize the more important advances which have occurred during the past year.

Solid Fuels

Production of bituminous coal during the year 1928 and the first nine months of 1929 was only slightly below the average for the five-year period 1924–1928. It is highly probable that the output for the year 1929 will be above that for 1928, owing to the general increase in industrial consumption and notwithstanding

TABLE 1 PRODUCTION OF BITUMINOUS COAL, 1918–1929

		Annual Production 1918 to 1928		
		Average no.	Average tons per man	
Year	Net tons	days worked	Per day	Per year
1918	579,386,000	249	3.78	942
1919	465,860,000	195	3.84	749
1920	568,667,000	220	4.00	881
1921	415,922,000	149	4.20	627
1922	422,268,000	142	4.28	609
1923	564,565,000	179	4.47	801
1924	483,686,538	171	4.56	781
1925	520,052,741	195	4.52	884
1926	573,366,985	215	4.50	966
1927	517,763,352	191	4.55	872
1928	500,744,790	203	4.73	...

First 224 Days of Each Year

Year	Net tons
1925	351,405,000
1926	388,336,000
1927	379,015,000
1928	340,118,000
1929	368,797,000

economies in use. In spite of the increased output, the amount of coal held in storage has shown a practically uniform decrease. On July 1 the commercial stocks of bituminous coal in the hands of commercial consumers amounted to about 33,100,000 net tons, the lowest amount reported since 1922.

Table 1 gives the annual production, 1918 to 1928, inclusive, and for the first 224 days, 1922 to 1929, inclusive.

Owing largely to low price levels (averaging the lowest in thirteen years) the past year has witnessed increased and effective efforts to reduce production costs and bring about a greater percentage of recovery. These efforts have been largely centered in the more extended use of mechanical loaders and conveyers of various types, the adoption of improved mining methods and systems, and more careful supervision. Improvements have been made in ventilation and timbering, and there has been a continued increase in the attention given to safety measures.

Particular care has been given to the cleaning and sizing of bituminous coal. Many new installations of mechanical cleaning equipment have been completed during the year, and more have been contracted for. There has been a progressive and important movement toward standardization of the "prepared sizes," with a reduction in their number. Standard sizes have been adopted during the year by two important producing and marketing organizations.

Production of anthracite during the year 1928 showed a slight decrease as compared with that in 1927, and the 1929 production has about kept pace with the 1928 output. In 1929 the output up to September 21 amounted to 52,193,000 net tons as compared with 52,388,000 tons in the corresponding period of 1928.

The anthracite operators and producing companies are continuing to give more concentrated and effective attention to improvements in mining methods and equipment. Preparation, cleaning, and sizing are being given greater attention than ever before, and have resulted in a marked improvement in the quality of coal as shipped.

Owing principally to conditions in the iron and steel trade, the past year has been one of unprecedented activity in the coke-making industry. The total coke produced during the period Jan. 1–Aug. 1, 1929, with that produced in the corresponding period of 1928, is as follows:

	1928	1929
By-product coke, net tons	31,577,542	35,950,237
Beehive coke, net tons	2,829,500	4,282,300
Total, net tons	34,407,042	40,232,537

On January 1, 1929, there were 12,544 by-product and 41,288 beehive ovens in existence; 8300 of the latter type were dismantled during the year. The 86 by-products plants in operation in August were producing about 93 per cent of their capacity.

Liquid Fuels

In spite of efforts to curtail production of petroleum, the first seven months of 1929 showed an increase in production of 12.8 per cent as compared with the corresponding period for 1928. For example, the total domestic production for July-August, 1929, was 576,581,331 bbl. as compared with 511,323,929 bbl. for the same months in 1928.

For several years past and continuing during 1929, improvements have been made in petroleum production methods resulting in appreciably greater recoveries. Repressuring, gas lifts, extraction of gasoline from casinghead gas, and gastight storage tanks are being more generally adopted. Particular attention has been paid to the problem of crooked holes. Improved methods of fractionating casinghead gasoline have placed on the market liquid propane and butane, which products are being adopted for the enrichment of water gas and also as fuel in industrial furnaces.

Refining methods have been improved to give better products, greater yields, and lower costs. The old type of inefficient batch shell stills are rapidly being replaced with continuous tube stills

equipped with heat exchangers, air preheaters, and furnaces designed for high efficiencies and for high rates of radiant-heat transfer. Continuously operating units producing only distillates and coke have come in largely as a result of the fuel-oil market. The use of stills operating at high vacua for handling lubricating stocks has been quite widely extended. Refinery boiler plants have been the subject of increased attention which has resulted in marked economies. The rotary clay burners used for many years are being replaced with shelf-type burners which use less fuel and give a better product. The volatility and anti-knock value of gasolines have generally increased. The end point of motor fuels marketed has shown a constant tendency to be lowered. This period has also been characterized by the development of several commercial processes for producing motor fuels of high anti-knock value and a more widespread use of such fuels by motorists. There seems also to have resulted an increased yield of gasoline as a percentage of the crude.

Perhaps the most interesting development in petroleum refining during the past year is the construction of a 5000-bbl-per-day hydrogenation plant which is now under way and the projected building of two others of equal size. The hydrogenation of coal has been carried out in Germany for several years, and in 1928 300,000 bbl. of gasoline were produced by this process. The plant being constructed in this country, while similar in principle to the German plant, will be used for the hydrogenation of fuel oils and heavy refinery residues to produce high-quality distillates. This process will be watched with a great deal of interest.

Natural Gas

There has been an enormous increase in the utilization of natural gas during the past year which has been brought about by the construction of long mains from the gas fields to various industrial centers. Certain of these lines are from 400 to 600 miles long, and a contemplated line from the Louisiana fields to Chicago will be even longer. The line construction has in almost all cases been steel pipe, using welded joints in the smaller sizes and Dresser couplings in the larger sizes. There has been some difficulty in changing over from manufactured to natural gas due to the fact that natural gas is essentially dry and therefore has a deleterious effect on the joints and meters of the typical system handling artificial gas. The difficulties have been overcome in several instances by supersaturating the gas with gas oil prior to its distribution through the city mains. The conservation of natural-gas resources has been the subject of much interest. Legal means have been taken to require the economic use of gas for repressuring oil fields. The biggest step in this direction has taken place in California during the past year, and is a distinct advance in the economic development of both gas and oil, with a minimum wastage of both.

Manufactured Gas

During the past year there has again been a decided increase in coke-oven-gas production and a decrease in water-gas production. Data given by 96 typical gas companies indicate that for the first five months of this year there was a 6 per cent decrease in water gas produced, a 12 per cent decrease in coal gas produced, a 39 per cent increase in coke-oven gas produced, and a 22 per cent increase in coke-oven gas purchased from coke and steel companies as compared with the corresponding five months of 1928. The total gas sold during these five months was 9.4 per cent higher in 1929 than in 1928, due largely to the increased use of gas for house-heating and industrial purposes.

There have been no outstanding new developments in the manufacturing field. The matter of dry quenching of coke and the dehydration of gas mentioned in last year's report are still very much in the limelight and are both apparently being received with increased favor. The Fort Wayne dehydration plant has now been in operation sufficiently long so that data are available, and the possibility of thereby producing non-corrosive, clean gas has awakened increased interest. A new dry-quenching plant is soon to be placed in operation at Flint, Michigan, which will be the second installation in this country. The original installation at Rochester has been in service three full years, and the results there have been very favorable. Considerable progress has been made this past year in the method of disposing of phenol from the coke-plant ammonia-still waste, and it is now possible to eliminate this source of stream pollution which has in the past been a very troublesome problem.

There has been a continual increase in the number of waterless holders for the storage of gas at relatively low pressures. The largest holder in the world, having a capacity of 20,000,000 cu. ft., which was recently placed in service in Chicago, is of the waterless type. Experience throughout the country in the use of these holders has been generally satisfactory. The use of high-pressure holders has met with increased popularity during the past year. Strategic location of these holders at points along the transmission line and at local distributing points has resulted in improved load factors on the transmission equipment.

Low Temperature Carbonization

In recent months a 600-ton-per-day K.S.G. low-temperature-carbonization plant has been placed in operation near New Brunswick, New Jersey, by the International Coal Carbonization Co. The semi-coke produced is being marketed locally as a domestic fuel, and the coal gas, mixed with blue gas, is sold to the Public Service Gas and Electric Corporation of New Jersey. Since this plant is the first of its type erected in this country and is moreover the largest of any type in the United States, much interest is being displayed in the results obtained. A similar plant of 50 tons per day capacity is under construction at Coatesville, Pennsylvania.

A "Lurgi" process plant having a capacity of 100 tons of briquets per day has been in operation at Lehigh, North Dakota, since December, 1928. It is adapted to North Dakota lignite and produces a carbonized-coal residue which is briquetted with pitch obtained from the process. So far it has given the results promised by the Lurgi Corporation.

Several experimental plants of other processes have been in operation during the year and have been more or less successful. It is reported that test work on the Carbocite plant at Philo, Ohio, has been completed, and that a commercial plant is to be designed. Plans are also under way for the erection of a Hayes process plant with a capacity of 400 tons per day.

Boilers

The outstanding point of interest in boiler development has been the radical increase in size as illustrated by the new units, each having a capacity of 800,000 lb. of steam per hour, which are to be placed in service at Hell Gate Station. These will supply steam for a 160,000-kw. turbo-generator, which is the largest single-shaft, single-unit generating machine in the world.

There has been an increase in the number of installations generating steam at pressures of 1200 to 1400 lb. per sq. in. Until recently high pressures have been used only for base-load conditions, but at least two high-pressure installations are now being erected which will operate at variable loads. In Europe pressures have gone much higher than 1400 lb. The Benson boiler, which generates steam at a pressure of 3200 lb. per sq. in., has been thoroughly tested experimentally, and large-size plants are being erected in England and Belgium.

The possibilities of using higher steam temperatures are attracting much attention. Practice in this country has not yet gone above 750 deg. fahr., but there are several experimental and

semi-commercial plants in Europe which are operating at maximum temperatures of 900 to 950 deg. fahr. A plant in Belgium is operating commercially with a steam temperature of 800–825 deg. fahr. The Detroit Edison Company will shortly have in operation a 10,000-kw. turbine designed for an initial steam temperature of 1000 deg. fahr. The success of this experiment will be watched with interest.

There are said to be a number of boiler plants in Europe operating on the Löffler principle. A plant of this type at the Vienna Locomotive Works which generates steam at 1700 lb. per sq. in. and at 900 deg. fahr., has been in operation for about a year and has given satisfactory service. Improvements have been made in the Atmos boiler (described in MECHANICAL ENGINEERING, vol. 45, p. 253, April, 1923), and a unit having a capacity of 26,000 lb. of steam per hour at 1400 lb. per sq. in. and 840 deg. fahr. is being erected.

Some experimental work on the development of drumless boilers has been carried out in this country, and it is possible that high-pressure boilers in future years will be of this type.

An interesting European development is the use of steam accumulators for handling peak loads in central power stations. An order has been placed for the installation of 16 accumulators at the Charlottenburg Station of the greater Berlin system. Each accumulator will be 14 ft. 9 in. in diameter and 69 ft. high, with a capacity of 11,000 cu. ft. The combined steam-storage capacity will be sufficient to generate 67,000 kw-hr.

Boiler Furnaces

The outstanding feature of the great majority of recent boiler installations is the almost universal use of furnace-wall cooling by means of air or water, with the trend apparently in favor of the latter. The reduction of refractory maintenance costs has been so marked that even the very expensive types of furnace-wall construction have proved to be good investments. At the same time the increased steaming capacity given by the extended radiant surface has reduced the fixed charges per unit of steam generated. In some cases, in which the furnace is entirely enclosed with water walls, as much as 40 to 65 per cent of the total heat in the steam is being absorbed by the water walls. Whether bare tubes or refractory-faced tubes are preferable is still a controversial question. Both types seem to be giving good service. In some cases both are used in the same furnace. In the latest pulverized-coal furnaces with bare tubes, blocks are being used for local protection against the effect of dripping slag. Facings of materials such as carborundum continue to attract attention because of their apparent ability to stand up under intense firing conditions.

The earlier expressed fears of incomplete combustion due to cooling of the reacting products have not been realized. The selection of the proper amount of water cooling to use in any particular installation has proved to be a question of economics rather than of combustion. Many stoker-fired furnaces are being equipped with local water-cooled-block-covered surfaces along the rear and side walls of the stoker. In the arched furnaces with chain-grate stokers the water-cooled surfaces covered with high-quality refractory blocks have made a good showing when applied to arches and side walls. The troubles with the arch support have been largely eliminated.

During the development stage the water-cooled furnaces had to be fitted to standard boiler designs. With such combination many downcomers were necessary to supply water to the water-cooled furnace walls and many risers to carry away the steam. Strong efforts are now being exerted to so modify and simplify the furnace and boiler design as to greatly reduce the number of risers and downcomers.

Although the development of the slagging-ash type of furnace is a comparatively recent one, its adoption has been rapid, particularly during the past year. Fourteen furnaces so equipped are in operation, and twenty more are under construction. The furnaces for the high-capacity boilers under erection at Hell Gate Station are of this type. They have proved particularly applicable to the handling of low-grade fuels.

Another development which may prove applicable to the use of low-grade fuels is the mechanical gas producer applied to boiler furnaces. Experiments being carried out in this country have given gratifying results. It is reported that several commercial installations have been ordered in England.

Stokers

The time elapsed since the writing of the last report has been marked by a keener competition between stokers and pulverized-coal firing, the stokers regaining some of the lost ground. The development of stokers has taken place along two different lines, in each case being a result of commercial demands. Stokers for large units have advanced principally in size, while the smaller units have been made more convenient. To meet the new demands, stokers have been increasing in size rapidly. Underfeed stokers have been installed for units of 500,000 lb. per hr. steaming capacity, and the manufacturers are ready to meet any demand up to 1,000,000 lb. per hr. The chain-grate stokers have lagged somewhat behind the underfeed, principally because of the type of fuel which they handle. Coincident with increases in size there have been increases in the working combustion rates of all classes of stokers. At the present time the practical limit of combustion rates is often determined not by the stokers but by the slagging of boiler tubes.

The use of air preheaters as a final heat-absorbing medium is widespread, and consequently stokers have had to be adapted for use of preheated air. This has been done by using improved materials, allowing for structural expansion, and increasing the size of the air openings in the various types of grates. The maximum air temperature allowable for use with stokers depends to some extent upon the fuel used. Some coals have a tendency to fuse at comparatively low temperatures, obstructing the passage of air through the grates. In some cases an air temperature of 300 deg. fahr. has been found to be the maximum allowable on this score. In general, it has been found that stokers may be operated with an air temperature of 350 deg. fahr. with only a slight increase in maintenance. Certain improvements in the materials used for tuyères and grates permit running with 500 deg. fahr. air in some isolated cases. However, the average allowable air-temperature range is already sufficient for a proper proportioning of the heat-absorbing surfaces in a complete steam generator.

The flexibility of stokers as to the range of statisfactory combustion rates and types of fuel burned has shown a marked improvement. The timing device of one stoker allows a hundredfold variation between the minimum and maximum rate of feed. The introduction of steam-turbine and hydraulic drives for stokers facilitates close regulation.

Concerning the types of coal burned by stokers, the progress made so far is closely connected with the furnace-wall construction and the temperature control effected by preheating the air on the one hand, and by water-cooled walls on the other. The overfire injection of air seems to be a success, especially with high-volatile coals. Small stokers developed for industrial boilers by many firms reached a high degree of efficiency and reliability of operation. Good practical results were obtained with an arch-less chain-grate stoker in which an early ignition of coal is secured from the grate bars upward. Many improvements in the mechanism and motion of the stokers have been effected, with the result that wear and tear is reduced.

The combustion efficiencies attained on stokers with improved draft distribution are excellent. With the deep ashpits the unburned carbon is reduced to a negligible amount—not higher than in the same size of pulverized-coal furnace.

Pulverized Fuel

Although recent improvements in stoker design have made possible the installation of large units, pulverized coal has proved more popular in most of the larger boiler furnaces. The new Hell Gate boilers, for example, which are the largest in the world, are to be fired with powdered fuel. This is due principally to the ease with which the pulverized-coal systems may be expanded, simply by multiplying the mechanical apparatus. An increase in size of stokers, on the other hand, generally means an entirely new design.

The storage and the direct-firing systems are now used both in central stations and industrial plants. The direct firing appeals strongly by its simplicity, especially for industrial plants where the steam-generating units are comparatively small, or where stoker-fired boilers are changed to pulverized coal. A number of central stations have also adopted the direct-firing system, and those plants that have been in operation for any length of time are obtaining satisfactory results. There are, however, few definite data available for comparisons of the economic performance of the plants using the direct-fired system with those using the storage system. There are still many engineers who prefer the storage system, at least for large installations, and they are able to present good arguments for their preference. In order to better compete with the direct-firing, the storage system is undergoing simplification. The driers are in most cases eliminated, and what drying is needed is done in the mill during the process of pulverization.

Although a great deal has been said and written about the possibilities of high rates of combustion of pulverized coal, the most practical rates are still between 1 and 2 lb. of coal per cu. ft. of total combustion space per hour, or, a rate of heat liberation of 12,000 to 25,000 B.t.u. per hr. There are a few exceptions where rates of heat liberation of 35,000 to 40,000 B.t.u. per cu. ft. of total furnace volume have been obtained, but such rates can be maintained only in water-cooled furnaces and with coals that have ash of a comparatively high fusion temperature. Generally the rates of heat liberation are kept lower on account of excessive erosion of furnace refractory and the slagging of boiler tubes.

In solid-refractory-wall furnaces and with coals of low ash-fusion temperature, rates of heat liberation of 10,000 to 12,000 B.t.u. are good practice; with coals having high-fusion temperature ash these rates are 14,000 to 17,000 B.t.u. In air-cooled-wall furnaces the rates are raised to 14,000 and 20,000 B.t.u. according to the fusion temperature of the ash. In water-cooled furnaces rates of 25,000 to 35,000 B.t.u. are practicable, provided that proper precaution has been taken to minimize the possibility of slagging the boiler tubes and stopping the gas passages through the boiler. Generally speaking, in smaller water-cooled furnaces the rates can be higher than in larger ones, because the ratio of water-cooled surface to combustion space is greater and the products of combustion are cooled to a greater extent by the time they enter among the boiler tubes, and there is less tendency for the ash to be deposited on the boiler tubes and stop the gas passages.

The above-mentioned rates of heat liberation are the average of the entire furnace volume. Near the burner where the combustion is very rapid the rate may exceed 100,000 B.t.u., while in the space near the boiler tubes it may be less than 1000 B.t.u.

There are three methods of firing in general use, namely, vertical firing, horizontal firing, and corner or tangential firing.

Vertical firing was the first to be developed, and many of the large plants burning pulverized coal are using this method. In most of the first installations only the primary air was supplied under pressure; the secondary air was drawn into the furnaces at low velocities through air ports in the front wall by the furnace draft. In more recent installations, and particularly where air heaters are used, the secondary air is supplied under a pressure of 1 to 2 in. of water, making it possible to admit the air into the furnace at high velocity. Part of the preheated air is also supplied under pressure through the burner bodies around the fuel nozzles. The air pressure and the resulting higher velocities produce intensive mixing and rapid combustion.

Horizontal firing is gaining in favor. Usually its installation is more simple and somewhat cheaper than that of vertical firing because no furnace arches are necessary. The burners are placed in the front wall, and generally all the secondary air as well as the primary air is supplied through the burner under a pressure of 2 or 3 in. of water. While passing through the burner the mixture of primary air and coal and the secondary air are given a somewhat rotary motion, which produces turbulent mixing in the furnace. These rotative burners are built in capacities of 2000 to 8000 lb. of coal per hr. Certain of these burners are adjustable so that the cross-section of the air passages can be decreased when the amount of coal to be burned is reduced, in order to make it possible to maintain fairly high velocities and good mixing.

In a few installations two sets of these burners are used, one in each of two opposing walls, so that the flames from the two sets meet in the middle of the furnace and produce intensive mixing. This is called "opposed horizontal firing," and it is especially well adapted to large units with deep furnaces in which it would be difficult to install enough burners in one wall and fully utilize the available combustion space.

In the tangential method of firing, streams of the mixture of coal and air are projected horizontally from the four corners of the furnace toward the center in such a way that the streams are tangent to a small circle in the center of the furnace. Preheated secondary air is supplied under a pressure of 2 to 4 in. of water around the burner nozzles in the same direction. The streams of coal and air impinge against one another and produce intensive mixing and rapid combustion. The furnace gases move in a helical path toward the gas exit from the furnace. This helical flow maintains the turbulence and rapid combustion. Rates of heat liberation as high as 40,000 B.t.u. may be obtained, with satisfactory results as to efficiency and continuity of operation. Inasmuch as the flames are confined by the furnace walls in their helical path, there is considerable impingement of the flames against the walls. Therefore tangential firing can be applied only to completely water-cooled furnaces. Refractory furnace walls would be rapidly wasted away.

A number of installations with tangential firing have been put in service during the last two years and show the highest rates of heat liberation of any of the powdered-coal plants so far installed.

Most of the pulverized-coal installations are designed for the removal of refuse deposited at the bottom of the furnace in granular form. In such installations the deposited ash is prevented from fusing by a water screen placed over the bottom of the furnace or by a water-cooled furnace bottom.

As mentioned elsewhere, a number of installations have been made within the last two years in which the ash deposited at the bottom of the furnace is fused and removed in the liquid state. A spray of water is generally used to chill and disintegrate the molten ash into granular form. The disintegrated slag is pumped away in suspension.

Inasmuch as no refractory will stand the erosive action of molten slag, it is essential for low maintenance that the walls of the furnace be completely water cooled. On the first installations

of slagging furnaces it was thought necessary to protect the furnace bottom with a layer of dolomite sand. Experience with more recent installations indicates that this is not necessary, provided that the tapping hole is not less than about 10 in. above the brick lining of the furnace bottom.

The slagging furnace is especially well suited for burning coals with very fusible ash. If the ash is of high fusion temperature the burners should be located so as to cause the flame to impinge against the refuse collected at the bottom of the furnace, heating the refuse to as high a temperature as possible.

Roller mills and ball mills are more extensively used for pulverization than other types, because they pulverize coal to a high degree of fineness, have high capacity, low power consumption, and low maintenance. They are built in capacities up to about 40 tons per hour. In the past they have been used almost entirely in installations using the storage system, but within the last two years they have been applied to direct firing with considerable success. It appears that in the future they will be used extensively for direct firing of the larger-size steam-generating units. They are now built with a continuous system of lubrication, and can be kept in operation indefinitely.

High-speed impact mills are used largely for direct firing of small and medium-size units. They are particularly well adapted for installations where the space available for mills is small. Being of high speed, they have a high capacity for their size. They are also less sensitive to the flow of air with variable output. However, the fineness of the coal is less constant with variable output and the maintenance is somewhat higher than with the roller and ball types of mills.

The ball or tube type of mill is used for milling anthracite, coke, and other hard and abrasive fuels, and in cases where extreme fineness is necessary. While with such hard and abrasive fuels the mill maintenance with other types of mill would be prohibitive, it is comparatively low with the ball- or tube-type mill. The tube mill being a machine of a very low speed, its size is large for a given output. It is used for direct-firing as well as for storage systems.

The fineness to which a coal should be pulverized is a controversial question and varies with the character of the coal and the size of the furnace in which the pulverized fuel is to be burned. The higher the fixed carbon content, the higher the fineness to which the coal should be pulverized. Thus, according to one authority, coke and anthracite should be pulverized to a fineness of 80 or 85 per cent through 200-mesh screen; Pocahontas and New River to 70 to 75 per cent, Eastern bituminous coal to 65 or 70 per cent, Illinois to 50 to 55 per cent, and Texas and North Dakota lignite to 40 or 45 per cent through 200-mesh screen.

In cases where the furnaces are of necessity very small, as in marine and locomotive boilers, all coal should be pulverized very fine to obtain satisfactory combustion results. A fineness as high as 80 per cent through 300-mesh screen is believed necessary for Scotch marine boilers.

The principal developments in pulverized-fuel burning during the past year are not so much improvements of the methods in use in central stations, but rather the adoption of powdered-fuel firing to other equipment. Experiments have continued on the problem of the marine use of pulverized fuel, and in most cases have been highly successful. Several ships were equipped during the past year and have given gratifying results. One vessel, three of whose six boilers were fired with pulverized coal, completed a voyage of 11,000 miles during which five kinds of coal were used with entire success. The results were so satisfactory that the remaining three boilers are to be equipped for burning pulverized coal.

The application of powdered coal to locomotives has apparently been successful in Germany, and at least one locomotive in this country is so equipped. There has been a growing interest in the use of pulverized coal in industrial furnaces of various kinds.

Much has been published recently concerning the use of pulverized fuel in Diesel engines. Although the idea is by no means new, the mechanical problems have in the past resisted solution. However, several European investigations have recently brought out designs which are said to be satisfactory and may prove to be popular.

Fuel Oil

With the continued overproduction of petroleum the price of fuel oil has remained low, and this fuel has become increasingly popular for certain classes of service. Partly as a cause of and partly as a result of the growing popularity of oil, there have been major improvements in burner design, and there has been a decided tendency on the part of the fuel user to adopt the more modern equipment. The home-made burner of pipe fittings which required large quantities of steam and still gave unsatisfactory atomization, is being replaced by more efficient steam atomizing burners and by mechanical atomizing burners which can be adjusted to give the type of flame desired with a minimum of excess air.

Diesel Engines

The greatest advance in this field during the past year has been in connection with the use of high-speed Diesel engines. Increases in speed and decreases in weight per horsepower have made possible new and interesting applications. Experiments have been made with a Diesel-motored airplane, and a plane so equipped was actually flown for several hundred hours. Investigations of Diesel-powered trucks have continued and give promising results. In Germany they have been in use for several years, and more than 100 are in regular service.

A recent development in the field of large Diesel engines is their use in central power stations to handle peak loads. Several European systems are now installing them for this service, but it is probable that their high first cost will preclude their rapid adoption for peak loads in this country.

Locomotives

It has been reported recently that an oil-burning locomotive of the Kansas City Southern Railroad has been equipped with a unit system of pulverized-coal firing and has given excellent results. It is stated that the fuel consumption is 25 per cent less than with a similar hand-fired furnace, and that the steaming capacity is equal to that of an oil-fired boiler. The fuel cost is only 40 per cent of the cost of oil firing. The coal fed to the pulverizer is 1-in. slack, costing only about half as much as run-of-mine.

Other than this there have been no fundamental innovations in the design of locomotive furnaces or spectacular reductions in fuel consumption in this country during the year. Superheaters, brick arches, thermic siphons, and mechanical stokers continue to be applied as a matter of course, but there have been no radical changes in their designs.

There has been a continued and perhaps an increased interest in the proper preparation of coal for locomotives and in delivering it on the tender in good condition. This last purpose has entailed some novelties in the design of coal chutes. Within the past four or five years there has been a considerable revival of interest in the design of locomotive grates, and this has received additional attention during the past year.

There continues to be a determined effort to make use of poorer grades of fuel where it is easily accessible, as, for example, the utilization of lignite in Texas and in the Northwest. On certain railroads this effort has been recently intensified.

In Europe the developments have been more radical. In ad-

dition to the Schmidt and Löffler-principle locomotives mentioned in last year's report, there is now under construction a turbo-locomotive the steam for which will be furnished by a Benson boiler under a pressure of 3300 lb. per sq. in.

Industrial Furnaces and Power Plants

There has been a noteworthy improvement in industrial fuel economy during the past year, particularly in the steam-generating end. There has been a decided attempt in the building of new units to select a pressure such that a proper balance is obtained between heat and power requirements. Several industrial plants have selected the present upper limit of American practice as the boiler pressure best suiting their requirements. In fact, one industrial boiler plant now being erected is intended to operate at 1800 lb. per sq. in., which will be the highest steam pressure in use in this country. Plant executives are waking up to the importance of fuel economy and are making much-needed improvements.

Heat-treating furnaces of various kinds have shown improvements recently. There is a strong trend in favor of continuous furnaces with automatic control. Insulation is being applied to a greater extent, although there are many cases where the high temperature level of the operation precludes the possibility of insulation without undue deterioration of the refractories. There continues to be a greater use of the fuels which lend themselves to ease of handling.

Domestic Heating

The demand of the householder for liberation from the drudgery of furnace tending is determining the developments in the field of domestic heating. There has been a great increase in the number of oil burners installed, and gas-fired furnaces also have had a steady growth. There are some isolated installations of electric house heating in which off-peak power is used.

The competition from gas and oil has caused the development of the domestic stoker designed for burning small sizes of anthracite and bituminous coal. During the past year these stokers have been improved, and many new designs have appeared which have been more or less successful. The major drawback to the general adoption of domestic stokers, not considering the fact that they require more attention than oil- and gas-fired furnaces, is their inability to handle all classes of coals. It is sometimes difficult or impossible for the householder to obtain the type of fuel for which the stoker was developed, and there are often great difficulties in attempting to burn coal for which the stoker is not adapted.

Refractories

Although the manufacture of refractories is not altogether a problem of fuel utilization, still refractories are such an important part of most combustion equipment that a short summary of the developments in this field will not be out of place.

During the past year there has been a general adoption of manufacturing methods in refractories which, while not entirely new, would have been considered revolutionary several years ago. The tunnel kiln has been adopted for burning fireclay, silica, and chrome brick, and many labor-saving devices have been installed. Several new refractories have come into prominence recently, among which are unburned magnesite brick, mullite, sillimanite, and refractory shapes made from a mixture of chrome and diaspore. Diaspore refractories which have been subjected to an unusually high heat treatment are available commercially, and development work is being carried out on the manufacture of light-weight cyanite brick which combine heat-insulating properties with high refractoriness. Progress has been made in the design of special refractory shapes for blast-furnace stoves and open-hearth regenerators. The new shapes are said to result in large gas savings.

Considerable research on the failure of refractories in service is under way in various laboratories, and studies are also being made on various phases of manufacturing methods.

Research

Fuel-research activities have shown a notable increase during the past year, both abroad and in this country. The subjects on which work is being done seem to cover almost the entire field, but unfortunately in the United States there is very little coordination of effort. There is a certain amount of cooperation between various professional societies, but this country has no really sound fuel-research program.

In spite of the handicap, however, research is going on and valuable results are being obtained. An important activity started two years ago and which is continuing is the study of the coking and gas-making properties of American coals conducted by the American Gas Association in cooperation with the United States Bureau of Mines.

The research projects sponsored by the American Petroleum Institute are being carried on by a large group of investigators located in various educational institutions throughout the country.

The subject of pulverized coal is receiving more attention than ever before; particularly its combustion and use, and including also the economics of pulverized coke as a fuel.

The work of the coal-classification committee of the American Society for Testing Materials, which work involves both pure and commercial research, is being continued and good progress is being made.

The fuels-research work of The American Society of Mechanical Engineers is being reorganized. A general conference of leading engineers and chemists was held in October for the purpose of considering and recommending subjects, plans, and methods.

Research work of the National Coal Association is progressing principally in the direction of marketing, distribution, and use.

During the year the Anthracite Operators' Conference has started studies of domestic heating appliances and the factors influencing their performances.

The United States Bureau of Mines, in cooperation with The American Society of Mechanical Engineers, has done a great deal of work on refractories and on the slagging-ash type of furnace. The Bureau is also resuming its study of brick kilns in cooperation with the University of Ohio.

Some of the university laboratories have made important contributions during the year and are continuing their efforts.

In general, it may be said that there is more fuel research now in progress than at any previous time, and the outlook is encouraging in spite of the handicaps. It is interesting to speculate as to what might be accomplished by an amply financed organization devoted to the cause of fuel research. It is to be hoped that such a speculation will not be altogether an idle one.

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Progress in Steam-Power Engineering

Contributed by the Power Division

Executive Committee: F. M. Gibson, *Chairman*, Alfred Iddles, *Secretary*, V. E. Alden, V. M. Frost, and W. F. Ryan

THE developments in 1929 have followed the trend of 1928, and a continuance through 1930 is probable. The year has witnessed larger mergers of invested capital, larger interconnections of central stations, larger industrial heating loads as well as larger power loads diverted to central stations, larger stations, larger equipment units, and larger demands for electrical power. The single word that is most descriptive of this advance is "magnitude." Improvement has been made in more clearly defined fields of application of types of equipment rather than in any radical change in design. So great is the number of possible combinations of types of equipment that can be made to meet the singular conditions of the individual station, that the employment of a particular type of equipment in the design of a new station is not necessarily indicative of the general trend of practice.

Recent and anticipated mergers of large power corporations presage a hastening of the interconnection of major distributing systems, thereby making possible greater modern stations with larger equipment units, the reduction of investment in stand-by equipment, the more effective use of water power, and increased research, all of which tends toward a lower cost of production with a consequent increased demand for power.

Developments in Boiler-Plant Equipment

The most outstanding developments in boiler-plant equipment are the rapidly increasing capacity and continuity of service of steam-generating units. Within the current year there will be placed in service several such units, each having a maximum capacity of nearly one million pounds of steam per hour. One plant, equipped with water-cooled walls and pulverized-fuel firing, operated eight months with no outage at time of demand. The effect of water-cooled walls upon the elimination of outages by means of their reduction of repairs to furnaces and combustion equipment, together with their quicker cooling at times of repairs, has led to their almost universal application in larger installations and their more general use in smaller units. There are now available individual steam-generating units capable of operating a turbine of over 80,000 kw. capacity and having a reliability nearly equal to that of turbo-generators. This fact will probably have considerable influence upon the design of future power plants.

In stations with a high load factor there has been an increase in the number of plants designed for steam pressures of approximately 1400 lb. In the design of these plants there has been a decided tendency toward straight-tube, cross-drum boilers, although the more recent and improved design of bent-tube boilers may alter this trend. The advantage of limiting hot-metal spots in superheaters has evidently been sought in boilers and water-cooled furnace walls, resulting in a reduction of heat transfer per square foot in order to reduce maximum metal temperatures. A few stations of moderate load factors have been designed for 650 lb. steam pressure, while nearly all new stations of low load factor are designed for pressures of about 450 lb.

There has been no trend toward higher steam temperatures except for experimental purposes, and though designers have preferred greater reliability and less efficiency at 750 deg. fahr. (and probably will until less expensive materials are available), there are positive indications that higher temperatures will be used in the near future. Two installations for use in this country have been purchased for steam temperatures above 800 deg. fahr., and actual experience with these temperatures should be available soon. In one comparatively new station, interdeck superheaters with plain steel tubes have been operating at a steam temperature of 780 deg. fahr., and this temperature will be increased gradually to a point in excess of 800 deg. or until signs of distress develop. Experiments are being made on an oil-fired superheater of a capacity of 6000 lb. of steam per hour at steam temperatures of from 700 to 1100 deg. fahr. for the purpose of determining the effect of high temperatures upon superheater tubes, valves, piping, and fittings. Manufacturers are prepared to build superheaters, turbines, and piping for steam temperatures of 800 deg. fahr. Other trends in superheater design that will assist in promoting higher temperatures are in the direction of increasing the velocity of steam through the elements, reducing it through the inlet and outlet headers, thus insuring proper distribution of steam between elements, and in providing greater protection of the superheater with an intervening bank of boiler tubes.

With a steam temperature of 750 deg. fahr. practically all stations designed for pressures over 450 lb. are reheating the steam. Of two stations designed for a steam pressure of 650 lb. at 750 deg. fahr., one does not employ reheating while the other reheats to a temperature of 500 deg. In plants of steam pressures higher than 650 lb. the steam is universally reheated to the original temperature of the steam. Single-stage reheating has been used in all of these stations with the exception of two now under construction. In these two stations there is a combination of a steam reheater with a gas reheater. The temperature of the reheat is maintained nearly constant by thermostatically controlling the temperature of the steam from the steam reheater by the temperature of the steam leaving the gas reheater.

In high-steam-pressure stations the total heating surface of air preheaters, reheaters, bleeder heaters, water-cooled furnace walls, and economizers has increased until the actual boiler heating surface is only 12 per cent of the steam-generating-unit heating surface, and only 0.3 sq. ft. of boiler heating surface per kilowatt of plant capacity. Under these conditions the terms "boiler rating" and "boiler horsepower" have little or no significance.

Trend in Fuel Firing—Feedwater Heating

The trend in fuel firing in recently designed or constructed large stations is decidedly toward pulverized fuel. In the central stations the bin system seems to predominate, while in the industrial field the unit system is favored. Pulverizing mills with a capacity of 25 tons per hour are now in operation, and one station with larger boiler units is using as many as five mills per boiler. The largest steam-generating unit soon to be placed in operation is designed for pulverized fuel with the unit system. The rate of combustion in recently designed pulverized-fuel furnaces of large size range from 18,000 to 30,000 B.t.u. per cu. ft. of furnace volume. Several slag-tap furnaces are in operation with a combustion rate of 42,000 B.t.u. Although this type of furnace requires a minimum fusion temperature of ash of 2500 deg. fahr., it may be of special advantage where the disposal of fine ash from the furnace is a problem. There is as yet no completely satisfactory method for the removal of ash carried up the stack.

While the trend has been toward pulverized fuel and there seems to be no limit to the capacity of the pulverized-fuel-fired furnace, the progress of the underfeed stoker is impressive. Nearly every month of the present year has established a record in the number of retorts sold, that for August being 1900 retorts. There are now available individual stoker units capable of producing 500,000 lb. of steam per hour, each which with double-end firing makes possible the output of 1,000,000 lb. per hour from a single stoker-fired boiler. One manufacturer has produced a ram capable of feeding 40 lb. of coal per stroke. Experiences of the past year indicate that a gradual and complete redesign is taking place. The development has not been confined to any one part, but the most noteworthy improvement has been in the handling of the fuel bed. This has been accomplished by reduction of wind-box air pressures and better methods of conditioning to provide a light and spongy fuel bed.

There has been no outstanding change in the design or application of air preheaters. Accurate test and operating data are now available for determining design. Gas spacing has been determined for different fuels, and means provided to maintain this space free from slag or ash. Better provision has been made for expansion to prevent warpage of elements, and considerable attention has been given to the sealing against air infiltration.

Feedwater heating shows an apparent trend toward the more effective use of the regenerative cycle. Four-stage heating is quite common, and in one station the feedwater will be heated in five stages. Increase in steam pressure justifies an increase in the temperature of feedwater by means of bled steam. This is evidenced by the heating of feedwater to a temperature of 430 deg. fahr. in a station recently designed for a steam pressure of 1400 lb.

Never in the history of the profession has feedwater treatment been given as much attention, both in research and in practice, as in the past year. One significant trend is the practice of combining processes. Combinations of chemical and zeolite softening, addition of coagulants with softening reagents, the continuous blowdown system, and steam purifiers have brought a change in feedwater treatment and provided the designing engineer with a greater number of methods from which he can choose to treat a specific condition. Although research is still being made to determine the process by which electrolytic methods prevent corrosion, several successful installations have recently been made. There has been a decided increase in the use of sodium aluminate and phosphates.

Several instances of severe embrittlement in boiler steel have occurred during the year, and it has been found in highly stressed areas, in expanded as well as riveted joints. When the ratio of sodium carbonate alkalinity to sodium sulphate was established by the Boiler Code Committee, there were not sufficient operating data to warrant carrying the ratio above 250 lb. pressure. Later sufficient data were developed to warrant extending the ratio to 600 lb. pressure, the ratio increasing in proportion to the increase in pressure as indicated in the ratios for lower pressures. When the ratio is applied to higher pressures, the amount of treatment becomes excessive. Laboratory investigation indicated phosphates as a desirable inhibitor of embrittlement. Many stations began the use of di- and tri-sodium phosphates and, more recently, mono-phosphate. There have been no known ill effects to date, but the use of phosphates has not been practiced for a sufficient length of time to positively substantiate laboratory results.

Larger Turbine Installations

The growth of power systems has made larger turbine installations economically possible. During the past year several noteworthy installations have been completed, and orders have been placed for others of unusual interest. All of the larger installations are of special designs built to suit special conditions to fit in with the power systems and the economic policies of the power corporations. There are also quite a number of installations using simple turbines of a more standard design. In one instance a public-utility corporation has installed eight duplicate units with machine tolerances limited so that all parts of the machines are directly interchangeable. The largest single generator unit, 60 cycles, 1800 r.p.m., now on order is a 115,000-kw. and the largest system of units with three generators, 60 cycles, 1800 r.p.m., in operation is for 208,000 kw. One outstanding installation is a tandem compound turbine with a single generator having a double winding with a capacity of 160,000 kw. An interesting development of the year has been the purchase of three steeple compound units. An indication of another possible development is the recent construction of a small turbine with a welded steel-plate casing.

Recently designed stations indicate a decided increase in the use of single-pass condensers. The amount of condenser surface per kilowatt of turbine capacity shows a downward trend. Several stations have been designed for 0.8 sq. ft. and one station for 0.575 sq. ft. of condenser surface per kilowatt of turbine capacity. One outstanding development of the past year has been the general use of condensers with tubes rolled in at both ends and methods for taking care of the expansion of tubes and shell. There are four methods of accomplishing this, but no positive indications exist which forecast the method of the future. Another outstanding development has been the installation of a welded steel-plate condenser. This unit, of 27,000 sq. ft. of condenser surface, serving a 30,000-kw. turbo-generator, was designed, constructed, and placed in service in five months at a cost considerably lower than any quoted on a cast-iron condenser. The performance of this unit shows the air leakage to be very low.

Installations with steam pressures up to 1200 lb. with a temperature of 750 deg. fahr. are common, and stations with a steam temperature of 850 deg. fahr. are being constructed. Experiments with a temperature of 1000 deg. fahr. are being made. A 10,000-kw. turbine is being constructed abroad for installation in a central station in the United States for operation at 365 lb. inlet pressure, an exhaust pressure (absolute) of 1 in. of mercury and an initial temperature of 1000 deg. fahr. This unit is for experimental purposes to determine the behavior of the metal in the turbine, and especially of valves and joints, at high temperatures under operating conditions.

Industrial Steam Power

Industrial power constitutes only a small portion of the total power generated in the country, and the fuel consumed by industrial boiler plants, exclusive of railroad and gas plants, is several times that used by central stations. This condition illustrates the comparative importance of the generation and utilization of power with that of heat in the industrial field. The increased use of steam-flow meters has resulted in greater efforts to improve the efficiency in the utilization of heat. This has constantly reduced the heating load, while the installation of automatic machinery and materials-handling equipment has increased the power load. For this reason the balance of steam systems is destroyed in many plants. During the past year there have been a number of complete boiler and power plants installed as part of extension or modernization programs, but the greater part of recently installed equipment has been for the purpose of restoring the balance of steam systems. This has been the situation in textile, food, chemical, and paper plants, particularly in the latter. Advantages have been taken of higher steam pressures and high-back-pressure and extraction types of turbine delivering steam to process or to the original boiler-steam mains.

There has also been a decided increase in substituting high-back-pressure turbines for motors in power-plant auxiliary drives as well as in service-equipment drives. The increased use of refrigeration has been of considerable help in maintaining steam balances. When extraction turbines are used, particularly in the paper and textile plants, the bled steam is sometimes superheated to a temperature injurious to materials in process and desuperheaters have to be used.

Exchange of power between central stations and industrial plants is gaining in favor, and a number of plants report that it is working to the advantage of both parties. Also, in several instances, central stations have built plants to supply both steam for process and power to one or a number of industrial plants in the vicinity. In such cases non-condensing high-back-pressure turbines are employed, exhausting to the process-steam mains and floating on the line, electrically in parallel with the system; the electrical load supplied by them being governed by the process-steam demand.

Several manufacturers of pulverized-coal equipment report greatly increased sales of unit pulverizers to industrial plants, but pulverized coal has not had the field to itself for stokers have been installed in a great many of the new or remodeled plants. In several cities where smoke-abatement activities prevail, the smaller stokers have found a ready market in the smaller industrial plants.

More attention is also being given to the better utilization of waste heat either in boilers or in air preheaters. About one-third of the newer plants use air preheaters, and a somewhat smaller number employ economizers.

Much greater attention is being given to feedwater treatment, and technical supervision is more generally employed. The trend in methods used is similar to that in central-station practice. In some instances evaporators have been installed to take a considerable portion of the boiler steam, returning the condensate to the feed lines and delivering pure vapor to process.

In conclusion, it may be stated that the present status of research, design, and performance forecasts an even greater progress in 1930.

Grateful acknowledgment is here made to the many engineers whose assistance has made this report possible.

F. M. GIBSON, *Chairman.*

Psychology of Smoke and Fuel Waste in Ceramic Plants

By VICTOR J. AZBE,[1] ST. LOUIS, MO.

IN A BOILER plant fuel efficiency is very poor when it is only 50 per cent and very good when 75 per cent. In a lime plant efficiency can readily be as low as 25 per cent and can also be as high as 60 per cent. In boiler plants the range represents a loss of fuel for the poorer conditions of one-third, while in a lime plant the fuel loss in the poorest compared with the best is considerably over half. Plants can readily be found that have a ratio of $2^1/_2$ lb. of lime per pound of coal, as well as others, rather scarce ones, that have a ratio of 5 lb. per pound of coal.

In boiler plants the efficiency leans in the upper, while in lime plants it leans decidedly in the lower, poorer direction. Efficient lime plants are extremely scarce, and what is even sadder is that definite information as to what is a good lime plant also is scarce. If one needs a book on lime-kiln design, about the only way to get it is to write one. To make conditions still worse there are only few qualified to write such a book, even if they had inclinations to do so.

In the brick industry, conditions, while not quite so backward as in the lime industry, are also very bad. Some time ago the author was escorting a German fuel technologist on his visit to St. Louis. While passing a brick plant this European had a hearty laugh at the crowned updraft kilns used, which he said had not been in use in Germany for 25 years. In this industry also the information is meager, and only some years back there was hardly anything available on the subject of fuel economy. There is a lack of definiteness, great disagreement, individuals are groping in the dark. What the industry needs very badly is trained combustion engineers with an inkling of ceramic knowledge, fully as much as ceramists.

The lime and brick industries are a great deal alike. Ordinarily they have small plants, kilns with small outputs, high-temperature operation, smoke, desirability for long flame, old kilns, older ideas, rule-of-thumb control, about the same type of managing and operating personnel, many old timers that have grown up with the business, and almost complete lack of agreement. What are the reasons for all this?

Not so long ago a survey of equipment and operating methods was made in a number of plants. It was thought that it would be interesting also if the survey would cover in a confidential manner the characteristics and opinions of the managing and operating personnel that such information might possibly reveal reasons for apparent backwardness of the industry. The opinion was that no plant could be better than the managing personnel. Abatement of smoke was the underlying reason for this survey. Following are excerpts from this report with comments:

The first plant visited, a brick plant, uses both updraft and downdraft kilns. The company has in the past made efforts in trying to overcome smoke. This is evidenced by the installation of steam jets in most of the updraft and some downdraft kilns. The jets have not been a success, and are no longer used to a great extent. Mr. —, the superintendent, a man of a naturally cheerful nature, seems to be a pessimist by development. Whereas in the past new ideas have been welcomed, due to past failures, new ideas are now regarded with suspicion.

Commenting on the foregoing, one must say that this is a sad case, a good man, willing to make improvements, disappointed and turned a pessimist. And this is a common case, too; new ideas are tried and fail. Sometimes the ideas are half-baked, sometimes the trial is not fair, and all the expense and hope go for nothing. Ordinarily, after all is over, it is not even known why the experiment was a failure. Ordinarily, no tests are made; the whole is gone into with crudeness; the so-necessary combustion engineer does not come anywhere near the experiment.

There are many lime plants pouring smoke of the very highest density into the sky. In these cases to try to use steam jets to reduce smoke would be absolutely senseless. The purpose of the jet naturally is to create turbulence and bring the oxygen in contact with the carbon, but in these cases there is no excess oxygen, and no amount of turbulence will reduce smoke. Injection of steam will make it even worse, because its equivalent amount of air will be displaced. The experiment fails if instruments of proper nature are not used by a man knowing how to use them to obtain definitely either positive or negative information about the value of what is being tried. Human senses are too imperfect; even the physician invariably uses a fever thermometer, although he can tell without it if the patient has fever. He wants the exact answer and so should the manager of a plant.

Mr. —, the head of the second plant visited, is a large, explosive sort of person, who will probably fill a stranger with terror or awe. The plant has not changed materially in 35 years. Mr. — seems to be the essence of the older heads of the brick industry, i.e., very much under the power of mental inertia, resisting progress.

A most unfortunate case, because the described gentleman is large and explosive, and one of the older heads. Due to his physique, glibness of tongue, and set mind, he can do a great deal of harm resisting progress, not only in his own plant, but due to his influence on others also. The only way he can be successfully controverted is with reliable figures, unquestionable information and proofs, and if they are lacking, as is the case in these two industries, then one has an uphill battle indeed.

In the third plant visited, the man in charge is a most interesting and accommodating person. He keeps up with modern ideas, is of an experimental turn of mind, and does a lot of experimental work himself. He is interested in smoke abatement, willing to accept suggestions, would like to observe tests, and would probably be willing to have tests conducted in his plant.

Fortunately, there are some like this. They are hungry for information, but do not know where to get it. They want to improve their plant and do improve it, but if conditions are not favorable, they are likely to revert into the type of the first man described, into pessimists by development. Due to the very meager amount of technical information presented in an understandable practical form, such men have their difficulties. They are willing, but do not understand the language of the college-trained technician to whom they are complemental.

Some brick plants make a dark colored brick, which is used as an excuse to make smoke. There are great arguments over this required smoke—the ethics of making it in a populous district, etc. This remark is made to make the next paragraph more lucid.

The next plant visited was a brick plant. Mr. —, the superintendent, is a very genial old man and quite friendly. The ware

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

manufactured does not require smoke, but they produce it just the same. One kiln is equipped with a stoker, but it has been used with rather indifferent success. The stoker is used only when absolutely necessary for production.

Evidently another experiment went wrong. Some miscalculation possibly on the part of the salesman anxious to sell the stoker, but knowing nothing about the very peculiar conditions under which it was to be used. All of it turned into more than wasted effort. One such wrong installation can delay progress in a score of plants.

The superintendent of — plant was a rather cynical man with a sense of humor. He, like a successful poker player, seemed to conceal behind an easygoing exterior an inquiring and alert mind. He was willing to talk about the industry and his own plant. Thought the tunnel kiln was the thing for any plant having enough business to use the output. He seemed dissatisfied with the conditions of the industry in general. "The brick industry has been too long in its infancy," was his comment. As to need of smoke to obtain color, he was truly skeptical. He believed the same results could be obtained in other ways, as for example by the use of manganese or similar material.

Both the brick and the lime industry were in their infancy too long a time. Most of the plants of both are now still in the stages where the steel industry was 50 years ago. Very little progress has been made and there are very few progressive workers. Inefficiency is subsidized by high prices. Recent keen competition and price cutting caused a tremendous howl on the part of those having plants with high production costs. They now are between the devil and the deep sea. When prices were good, they had money to modernize their plants, but did not do so because they were making money anyway. Now they have to sell their output for a low price and do not make any profit, and do not have any money available to make the necessary plant changes. Keen competition, elimination of the weak, and survival of the fittest do more for conservation of natural resources than all the commissions, conferences, etc., combined. The average small industrialist seldom foresees; and how can he, when the sea he sails is foggy, his charts are incorrect, the actions of his compass are awry—even the trained navigator, the engineer, becomes confused. The brick and lime industries need charting, need sharp-cut, practical information based upon technical data, and more than anything else they need men having experience, perspective, and the necessary information and imagination to do this charting. The whole must be boiled down from the state of partially sound and mostly unsound confusion into as few as possible of evident facts and be clearly presented. That this is necessary is further proved by the characteristics of the next man, who is quite an influential factor.

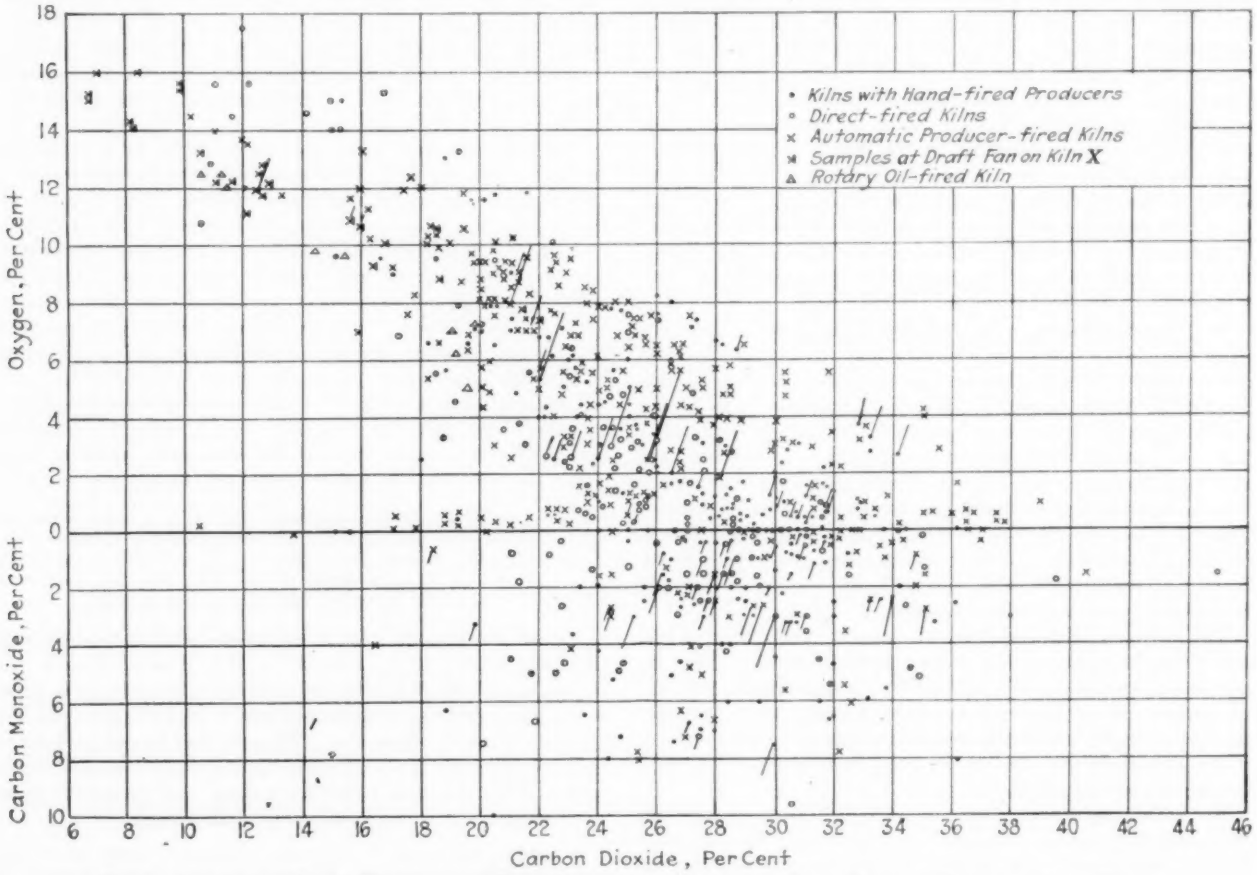


FIG. 1 ANALYSIS OF SEVEN HUNDRED AND FIFTY GAS SAMPLES OBTAINED FROM VARIOUS TYPES OF LIME KILNS

Mr. —, the superintendent of the — plant, was quite cordial during the visit. He is very suave and seemingly open minded, but, at the same time, his opinions are very deeply rooted and it is hard to conceive of him as being moved by anything but most striking and well-founded facts. This company enjoys the reputation of being hard boiled, not willing to give information and very willing to acquire information. Experience shows that this reputation has not been justified.

To cope with such a man, to convert him, needs not only technical training, it needs a knowledge of human nature, it requires the matching of wits, and a rather clear practical insight—as clear as his own anyway.

Mr. —, superintendent of the — street plant, was about of the

same type and even more hospitable. He admitted that on the question of smoke there was basis for complaint in some quarters, and that public opinion must be regarded in this respect. He raised the question of priority rights of people who have moved into the vicinity after the existence of the plant to complain of the plant as a nuisance.

But there is no question of moving or padlocking the plant, and not even a question of pollution of immediate vicinity. First, a brick plant and a lime plant can be operated smokelessly. Second, when they do smoke, the air is polluted more or less for miles. In fact, aerial investigations show pollution extends under certain conditions for hundreds of miles, which with increased air travel and desirability for maximum visibility will become quite a problem.

Some time ago two tests were run on a certain kiln. Both were under observation of qualified engineers, but, during the first, the firemen fired as they thought proper, heavy infrequent firing, of course, and the smoke was almost continuous. For the second test, the firemen used were entirely green men, who never had fired a kiln before. They were told by the engineers how often and how much to fire, light and often, of course, and there was practically no smoke. This proves that the practical experience of many years' duration is valueless if a man is set in his ideas. The fact that the quality of ware, length of burn, etc., during the second test with green men, were at least as good as during the first test, proves that what is needed in a plant is one man thoroughly trained, who knows a great deal; the rest need be men who know little, but are willing to do what they are told.

At plant —, the head burner was interviewed, a man of some twenty years' experience. One kiln is equipped with — type firebox and they are not at all satisfied with it. From the conversation it was gathered that this was largely due to personal dislike of the inventor, rather than the performance of the box.

Prejudice enters into the problem on all hands. It is likely that there may have been considerable reason for the dislike of the inventor. It will be noted that the investigator quoted here gained a desirable reception in every plant, even in those of bad repute for sociability. It is not at all hard to get along if one is charitably minded and approaches the operating force in a sympathetic manner. They are all doing the best they know how; they may have been doing it in this way for years and years, and is it not too much to expect to see them change immediately without even being convinced of the right or wrong of the matter? A knowledge of human nature and ability to get along and make friends is as important to the engineer striving for improvement of plant operations as is technical and practical knowledge. This is particularly so in industries with stagnated development where the processes are regulated by empirical rather than scientific rules.

Mr. —, owner and operator of this plant is one of the brick men who most stanchly ridicules the idea of the need for smoke in brick making. In his 35 or more years' experience he claims that the idea of the need for smoke is a new one. In his plant the coal is burned in an external box, instead of inside the kiln as is the usual practice. Nevertheless, smoke is produced in the same volume as with the older type box. He is willing to try to reduce smoke—anxious in fact. He wants to install steam jets and will do that if he can find any one to advise their use. He asks every visitor to his plant for their opinion on the use of steam jets.

A long radiant flame is very desirable in both the lime and brick industries. The lime and brick turned out are then more uniformly burned. With short flame, temperatures are not so uniform in the burning zone, and overburning and underburning result. A smoky flame ordinarily means delayed combustion; it is for this reason that there is so much disagreement on the subject of smoke. Further, if there is more coal distilling its volatile matter than there is air to cope with it, then no matter what kind of box or steam jets are used, there will be smoke. If the boxes would be arranged with baffles for downdraft burning with insufficient air, and an additional delayed supply of secondary air, then volatilization would have to be constant instead of intermittent, and the flame long without smoke. Then why is recirculation of kiln gases not tried for lengthening out of flame? It is used in the lime industry, and there long flame while desirable is not as important as in the brick industry.

The — Company has been interested in smoke abatement and improvement of conditions for years. Efforts during the past year have been particularly productive. The smoke record during that period has changed from worst to best among the ceramic industry in St. Louis.

If one can do it, why not the others? The reason is that experience obtained in one plant only is not the best of experience. It is culled experience, the findings of many plants applied, that is best. Why is one plant smokeless and efficient and the other not? It is because the interchange of ideas is too slow. Each man proceeds in very much his own way, and often is too conceited to admit that the other man has something worth trying. If he is unusually open minded he may try it, but not with any great degree of enthusiasm. He, because the thing is not his own idea, may even be subconsciously prejudiced against it; he may be unfair, and so the whole may be assured of becoming a failure.

The — Company presents a strange mixture of ancient and modern ideas. They have well-trained ceramic experts, a well-equipped laboratory, and yet are very slow to change existing practices. Mr. — gives the impression of a cold wet blanket at first acquaintance. Future relationship may or may not cause a change in this feeling. He is willing for new ideas to be tried, with the ever-present idea of low cost in view.

Not only small concerns, but large also, can be backward. In some respects it is hard to blame them. Suggestions for improvement too often are of too revolutionary a nature, too costly for a large plant. Assurances for their success are not always without some doubt, and with competition to keep prices low, there is little wonder if the manager turns into a wet blanket, particularly if he had some sad experiences in the past.

In a large plant there are many kilns, and supervision becomes divided, and often this supervision leans first toward upkeep of mechanical equipment, then to quality of the output and the various attending troubles. There is little left over to be applied to combustion, and almost none to develop new firing methods.

There are still other types of men with characteristics not so far exemplified. There is the one who does not care, who completely lacks any sense of duty, any pride of accomplishment, who is physically as well as mentally lazy. Fortunately, however, he very seldom finds his way into any of the governing positions. Then there is the still worse type, who is positively unscrupulous, untruthful. Under certain conditions he may deliberately maneuver to wreck an experiment. He is even more scarce. Then the visionary who if left alone would bankrupt the company with "improvements" without any reason to them. Then the man who after gathering in a little knowledge thinks he knows it all, not realizing that a little knowledge may be very dangerous. Then the obstinate one who never changes his mind; whom nothing can change from his preconceived idea. And finally, there is the man who never makes a mistake.

All of these characteristics are reflected in the conditions of plants of these two industries and in many others as well. Lime kilns are found with fire on top and brick kilns with gas burning in the chimney. At one time the fuel is burned with a large amount of excess air escaping at temperatures from 1000 to 2000 deg. and at another carbon monoxide in waste gas as high as 10 per cent. In a lime plant the desirable gas analysis is 32

per cent. In a lime plant the desirable gas analysis is 32 per cent CO_2, and 2 per cent oxygen and no carbon monoxide. The accompanying chart shows 750 gas analyses. How many are of the desirable characteristics? Most of them range all over the field from 10 per cent oxygen to 10 per cent carbon monoxide. This is a terrific waste even for a boiler plant, and much more so for an industrial furnace with a high temperature level. The brick man does not realize that the heat generated by the carbon burned to CO during the latter stages of the burn is all wasted, and while he may need reducing conditions, he does not need 7 per cent of CO in the waste gas as was found on tests run under actual operating conditions. When the stack temperature is 1500 deg., there is a tremendous difference if the stack gases contain 10 per cent oxygen or only 4 or 5 per cent, still both conditions are oxidizing.

To make improvements in the lime- and clay-ware manufacturing plants, due to the indefiniteness of the general technique, is very much of a battle and requires considerable persistency. If it were always possible to rip down the old plant and build a new tunnel kiln for clay ware or large kilns equipped with modern automatic gas producers for lime, then the whole would be comparatively simple, because with such equipment not so much is dependent upon the operating force. But this is not always possible, even though in most cases it would be advisable. Ordinarily, one has to fight forward inch by inch. For example, the change to more frequent light firing, simple as it sounds, is ordinarily not easy to accomplish. The whole depends upon how strong the convictions of the superintendent are and how determined he is.

One plant, completely converted by construction of new kilns, increased its product per ton of coal from a ratio of 2.8/1 to 4.3/1. The investment made was very great, even somewhat speculative, but final results proved that it was more than justified. In another plant complete change may not be possible, and one has to do the best with the existing equipment. The accompanying chart shows the results of such a plant. It represents very interestingly the performance from the fuel ratio and capacity standpoints over the period of five years, extending over the régime of three superintendents. Under the first man, the improvement began. The beginning, however, was slow; he was not entirely sure of himself. The second man was practically green, but very open minded. He realized that he knew but very little, but was willing to act upon good advice as well as willing to try out his own ideas. This man was about the best of all the author has encountered in a hundred plants or so. He was without education, but had the will which often is more important. He brought the plant to a high state of efficiency, if the type of equipment he had is taken in consideration. During the third régime, plant performance slumped again. The superintendent, although a good man, was just simply misplaced.

The peak in efficiency is 2.85 tons of lime per ton of coal. At the beginning of the four-year battle, it was only 2.1 to 1; this means a reduction in fuel consumption of 25 per cent. In addition, the output was increased 38 per cent, and there was but little difference in appearance of the plant before and after. The whole can be laid to the operating men. Of course, efficiency at 2.85 to 1 was still far from good. If the management had decided to build a new plant, the ratio would have been $4^1/_2$ to 1 with the somewhat low grade of coal burned, but the management did not see fit to do so and therefore the superintendent proceeded to do the best in his power with what he had.

FSP-52-4

Economics and Design of Water-Cooled Furnaces

By J. S. BENNETT[1] and P. N. OBERHOLTZER,[2] PHILADELPHIA, PA.

Experience of one manufacturer in designing and operating water-cooled furnaces is given in this paper. It deals particularly with water-cooled furnaces for underfeed stokers, and a study of results and conditions of several actual installations is presented. Determination of the proper amount of water cooling is an economic rather than a combustion problem. Ordinarily the most critical portion of the furnace is the rear wall, so that if but one wall is to be cooled, it should be the rear wall. Sometimes the sizes of stokers used in older-type plants are limited by the ability of the furnace to withstand temperatures resulting from higher combustion rates. In many such cases a moderate investment in water cooling will give a substantial increase in steam output.

THE object of water cooling is to permit the generation of cheaper steam than with the ordinary type of furnace. Specifically, the objects of water cooling for underfeed-stoker furnaces are:

1 To prevent fused ash from adhering to the furnace walls.
2 To prevent mechanical abrasion along the fire line.
3 To prevent failure of brickwork in the upper part of the furnace due to erosion and undercutting of the furnace construction along the firing line.
4 To reduce boiler outage.
5 To reduce furnace maintenance.
6 To reduce stoker maintenance.
7 To reduce slagging on the boiler tubes.
8 To simplify ash removal.
9 To permit increased capacity.
10 To increase overall efficiency
11 To permit the burning of poorer coal.

The value of water cooling is well accepted, but the selection of the proper amount of water cooling for a specific case is more difficult. By presenting a study of the surrounding conditions of actual installations, it is hoped that some of the debatable points will be cleared up and that the engineer selecting the furnace design for his specific case will be aided in reaching a decision.

The first water-cooled furnaces of the modern type were installed at the Saginaw River plant of the Consumers Power Company, Zilwaukee, Michigan, and at the Hell Gate plant of the United Electric Light and Power Company, New York. These two water-cooled furnaces were put into service at approximately the same time. At Hellgate, side walls only were protected with water cooling. Nearly half of the water-cooled surface was covered over by refractory blocks, as it was believed that the exposure of a greater amount of water tubes would have an unfavorable effect on combustion, and the danger of blow-torch action on the tubes and mechanical erosion along the fire line was recognized.

The results obtained at Hell Gate were remarkable, but the full possibilities of water cooling were not realized because the most critical points, the rear wall and the side wall at the overfeed sections of the stokers, were not protected. This naturally hindered the ash-disposal system, and in turn had an unfavorable reaction on the whole fuel bed.

The application of this same refractory-protected side wall

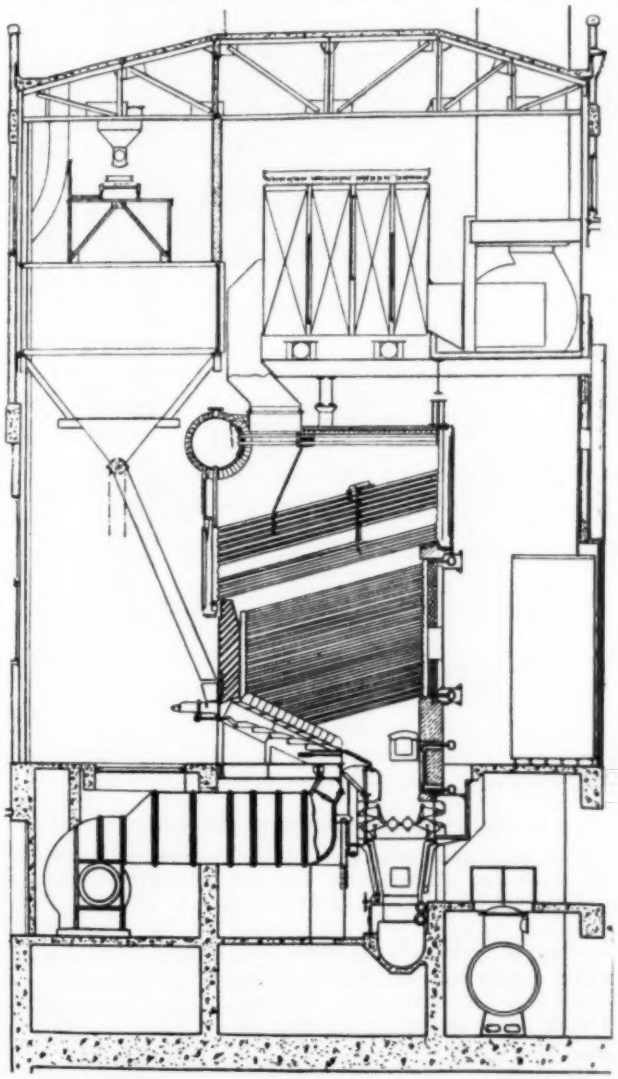


FIG. 1 WATER-COOLED WALLS AT THE SAGINAW RIVER STATION OF THE CONSUMERS POWER COMPANY

at other plants with lower grades of coal has demonstrated the need of cooler surfaces along the fire line to prevent adhesions of slag. Also, it has been found that the brickwork washes away very quickly with certain grades of coal, especially when using preheated air. In one case, the brickwork was washed away down to the tubes in seven days of operation.

The Saginaw River installation (Fig. 1) differed from the

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Hell Gate installation in that the side walls were formed by sloping tubes instead of vertical tubes as at Hell Gate. The protection along the fire line at Zilwaukee was better than at Hell Gate, but the side walls along the lower end of the stoker were not protected by water cooling, and the rear water wall did not run down far enough into the ashpit, so that the small sections of brickwork between the stoker aprons and the bottom of the water wall were worn away, forming a shelf which obstructed the flow of ashes to the rolls.

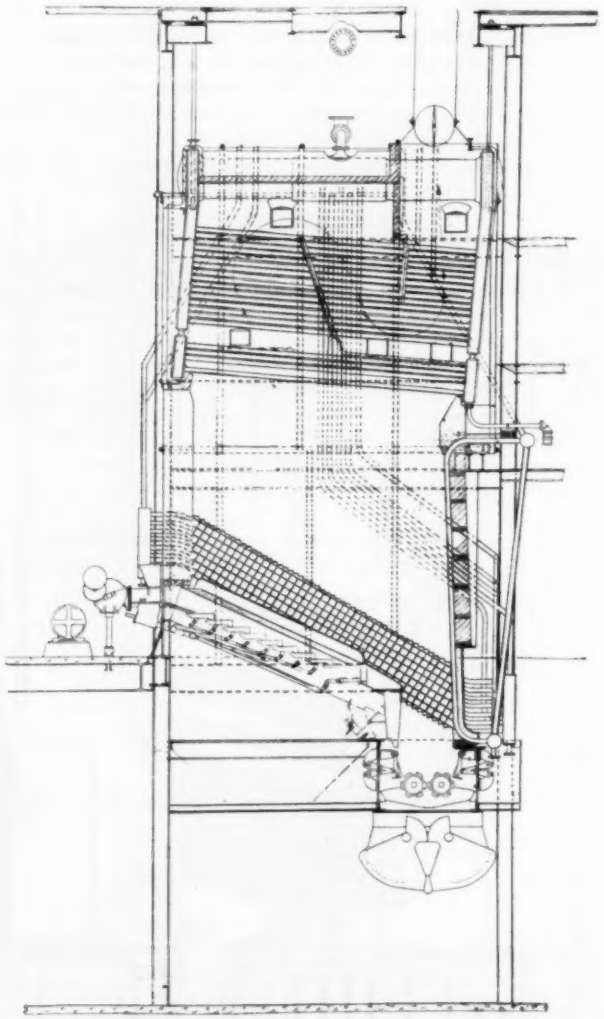


FIG. 2 JUDICIOUS USE OF WATER COOLING AT THE NEW TERMINAL PLANT OF THE CONSOLIDATED GAS AND ELECTRIC COMPANY OF BALTIMORE

At Zilwaukee, the tubes along the fire line were protected by cast-iron blocks shrunk on the tubes. It was demonstrated at this plant that the combustion of the fuel was not affected adversely by this cool surface along the fuel line, as had been feared.

The first Hell Gate units operated at 338 per cent of boiler ratings for 25 hr. with three-hour peaks of 450 per cent and a ¾ hour peak of 540 per cent. The Saginaw installation, with a somewhat greater ratio of stoker to boiler surface, operated repeatedly for prolonged periods at boiler ratings in excess of 450 per cent continuously. There was 485 per cent of boiler rating maintained for 48 hr. and 505 per cent of rating for 24 hr.

Experience at these two plants and with many other of the earlier water walls has shown that the best construction along the fire line consists of a smooth bare-metal surface. A number of means have been employed to provide this. The best method is undoubtedly the renewable metal block. This construction permits the replacement of blocks which have been scoured away mechanically by the abrasive action of the fuel bed, and also prevents flame impingement against small spots on the tubes, which, with the bare tube, has proved very objectionable.

In contrast to the older water-wall installation at Hell Gate, there is the modern unit installed at the new Terminal Heating plant of the Consolidated Gas and Electric Company of Baltimore. (See Fig. 2.) The side and rear water walls along the fire line are faced with renewable, cast-iron blocks. The side walls above the fire line, where the duty is light, are brick construction. The overfeed section of the stoker and the ends of the ashpit are protected by a band of water cooling, extending down into the ashpit.

This unit went into service on Dec. 22, 1928, and was operated at boiler ratings as high as 400 per cent with frequent prolonged peaks of 360 per cent, and at a normal rating of 250 per cent. This boiler was kept on the line until the early part of May of this year. The unit was then taken off the line and presented the appearance shown in Figs. 3 and 4. Absolutely no repairs of any kind were necessary to the brickwork, water walls or stoker.

DETERMINATION OF THE PROPER AMOUNT OF WATER COOLING

So far as combustion is concerned, there is no possibility of too much water cooling being applied in the furnace with underfeed-stoker firing. It is obvious, however, that the application of more than the economic amount of water cooling cannot be justified, so that the determination of the proper amount is an economic problem, rather than a combustion problem.

Table 1 presents a number of installations which will be discussed in this paper, and an effort will be made to analyze the conditions and reasons for the selection of the amount of water cooling employed in each instance. In presenting these figures, for simplicity it has been assumed that all makes of bent-tube boilers cost the same per boiler horsepower and that the same applies to straight-tube boilers. The ratio of the costs of the two types of boilers has been taken as 7 to 9, as it has been found the price of straight-tube boilers generally exceeds that

TABLE 1 DATA ON WATER-WALL COSTS FOR SIX PLANTS

Name of plant	Boiler hp.	Water wall cost in per cent of boiler cost	Maximum boiler rating, per cent	Tons of coal burned per year	Water wall price in per cent of annual coal cost	Walls water cooled	Coal	Remarks
Terminal Heating Co. . .	1261	37.5	400	14,400	19.6	Side and rear	Good	Note (1)
Brown Corporation	518	33.0	300	8,300	12.4	Side front and rear	Very bad	Note (2)
Brown Corporation	764	76.0	350	13,400	20.0	Side, front and rear	Very bad	Note (2)
St. Croix Paper Co.	745	28.3	310	13,000	7.5	Rear	Good	
Ball Bros. Co.	507	58.0	300	10,500	13.0	Side and rear	Poor	
Ohio Public Service Co. (Lorain)	1464	60.0	500	35,000	13.3	Side, front arch and rear	Poor	Note (3)

Data based on same price of coal, and on same boiler prices, with bent-tube boilers priced at 78 per cent of horizontal-tube boilers.

(1) Heating plant has low load factor. Coal consumption expected to increase.

(2) Furnace maintenance would be very high with this coal especially on front walls, without water cooling.

(3) Coal consumption expected to increase.

of bent-tube boilers approximately in this ratio. It is realized that this ratio may not hold with some manufacturers and that in some cases the bent-tube boiler of one type might cost more per horsepower than a straight-tube of another type. Any one who wishes may apply a corrective factor to the figures given to conform with his own views. The water-wall costs used are the present selling prices.

The annual coal costs used are based upon the same price of coal delivered at the plant for all installations shown. The figure taken is a representative one, and is the same for all installations, for simplicity in comparing ratios of costs.

In order to illustrate the application of the data in Table 1, two typical cases will be taken, as follows:

Case I Completely Water-Cooled Furnace:

Water walls costing 35 per cent of annual coal cost
Fixed charges 15 per cent
Coal at $4.25 per ton

Water wall fixed charges = $4.25 × 0.35 × 0.15 =	$0.22 per ton
Efficiency without water walls = 80 per cent	
Efficiency with water walls = 83 per cent	
Coal saving per ton = $\frac{0.03}{0.80} \times \$4.25$ =	$0.16 per ton
Reduction in furnace maintenance with water walls	0.15 per ton
Reduction in stoker maintenance with water walls	0.03 per ton
Total saving per ton of coal	$0.34

A saving of $0.34 per ton of coal burned results from the use of a completely water-cooled furnace, which is $0.12 per ton more than the fixed charges on the water wall.



FIG. 3 INTERIOR OF FURNACE AT THE TERMINAL PLANT AFTER FOUR MONTHS' RUN

Case II Partially Water-Cooled Furnace:

Water walls costing 14 per cent of annual coal cost
Fixed charges, 15 per cent
Coal at $4.25 per ton

Water-wall fixed charges = $4.25 × 0.14 × 0.15 =	$0.09 per ton
Efficiency without water walls = 80 per cent	
Efficiency with water walls = 82.5 per cent	
Coal saving per ton = $\frac{0.025}{0.80} \times \$4.25$ =	$0.13 per ton
Reduction in furnace maintenance with partial water cooling	0.10 per ton
Reduction in stoker maintenance with partial water cooling	0.01 per ton
Total saving per ton of coal	$0.24

A saving of $0.24 per ton of coal burned results from the use

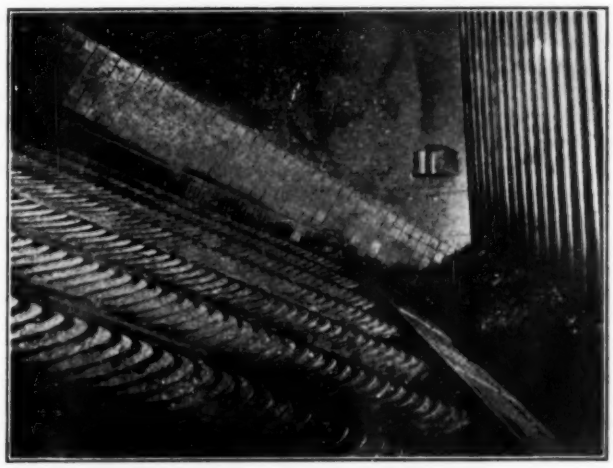


FIG. 4 SIDE VIEW OF FURNACE SHOWN IN FIG. 2 WITH SLAG REMOVED TO SHOW EXCELLENT CONDITION OF WALLS AND EXPANSION JOINT IN CENTER OF SIDE WALL

of a partially water-cooled furnace. This saving is $0.15 per ton more than the fixed charges on the water wall.

Complete Water Cooling

Total saving	$0.34 per ton
Fixed charges	0.22 per ton
Saving over and above fixed charges	$0.12 per ton

Partial Water Cooling

Total saving	$0.24 per ton
Fixed charges	0.09 per ton
Saving over and above fixed charges	$0.15 per ton

These figures show that a partially water-cooled furnace results in a net saving per ton of coal burned of 3 cents per ton greater than the saving secured by using completely water-cooled walls.

This comparison can be made directly because the capacity of the completely water-cooled furnace is practically the same as that only partially cooled.

However, in comparing any water-cooled furnace with a refractory furnace, it must be borne in mind that the extra capacity secured from the water-cooled furnace may be sufficient to justify the cost of the water walls without considering any of the other factors involved.

Ordinarily the most critical portion of the furnace is the rear wall. If one wall only is to be cooled, the rear wall should be the one. The rear water wall is always used with rotary ash-discharge stokers, as the successful operation of this type depends upon the ash being fed by gravity to the rolls. Obviously fused ash sticking to the rear wall upsets this condition.

The benefits of water cooling of rear walls with the dump type of stoker are beginning to be realized. A typical installation of this type is that at the St. Croix Paper Company (Fig. 5), Woodland, Maine, burning New River coal with a heating value of 14,000 B.t.u. per lb. as fired. Fig. 6 shows the furnace interior after the unit had been in operation for about one year. A

careful check of the operating results for the four months immediately preceding the application of the water wall, and during the four succeeding months showed that the operating efficiency jumped from 80.8 per cent to 81.8 per cent with a normal boiler rating of 210 per cent with peaks of 310 per cent of one hour's duration four or five times daily.

Eighteen per cent make-up water was used with 2.32 grains of total solids per gallon with 17 parts of hardness per million.

It was found that the capacity of the stoker was increased, and that the maintenance of the rear wall which had been a substantial item was practically eliminated. Taking the annual coal consumption as shown in the table on this installation at 13,000 tons at $5 per ton, it is found that the annual coal bill would be $65,000. One per cent increase in efficiency is equivalent to a coal saving of $65{,}000 \times 1/80.8 = \$810$ per year annual coal saving. Assuming that the annual rear-wall maintenance was $200, it is found that $\$810 + \$200 = \$1010$, which capitalized at 20 per cent will justify an expenditure for water

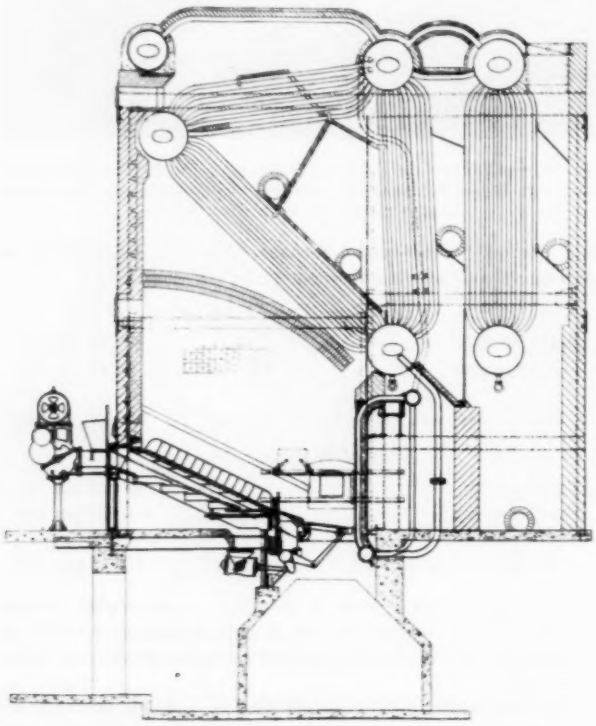


FIG. 5 A MODERATE AMOUNT OF WATER COOLING WILL GIVE A HANDSOME RETURN

walls of $5050. This, of course, does not allow credit for the reduced stoker maintenance which inevitably results, nor for the increased boiler capacity available, generally the most important factor.

In Table 1 it is seen that the water-wall cost is 28.3 per cent of the boiler cost and $7\frac{1}{2}$ per cent of the annual coal cost.

REVAMPING EXISTING BOILERS

Not infrequently the size of the stokers used in older plants was limited by the inability of the furnace to withstand the temperatures resulting from higher combustion rates. In many cases of this kind a moderate investment will give a substantial increase in steam output.

For example, the La Tuque plant of the Brown Corporation needed more steam from existing boilers set in battery. This condition was met by installing underfeed stokers and water walls on all four sides. (See Fig. 7.) A radiant superheater above the rear water walls provided the added superheat needed. For two years these units have been operated normally at 250 per cent of boiler rating, and have a capacity of 300 per cent continuously. Twenty-five per cent make-up water caused no difficulty.

The operating efficiencies of these units, excluding economizers



FIG. 6 REAR WALL OF ST. CROIX PAPER COMPANY'S FURNACE AFTER ONE YEAR'S SERVICE

or preheaters, is approximately 75 per cent, which in view of the high exit-gas temperatures shown in the summary of tests is a very remarkable result. Dominion slack coal with an ash-fusing temperature between 2000 and 2100 fahr. and containing 32 per cent iron oxide is used.

As a result of experience for two years with the units described, the new unit, Fig. 8, going into service will operate at an average boiler rating of 300 per cent (50 per cent more than the old units), using the same type of furnace construction as on the older boilers, but with the addition of extended water cooling in the front to permit the use of a longer stoker.

The cost figures as shown in Table 1 on the old and new installations are very interesting. It will be seen that in the old installation, the cost of the water walls is only 33 per cent of the boiler cost and 12.4 per cent of the annual coal bill, while with the new installation these figures are increased to 76 per cent and 20 per cent, respectively. A number of factors influence this, as will be noted by referring to the illustrations, Figs. 7 and 8. The amount of water cooling used in Fig. 7 of the old installation is reduced, because a radiant superheater forms part of the rear wall. The new installation with a bent-tube type of boiler requires more extensive water cooling of the front wall, but this is offset by the lower cost per unit horsepower which is used for the bent-tube boiler.

The new furnace undoubtedly will show a higher efficiency

than the one in Fig. 7, as the furnace designer was not limited by existing conditions.

The selection of water walls by engineers of large public utilities is a decidedly different problem from that confronting the designer of an industrial plant. The return on the investment in the case of the industrial plant must ordinarily be higher, as the management of the industrial plant is continually seeking funds for improvement in the manufacturing portion of their works, which, of course, is the very heart of their existence. In the central station, however, the business of manufacturing steam is one of the main items, and can be given much more attention by the highly skilled designers and carefully trained operators. Under equal conditions, therefore, it is naturally expected to find more extensive water cooling used in the large central stations, especially as the steam output per square foot of floor area generally exceeds that in the industrial plants by a large margin.

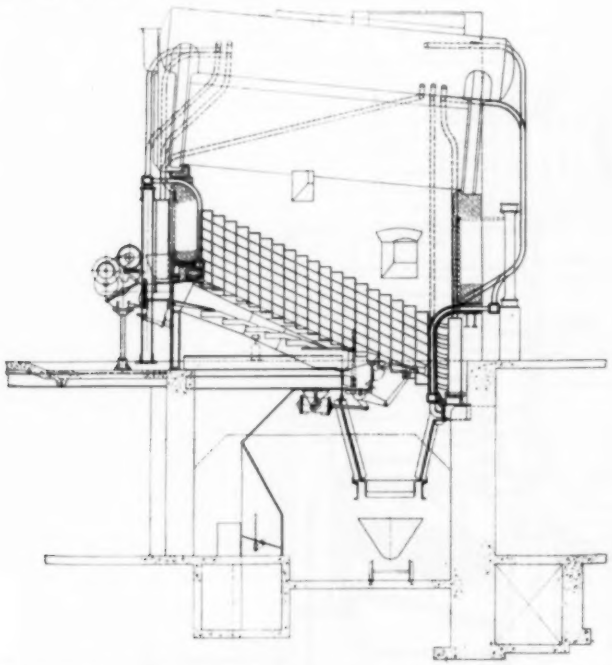


FIG. 7 WATER COOLING ON EXISTING FURNACE IN COMBINATION WITH THE USE OF STOKERS MET STEAM REQUIREMENTS WITH LOW CAPITAL INVESTMENT

The large public-utility operators were the first to recognize the economic advantages of water cooling and have devoted much time and thought to the best disposition of the component parts of the furnace. Their need for a large amount of steam delivered from a restricted floor space has led them to the use of combustion units of large size, which in many cases have outgrown the old conventional furnaces. This has been met by extended water-cooling surfaces, as for example, at the Lorain plant of the Ohio Public Service Company (Fig. 9) designed by the H. L. Doherty Company to burn coal with the following typical analysis:

Ohio Bituminous Coal

	Per cent
Moisture	4.0
Volatile matter	32.0
Fixed carbon	54.0
Ash	10.0
Sulphur, separately determined	3.0
Ash-fusing temperature	2150 deg.

Both the rear and front walls have been extended by water cooling, the rear wall forming a slag screen for the boiler, while the sides have been protected by a wall of economic height.

Table 1 shows a water-wall cost amounting to 60 per cent of the boiler cost, and with the water-wall cost 13.3 per cent of the

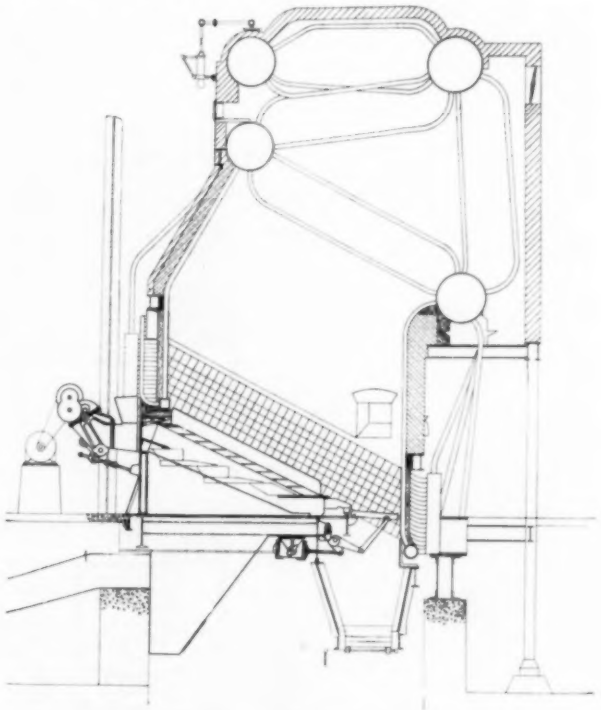


FIG. 8 FURNACE SETTING AS USED AT THE BROWN CORPORATION, LA TUQUE, CANADA

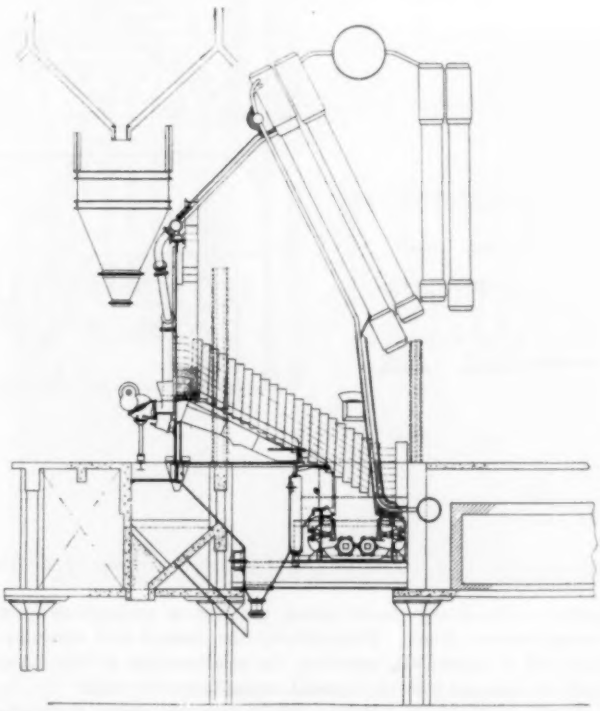


FIG. 9 WATER-COOLED FURNACE WITH EXTENDED WALLS TO PERMIT THE USE OF A LONGER STOKER

annual fuel bill. The cost of the water wall as compared to the boiler is high, but this may be explained by the fact that the water wall really becomes part of the boiler, forming a slag screen. The boiler construction requires a water-cooled extended-furnace front. Undoubtedly the annual coal consumption will be increased, reducing the relative cost of the water wall as compared to the annual expenditure for coal.

In the ordinary installation, the brickwork above the water walls is subjected to very little punishment, but where the service may be more severe, the refractory construction can be protected by air cooling, which also permits recovering the hot air in the upper portion of the boiler room and securing the benefit of combustion with preheated air at a very moderate extra cost.

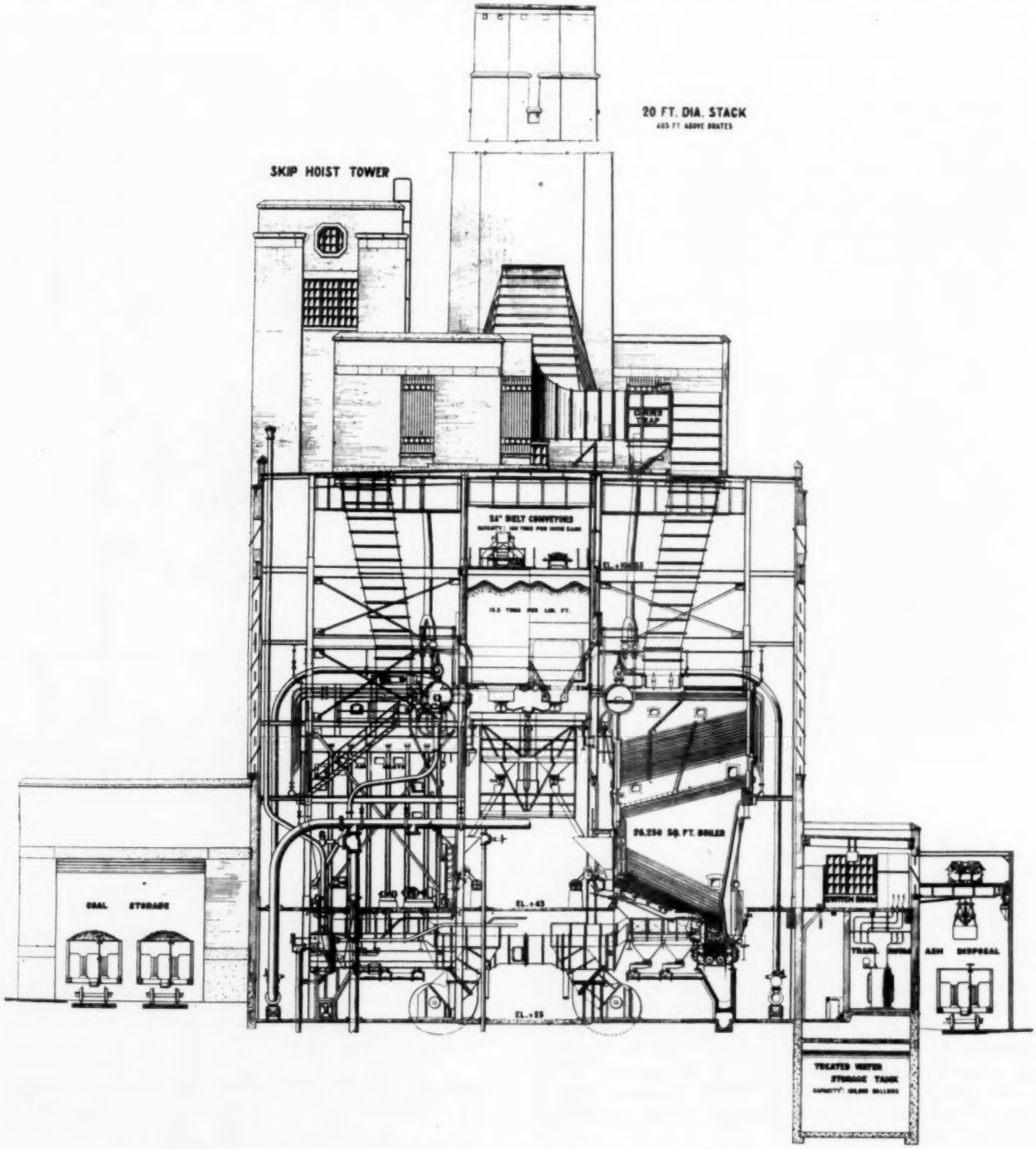


FIG. 10 AIR COOLING IN COMBINATION WITH WATER WALLS CAN BE USED TO ADVANTAGE

The new West Philadelphia heating plant of the Pennsylvania Railroad, designed by the United Engineers and Constructors, will employ this method. Air before delivery to the stokers (Fig. 10) is drawn through the side and front walls by forced-

draft fans placed in the basement. The boiler will be 2623 hp., fired by 15-retort 45-tuyère stokers, with electro-hydraulic drive, having a speed range of 20 to 1. Each boiler is designed for a steam output of from 34,500 lb. per hour to 276,000 lb. per hour continuously; a range in steam output of 8 to 1, with peaks substantially higher.

Water Walls for Preheated Air

The effect of preheated air on combustion is to produce a fuel bed with characteristics similar to that with a fuel of somewhat inferior grade. The almost universal use of water walls with preheated air therefore has been the result of the same experience which made water walls popular with low-grade coal. Contrary to the belief of some engineers, the harmful effect of preheated air on the furnace walls is not due so much to the

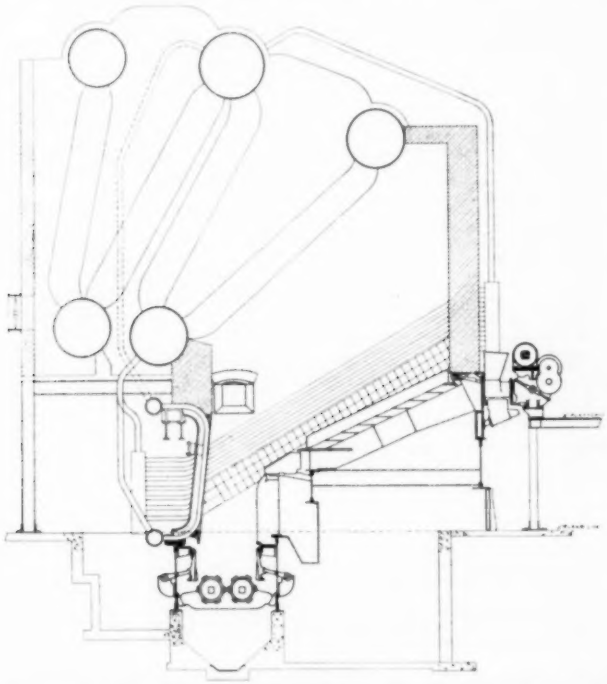


Fig. 11 Water-Cooled Furnace Designed to Burn Indiana Coal With High Preheat

increase in furnace temperature, which may be very slight, as to the greater fusion of the ash. For this reason, portions of the brickwork in the furnace not subjected to erosion and washing away by the fluxed ash are not affected adversely. A novel installation employing this principle and designed to burn Indiana coal with high preheat (Fig. 11) operates at the Noblesville plant of the Ball Brothers Company. These furnaces (Fig. 12) use air for combustion heated to a temperature of 400 deg. The fifth vein Indiana coal used has the following typical characteristics:

B.t.u. per pound as fired	11,220
Moisture	10.30 per cent
Volatile matter	36.30 per cent
Fixed carbon	41.65 per cent
Ash	11.75 per cent
Sulphur	4.20 per cent

The stokers are placed completely above the normal floor line, as the boilers replaced were fired by pulverized coal with manual removal of ashes from the furnace. The building used for powdered-coal firing permitted the installation of modern rotary ash-discharge stokers with more capacity than the unit-mill equipment removed.

Where high preheat in combination with high wind-box pressure

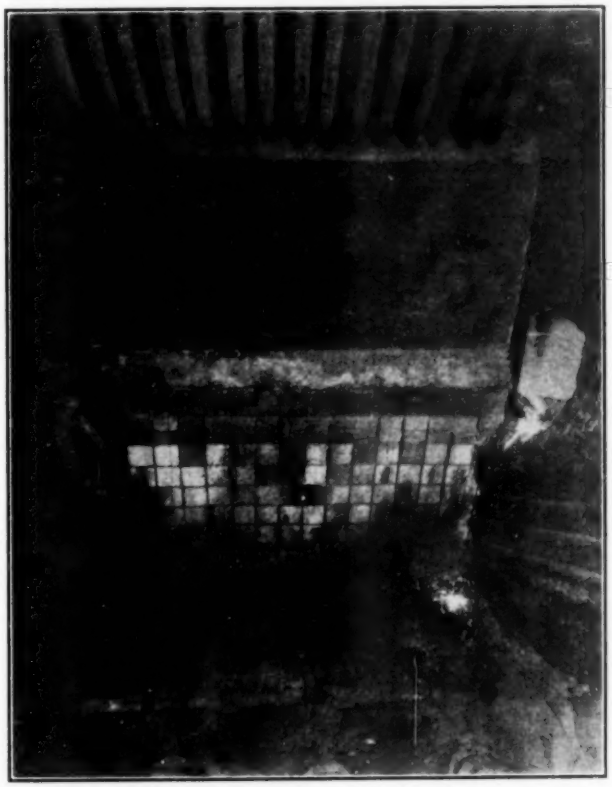


Fig. 12 Interior of the Furnace Shown in Fig. 11



Fig. 13 Effect of Slag Running Down an Insulated Wall

is employed, extra care must be taken to prevent the fused ash on the walls from melting and running down into the fuel bed, interfering with the normal functions of the stoker.

When the furnaces for the high-pressure boilers at the Edgar

plant of the Boston Edison Company were designed, this possibility was realized, but the fear that the large amount of water cooling might interfere with the combustion of the gaseous products led to the installation of insulated water-wall blocks above the fire line with bare metal blocks along the fuel bed. The design was so arranged that this could be modified, if necessary. When subjected to high furnace temperatures, however, it was found that these insulated walls did not give the protection required due to the running of the slag (Fig. 13) and its attachment to the fuel bed. When part of these insulated water-wall blocks were replaced by the bare metal type, the operation was much improved. A still further extension of this principle to the side and rear walls of the two latest units shows still greater improvement.

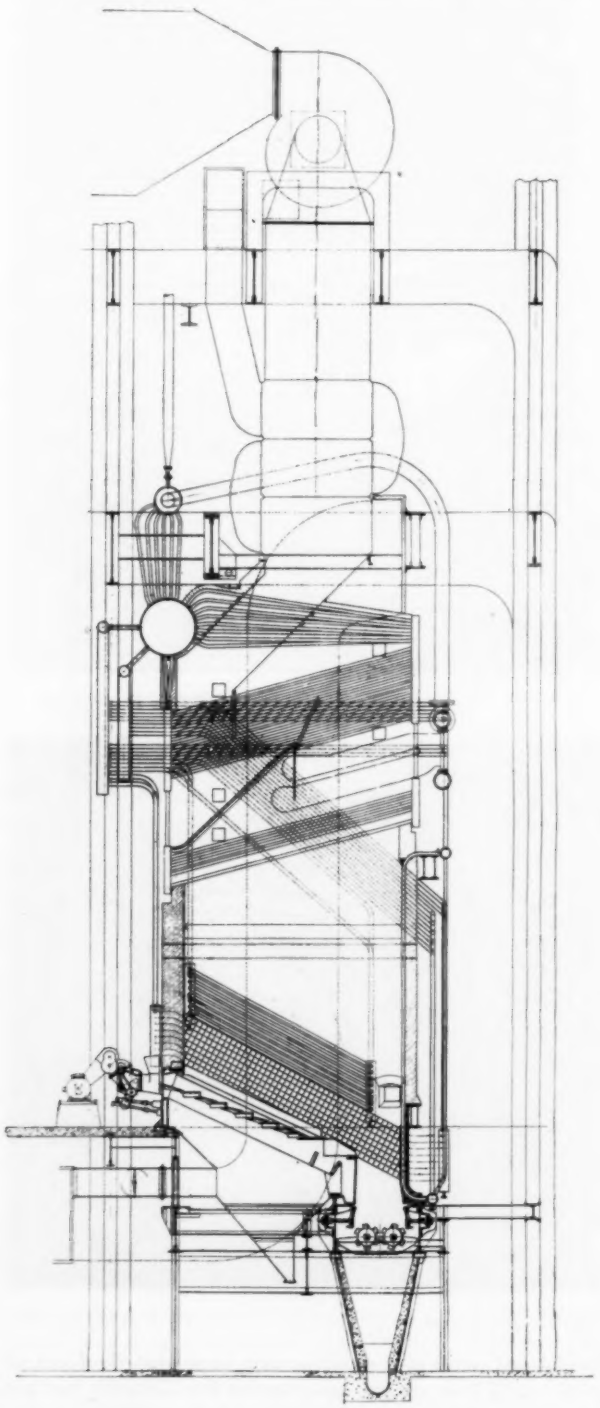


FIG. 14 FURNACE DESIGNED FOR USE OF HIGHLY PREHEATED AIR—UPPER PORTION OF SIDE WALL FORMS SLAG DRIP

TABLE 2 SUMMARY OF TESTS AT BROWN CORPORATION, LA TUQUE PLANT, QUEBEC, CANADA

Boiler				
Type	Heine			
Number of passes	3 horizontal			
Tubes high	16			
Setting height, low end	10 ft. 5 in.			
Furnace width	8 ft. 9 in.			
Furnace vol., cu. ft.	1280			
Furnace vol. per rated boiler hp., cu. ft.	2.47			
Boiler heating surface, sq. ft.	5180			
Side-wall heating surface, sq. ft.	128			
Rear-wall heating surface, sq. ft.	50			
Front-wall heating surface, sq. ft.	43			
Ratio of grate surface to boiler surface	2.39			
Superheater. Radiant type				
Economizer. None				
Air Preheater. None				
Stoker				
Type	Taylor underfeed			
Size	5 retort, 33 tuyère			
Grate area, sq. ft.	123.7			
Test Data				
Boiler No.	5	5	5	5
Number of test	1	2	3	4
Date, October, 1928	23	24	25	26
Duration, hr.	10	10	10	4
Per cent rating (boiler and superheater)	211	234	253	291
Actual steam per hr., thousand lb.	31,746	35,426	38,676	44,300
Combined efficiency (boiler, superheater, and water walls), per cent.	79.1	83.5	76.9	77.0
Steam press., lb. per sq. in. gage	135.7	139.4	137.7	136.2
Factor of evaporation (boiler only)	1.18	1.18	1.17	1.17
Coal burned, lb. per hr. per sq. ft. proj. grate (as fired)	30.6	30.5	35.0	39.1
Coal burned, lb. per hr. per retort	757	755	860	963
B.t.u. liberated per cu. ft. of furnace, vol. per hr.	35,900	37,900	44,600	51,000
Temperatures				
Total temperature of steam, deg. fahr.	402.4	399.9	393.0	389.0
Degrees of superheat, deg. fahr.	44.0	40.0	34.0	30.0
Temperature of gases leaving boiler, deg. fahr.	678.0	692.7	719.0	763.0
Temperature of air at fan, deg. fahr.	70.8	69.6	68.3	65.5
Temperature H_2O entering boiler, deg. fahr.	105.1	108.0	111.0	109.1
Air Pressure. Windbox, in. water	3.24	2.89	3.30	3.83
Draft. Over fire, in. water	0.11	0.10	0.06	0.14
Coal and Ash Analysis. Fuel, Dominion slack. Iron oxide (average all tests), per cent	31.95			
Moisture in coal, per cent	8.30	6.50	3.48	3.15
Volatile matter in coal, per cent	33.26	33.23	33.22	33.88
Fixed carbon in coal, per cent	47.45	51.92	54.31	55.41
Ash in coal, per cent	10.99	8.35	8.99	7.56
Sulphur (determined separately), per cent	2.52	2.86	3.12	2.78
Comb. in refuse, per cent	11.51	10.27	14.79	21.76
B.t.u. per lb. coal as fired	12,129	12,849	13,220	13,484
Fusing temperature of ash, deg. fahr.	2090 average			

DESIGNING WALLS FOR SEVERE CONDITIONS

In anticipation of this same condition, at the new Westkraft station in Berlin, Germany, the side walls are being constructed with eight bare tubes (Fig. 14), overhanging the lower section of metal-block covered tubes on the side walls, thereby forming a drip ledge to prevent slag interference with the normal action of the fuel bed. The rear wall is entirely water cooled, with cast-iron protection on the tubes in the lower portion.

This installation, which is the most important power-plant development in Europe, will consist of eight Borsig boilers, with a horsepower of 2580, operating at 410 lb. per sq. in. pressure and with 800 deg. fahr. steam. The combustion-air temperature will be approximately 500 deg. The Taylor stokers firing these boilers will be 20 retorts wide, and 49 tuyères in length, having a total projected area of 713 sq. ft. These units, based on American practice, have a continuous steam output of approximately 500,000 lb. of steam per hour. The coal to

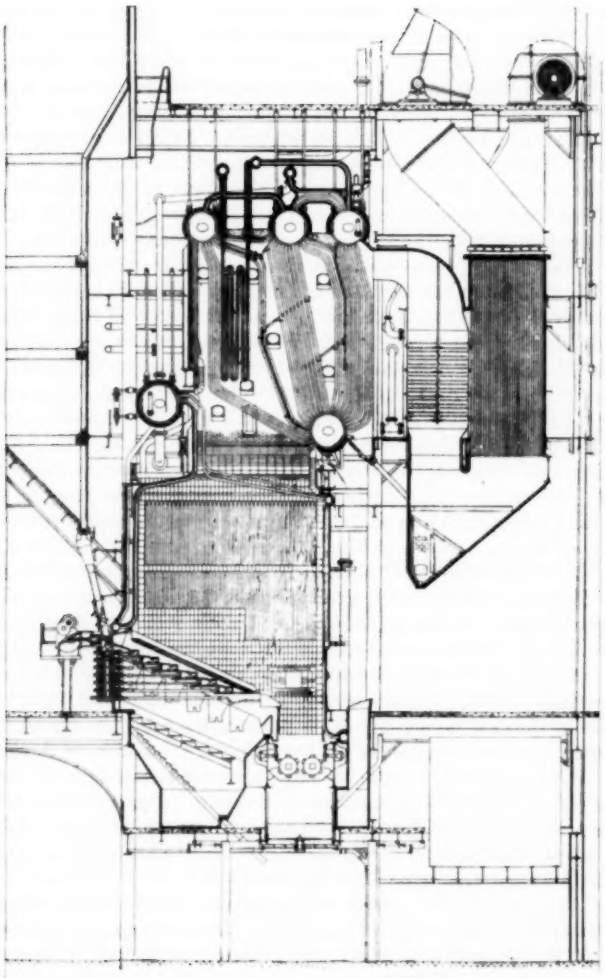


FIG. 15 COMPLETELY WATER-COOLED FURNACE DESIGNED FOR HIGH DUTY AND PREHEATED AIR

be used will vary in heating value from 11,000 to 14,000 B.t.u. per lb. as fired.

For extremely high duty, using preheated air, the maximum amount of water cooling should be used. Such an installation (Fig. 15) is operating at the Delray station of the Detroit Edison Company, where all four sides are water cooled. The sides and rear have bare metallic blocks, carried well above the fire line, while the front wall is completely covered by insulated blocks.

At the time this is written, the full capacity of these units has not been utilized, but they are expected to generate approximately half a million pounds of steam per hour.

FUTURE DEVELOPMENTS

Future developments will undoubtedly include more extensive use of the extended furnace, such as is shown in Fig. 16. Furnaces of greater width than the boiler have also been employed. An increasing use of these ideas may be expected, resulting in furnaces much larger than the old conventional ones.

Discussion

JAMES W. ARMOUR.[3] In the application of water walls one of the main points to decide is to what extent the tubes should be covered. The authors' statement is that tubes should be covered to prevent abrasion and impingement. If this is true, is it to be assumed that, in cases where the rear wall is inwardly inclined where impingement must exist the entire height of the wall, the entire wall should be covered? If this must be done, then the heat transfer would of course be reduced, requiring an increased amount of water surface to obtain the cooling effect desired.

This immediately brings up the question as to whether impingement of the flame on the water wall is detrimental to the functioning of the wall or to the material of which the wall is composed. Impingement of flame on water walls having natural circulation may cause localized formation of steam pockets and possible burning out of the wall tubes at that point. It would seem as if forced circulation would be the answer to this problem, for if operation of the wall is such that impingement of the flame against it is not detrimental, then the tubes can be left bare and full advantage of the absorption surface obtained. The wall would be more efficient and less costly.

The authors gave comparisons of completely and partly cooled walls to bring out the economical point of coverage. From their figures the main saving is from increased efficiency. With a wall of 14 per cent coverage, efficiency is only $^1/_2$ of 1 per cent less than

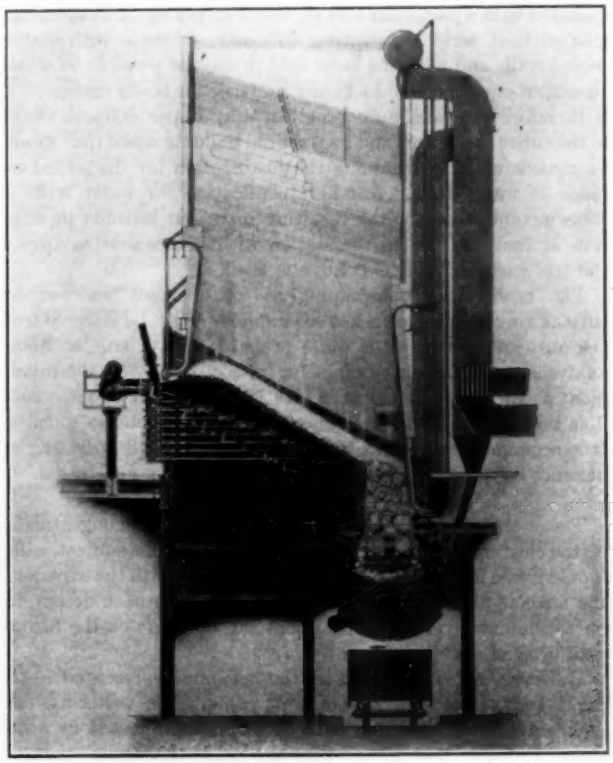


FIG. 16 PREVIOUS LIMITATIONS OF FURNACE DIMENSIONS NO LONGER EXIST IN THIS MODERN UNIT WITH EXTENDED FRONT AND REAR WALLS

with 35 per cent coverage. Would it not be difficult to measure this difference in efficiency or to determine that it was primarily due to variation in amount of coverage?

Interest in design of the wall is primarily as to how the amount of wall necessary can be predetermined. It would be interesting to know what basis can be used to predetermine the increased efficiency due to the addition of varying amounts of wall coverage.

The authors have made no mention of the amount of heat

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liberation in the furnace and its effect on determining the economic effect of the wall coverage. It would seem as if this was of prime importance and should be the main basis for determining the amount of wall coverage to be used in any particular case. High heat liberation means high furnace temperature, consequently high refractory maintenance, high stoker maintenance, and poor operation due to formation of clinkers in the fuel bed. It would seem that a proper balance between heat liberation in the furnace, characteristics of the fuel, together with heat absorbing capacity of the water wall, would be the basis for determining the amount of water cooling to be installed.

S. B. Flagg.[4] The authors refer to the "extra capacity secured from the water-cooled furnace" which they state may be sufficient alone to justify the cost of water walls.

There are two viewpoints from which this statement may be considered. If a major portion of the furnace is water cooled, there will be a credit against the cost of the water wall due to the replacement of expensive refractory construction otherwise required, and a further credit because the water-wall surface will, at times of maximum output, absorb heat at several times the average maximum rate of absorption for the boiler surface proper.

The other viewpoint is, in most cases, the more important. Without water cooling, a boiler of given heating surface may be operated with a particular fuel at, say, 7 lb. per sq. ft. evaporation rate without serious operating difficulty, whereas with water-cooled walls and with the same fuel it may be possible to attain an output equivalent to 14 lb. per sq. ft. of the boiler surface. It is therefore the maximum practical output per dollar invested in the entire boiler unit and its required building space that should be considered in determining the justification for the added expense of water walls. Another justification for water walls in some instances may be the resulting increased latitude in selection of fuels for use in the station with consequent savings in the fuel cost per million B.t.u. absorbed.

The problem of determining to what extent water-cooled surface may be justified is not so simple as might be inferred from the authors' paper. It is quite probable that a true economic analysis would show a diminishing rate of return on the investment as the water-cooled area was increased, but in most cases it is believed that the decisions are predicated not so much on true economic analyses as upon the judgment and operating experience of those responsible for steam or energy generation.

Theodore Maynz.[5] The authors are to be congratulated on the clear manner in which they have treated this subject, which is of extreme interest to all power engineers. With the experience the authors and their company have had in furnace design and operation data, very little comment can be made on the furnace wall designs that are shown.

However, a few remarks on the economics presented in this paper may be pertinent and of some value. The writer has been among the early engineers who have been interested in water-cooled furnaces, and many years ago made the remark that "refractories had no place in modern furnaces," purely from an engineering and thermodynamic standpoint. However, as we must pay for everything that we get, there are many things that are purchased, which, though inherently good, are economically bad. This is frequently, too frequently, the case with water-cooled furnaces. This applies to industrial plant boilers more often than to central stations, but also at times to them. That large boilers can be designed to operate economically and with low maintenance with refractory furnaces has been proved by the 3060-hp. boilers of the Cleveland Electric Illuminating Company, where the experience gained at the Lake Shore Station led to its duplication at Avon Station for both the first and succeeding units. The coal used is similar to that shown in the paper; in fact, the analysis shown for Ohio Coal is superior to that usually obtained, the ash content being more often 12 to 13 per cent and the fusing point as low as under 2000 deg. The writer takes exception to the coal cost in case 1 and case 2, believing that $4.25 is too high and therefore too favorable. A cost of $3.50 per ton would more nearly equal the true cost of coal at the stoker in the Eastern and Middle Western part of this country. This would change the coal savings to $0.13 per ton, the cost of the water walls to 0.42½ per cent, the total savings to $0.31, and the net savings to $0.09 per ton of coal. This is a net return of 6 per cent on the cost of the walls, after fixed charges. This might be permissible for a public utility, but it is doubted if any board of directors of an industrial plant would allow such an expenditure with so little net return if these facts were brought to their attention. Using the coal cost as shown in the paper, which is very much more favorable, the net return on the investment is only 8 per cent.

With partially cooled walls and $4.25 coal a return is obtained that begins to look attractive, provided there is any money left after paying for the essentials such as boilers, stokers, pumps, and building. With this coal cost the net return is about 25 per cent, and with $3.50 coal and water walls costing 17 per cent of the annual coal cost (which would be the case under this cheaper fuel), the coal savings are $0.11 per ton, the total savings $0.32 per ton, net savings $0.10 per ton, and the net return on the investment slightly under 17 per cent, not quite so attractive. The writer is using the same figures as given in the paper on the increase in boiler efficiency, which seem reasonable, and for reduction in maintenance, which seem rather high, when considering a well-designed refractory installation.

In considering the cost of water walls one cannot consider only the cost of the wall. When higher pressures are used (over 250 lb.) the cost of blowoff valves, drains, piping, and insulation brings the cost of this equipment considerably higher than one is led to believe from the bare-wall price quotation. As an example, in a certain installation the cost of the walls (side and rear) cost $8 per rated b.hp., $23 per square foot of wall (total water wall was 3½ per cent of the boiler heating surface), the blowoff and drain valves cost almost 15 per cent of the water-wall cost. These boilers are of medium high pressure and cost $2.30 per square foot, delivered and erected, so the high cost of water-wall construction can be compared. In respect to increased capacity and efficiency, in a new installation, the increase in capacity and efficiency can be obtained cheaper in additional boiler-heating surface than in water-cooled walls. At $2.50 per square foot of heating surface 10,000 B.t.u. can be obtained, or 4000 B.t.u. per dollar. In water-cooled walls at $25 per square foot about 30,000 B.t.u. is obtained, or a 1200 B.t.u. per dollar, a ratio of 3½ to 1 in favor of boiler-heating surface.

However, when the manufacturers bring their costs down to a reasonable figure, then a real economic use of water walls will result, and industrial boiler plants can justify their use by economics, rather than their present methods of lavishly following central-station practice, whether these practices are applicable to their own conditions or not.

The use of clinker grinders on stokers necessitates the use of water-cooled surfaces on the rear wall, otherwise operation is very bad, but, then, it is very doubtful if a clinker grinder pays for its cost on medium- and small-sized boilers (5000 to 12,000 sq. ft.) even with $4 coal and good load factors, exclusive of the extra cost in the required water-cooled bridge wall.

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[5] Consulting Steam-Electric Engineer, Cleveland, Ohio. Mem. A.S.M.E.

I. E. Moultrop.[6] The most important factors justifying the use of water-cooled furnace walls are undoubtedly greater furnace capacity as mentioned by the authors, and, secondly, greater availability. These two factors are important because of their relation to the question of the unit cost of construction. Since the greatest problem today is to find ways to lower the unit cost of construction, any apparatus that helps in this direction should be developed and used to the fullest extent possible. The one drawback to the even greater use of water-cooled furnace walls is the high cost of such apparatus. There does not appear to be any inherent reason why these costs should not be reduced materially, and undoubtedly the manufacturers' intensive study along this line will be greatly appreciated by all users and potential users of water-cooled furnace walls.

From information available at this time it would appear that the authors are crediting the water-cooled furnace walls with a greater increase in overall boiler and furnace efficiency than has been obtained in practice. If an increase of 3 per cent in efficiency is possible, as stated in case 1, all would like to know how it can be accomplished. A careful study of the problem fails to show that such an increase in efficiency is "in the wood," or probably it should be said, is "in the water wall."

In the case of the side furnace walls well above the fuel line they are subjected to terrific punishment in furnaces operated at large heat releases per cubic foot. This experience and other similar experiences would indicate that for best results the water cooling should be carried well above the fuel line.

Furnace maintenance costs from modern central stations indicate that the figured saving in furnace maintenance given in the paper is too high. For a furnace burning $12^1/_2$ tons of coal per hour and operating at a load factor of 40 per cent a saving of 15 cents per ton for furnace maintenance would amount to $8750 per year. The maintenance on such a furnace would probably not approach this figure, and it would therefore appear rather optimistic to credit the water walls with any such saving.

Experience so far has shown that greater capacity and availability are the real factors justifying the installation of water-cooled furnace walls.

Edwin Smiley.[7] The paper seems to be a step forward in the field of commercial engineering, as applied to underfeed-stoker installations. Referring to the two cases cited—namely, a completely water-cooled furnace and a partially water-cooled furnace—the difference is all the more marked if higher fixed charges are assumed. It is felt that a 15 per cent fixed charge allows for too little depreciation on water-cooled walls. This figure of course can be more accurately determined with further use of water-cooling.

The authors state that "extra capacity secured from the water-cooled furnace may be sufficient to justify the cost of the water walls without considering any of the other factors involved." This is especially true regarding the revamping of existing installations. However, in the case of new installations, extra boiler capacity can be obtained more economically by a wider or longer boiler for an underfeed-stoker installation. A square foot of water wall usually costs more than 1 b.hp.

Based on the measured evaporation from water walls it usually takes 1 sq. ft. of water wall to generate 1 b.hp. This raises the cost of boiler capacity from water walls to more than the cost in boiler heating surface. Boiler capacity from boiler-heating surface has the added advantage of more positive circulation and lower tube maintenance due to scale trouble.

It is felt that the design of the new West Philadelphia Heating Plant of the Pennsylvania Railroad, as described in this paper, is representative of the best modern practice.

The type of sectionally supported air-cooled walls used on such an installation will usually show a higher return on the investment than water walls, with possibly the following exceptions:

a Lower rear walls with rotary ash-discharge stokers
b Sections of the grate line exposed to abrasion
c Whenever furnace temperatures exceed fusion point of ash
d Extremely high-duty installations with preheated air.

Incidentally, air-cooled walls on a job of this size cost very little more than a solid firebrick setting.

As to "future developments on extended furnaces on underfeed-stoker installations," the use of sectionally supported walls is to be recommended for walls overhanging the fuel bed. On the rear wall of such installations, the fire face of the tile should be protected by some form of water cooling.

It will probably be found more convenient, on front walls, sectionally to support and insulate these, rather than to use water cooling or air cooling.

The tendency to use comparatively thin tile walls on high boiler settings because they are sectionally supported is certain to cause excessive radiation losses at higher ratings. Provision should therefore be made for air cooling the tile to recover this radiation loss wherever possible.

H. W. Leitch.[8] In reading over the enumerated lists of advantages of water-cooled underfeed-stoker furnaces it seems that the set-up is not such as sufficiently to emphasize the chief reason for water walls. The most valuable benefit of such an installation is to permit increased capacity. This increased capacity is obtained in three ways: The first way is by the reduction of boiler outage, enumerated as No. 4 on the list. If there was a reduction in boiler outage of, say, 10 per cent by the use of water walls, this would be the same as a 10 per cent increase in capacity in a plant where there is a multiplicity of boilers. The second is the ability to operate at higher rates of combustion. This of course is a direct effect of increasing the available capacity. The third is by the addition of very effective heating surface. This is the most valuable quality of water-wall surface in that it is directly exposed to radiant heat, and so introduces a surface of high rate of heat absorption.

With these facts before him, the writer does not agree with the comparison set-up which shows a saving by installing partial water cooling over complete water cooling. The authors state that the comparison can be made directly because the capacity of completely water-cooled furnaces is practically the same as only partially cooled. This is certainly not true where the furnace is fired by powdered fuel, and in a well-designed completely water-cooled furnace it is not true for stoker firing. It is apparently true for the latter in situations in which it has not been necessary to operate at very high ratings. Therefore, in a set-up as given, there should be added a fixed charge on capacity which would change the conclusions given by the authors and perhaps in some cases give entirely different results.

Again, it is questionable whether there should be any substantial gains in efficiency credited to any kind of water cooling alone—certainly not more than 1 per cent.

Another advantage which should be credited to water walls which is particularly applicable to powdered-fuel furnaces is the reduction in the tendency of slag formation in the tubes. This works in favor of the 100 per cent cold furnace.

The paper states that if only one wall is to be cooled it should be the rear wall. This is true, but for a further reason that on ac-

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[7] King Refractories Co., Philadelphia, Pa. Assoc-Mem. A.S.M.E.

[8] General Superintendent, Power Plants, United Electric Light and Power Company, New York, N. Y. Mem. A.S.M.E.

count of the inclination of the stoker the absorption of the rear wall is greater than that of either side wall. This in turn protects the stoker and reduces maintenance. The side walls, however, are desirable to eliminate refractory maintenance.

It should be pointed out that the design of such furnaces has not yet been so standardized that errors are not apt to creep in. There are certain essentials that must be borne in mind in order to have a successful installation. Some of these are as follows:

1 A circulation arrangement to insure adequate hydraulic head as insurance against overheating of metal
2 Rapid circulation and absence of inclination of tubes in a direction favorable to application of heat to steam-bound surfaces
3 Simplicity and symmetry, eliminating numerous special tube sizes and other parts
4 Freedom from parts exposed to furnace heat not in homogeneous contact with water tubes. The ability to withstand severe impingement depends largely on this feature of construction
5 Adequate provision for expansion without injury to the system or external walls
6 Adaptability to various boiler designs without excessive design or development costs.

A. G. Christie.[9] The subject of water cooling of furnace walls is one in which the writer has had an intensive interest for many years. In 1922 he advocated the more extended use of water-cooling for the walls of powdered-coal furnaces, but encountered considerable opposition in many quarters. This resulted in the preparation of a report on the influence of water-cooled walls on furnace performance which was published in condensed form as a paper in *Power* in 1923. This paper aroused much interest and discussion. While the general ideas of this early paper were sound, its arguments would undoubtedly now stand some modifications in view of the more extended study of the subject since that time, the more complete data available, and the extended experience in the use of water-cooled walls that has accumulated.

The paper is a valuable contribution to the knowledge of water walls, for it indicates a more complete and more accurate appraisal of the economic effects of water cooling of furnace walls.

The authors state that "so far as combustion is concerned, there is no possibility of too much water cooling being applied in the furnace with underfeed-stoker firing." Some engineers have feared that the cooling effect of the water-cooled furnace walls would seriously affect combustion with low-volatile coals. This statement should allay those fears.

One may ask what is the effect of the water-cooled walls on the fuel bed of an underfeed stoker. The authors show that the water-cooled rear wall will decrease the slagging and sticking of the ash in the ashpit. But let us assume that a high-volatile, high-sulphur coal with low-fusing ash is being burned. The furnace will be filled with luminous flame which tends to shield the fuel bed from the walls. Will the presence of water-cooled walls in such a large furnace prevent the ash from slagging on the main body of the fuel bed?

Mr. Craig in his paper indicates that any gains in boiler thermal efficiency with water-cooled walls are due to the use of a smaller quantity of air together with a higher percentage of CO_2 than would be permissible with refractory walls. These considerations of decreased air and higher CO_2 must therefore have been assumed in this paper in estimating the increased efficiencies resulting from the use of water-cooled walls.

The authors of this paper also indicate that water-cooled side walls are necessary with preheated air. Engineers would be interested in information concerning the allowable degrees of preheat that can be used on stokers with various classes of coal when different areas of water-cooled walls are used.

The authors base their economic gains on the improvements in boiler efficiencies that result from the use of water-cooled furnace walls. The paper indicates that boiler capacity is increased by the use of water walls. These walls are expensive when compared to costs of equal boiler surfaces on a square foot basis. However, from a plant designer's point of view, one of the most important economic gains from the use of water walls is the saving in total investment in the boiler plant for a given total steam output. Such increased steam outputs can often be secured through the use of extensive water-wall surfaces at a lower total cost than by additional boilers, stokers, coal bunkers, and boiler-house space. This is a consideration that cannot be overlooked.

The authors show many boiler designs with water cooling along the fire line of the stokers. Experience has shown that this greatly improves operation and maintenance. Its use should be extended with both underfeed and chain-grate stokers, and even in the small sizes an improved performance would result therefrom.

Authors' Closure

It is very gratifying to have this paper supplemented by such carefully prepared discussion. Several speakers have questioned the increased efficiency attributed to water walls. The difference in efficiency mentioned in the paper is that obtained under operating conditions. When tests are run with refractory

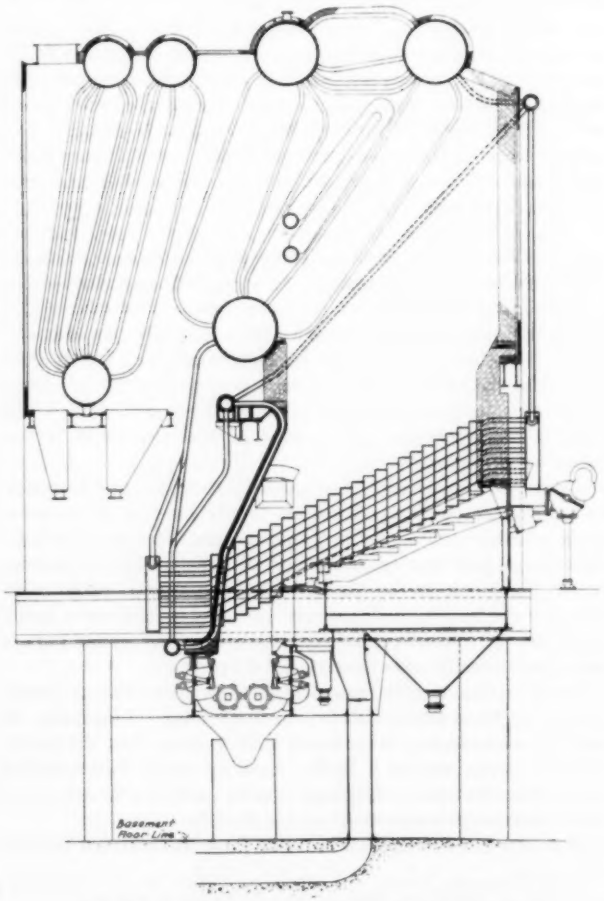


Fig. 17 Boiler Equipped With Water Walls

[9] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

furnaces, it is common practice to raise the CO_2 much higher than could be used in daily operation; with water walls, test conditions can be maintained in every-day operation. The combustible lost in the refuse with refractory furnaces is greater than with water walls, especially with inferior grades of coals, due to the interference with the proper combustion conditions produced by growth of slag on the side and rear walls. This condition does not occur with a properly designed water-cooled furnace.

Fifteen per cent CO_2 is commonly maintained in water-cooled stoker furnaces, while much lower figures are usual with entirely refractory-lined furnaces. It is a well-known fact that water walls show less increase in efficiency with good coal than with poorer grades, yet N. E. Funk (A.S.M.E. Trans., 1926, page 373) has reported an increase of from 2 to 3 per cent with water walls burning excellent eastern coals.

H. W. Lietch (A.S.M.E. Trans., 1926, page 129, fig. 8) shows tests on three boilers, Nos. 51, 63, and 73 with partial water cooling, and boiler No. 13 with no water cooling. Up to 150,000 lb. evaporation per hour, the three water-cooled furnaces show an average of 3 per cent higher efficiency even though the boilers are substantially smaller and the stokers have 25 per cent less grate area than that in the refractory furnace.

An interesting test was conducted on two boilers arranged as shown in Fig. 17 equipped with American Engineering Company's water walls and fired by Taylor stokers. The two units

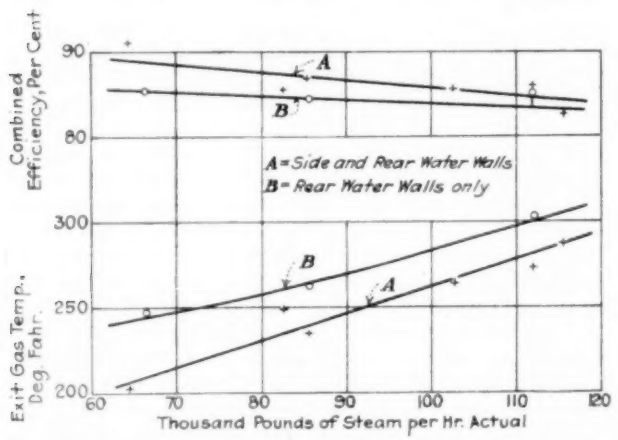


FIG. 18 PERFORMANCE GRAPHS SHOWING EFFECT OF VARYING WATER-WALL SURFACE

(1014-hp. Stirling boiler; 11 retorts, 37 tuyère Taylor stoker; 11,300 B.t.u. average per pound of coal as fired; 323 sq. ft. rear water wall; 143 sq. ft. side water wall.)

were exact duplicates but with the side water walls omitted from the second unit, decreasing the total water-wall surface by 32 per cent. The test results obtained are indicated in Fig. 18, showing an average gain of about 2 per cent for the side walls alone.

With reference to Mr. Moultrop's comment on the possible reduction in brickwork maintenance, the authors have found cases where the maintenance on completely refractory-lined furnaces averaged as high as $0.27 per ton for one year. Over a three-year period at one plant, it was found that the partially water-cooled furnaces effected a saving of $0.13 per ton over the complete refractory furnaces even though the water-cooled furnaces were subjected to more severe service.

Mr. Maynz brings out the fact that coal at $3.50 instead of $4.25 per ton would change the entire financial set up. This is true, but he has not credited the water walls with the increased capacity at which the boiler can be operated when equipped with a water-cooled furnace.

Boiler outage, which is a factor of prime importance in industrial plants, is also materially reduced by water walls. As the authors understand it, however, Mr. Maynz's criticism was

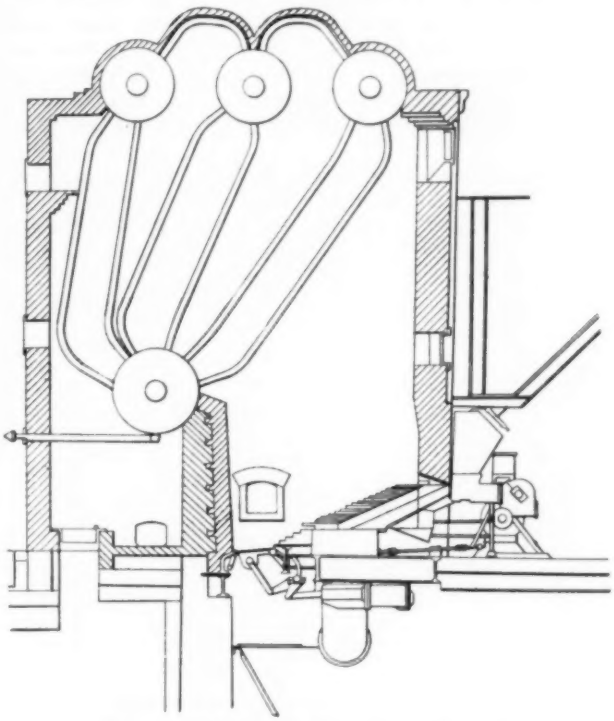


FIG. 19 BOILER WITH DUMP-TYPE STOKERS

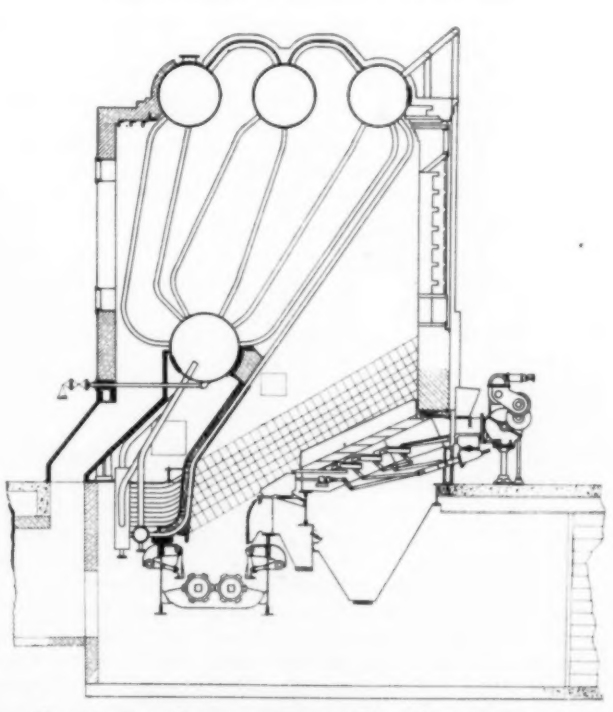


FIG. 20 SAME BOILER AS FIG. 19 WITH TAYLOR STOKERS

directed at the completely water-cooled furnace for the smaller boiler, whereas the partially water-cooled furnace shows a much better return.

The authors do not agree with Mr. Maynz that the rotary ash-discharge stoker cannot be justified for smaller boilers. Their use is increasing rapidly in the industrial field. In one case, an 823-hp. boiler, fired by a dump-type underfeed stoker, burning Ohio coal at $3 per ton, was replaced by a slightly larger rotary ash-discharge stoker with a rear water wall. The annual coal saving, using the same fuel, was $8400 per year, which, capitalized at 20 per cent, would justify an extra expenditure of $42,000 on the rotary ash-discharge stoker and water wall, far less than is required.

The dump-type stokers shown in Fig. 19 were replaced by Taylor stokers and American Engineering Company's water walls (Fig. 20) and gave 60 per cent more steam with a higher efficiency than before, at a cost of only $1.30 per pound of steam per hour extra capacity on a 312-hp. boiler.

Professor Christy asks about the effect of water-cooled walls on the stoker fuel bed when a high-volatile, high-sulphur coal, with low-fusing ash, is being burned, and the furnace is filled with a luminous flame which tends to shield the fuel bed from the walls.

Stoker practice has shown that an even flow of fuel across the width of a stoker can be maintained if the movement of the fuel at the sides is not retarded. When brickwork or other construction permits the fused ash to adhere to the wall, the flow of fuel along the sides is retarded, and, as the burning continues, the hindrance to the flow increases. Likewise, fused ash may accumulate on a refractory rear wall, impeding the discharge of the ash and consequently upsetting the combustion conditions in the entire fuel bed. Obviously, the use of water walls rectifies this condition. It is doubtful whether water walls have any real effect in modifying the furnace temperatures at points considerably distant from the walls.

Smoke-Abatement Progress in Salt Lake City

By J. BILLETER,[1] SALT LAKE CITY, UTAH

An observation tower 267 ft. above the street provides a convenient means of observing smoke conditions in Salt Lake City. A decrease in smoke from 10,891 min. per week in 1921 to 1050 min. per week in 1925 resulted from an active campaign. In 1925 active smoke-abatement work was discontinued, but was resumed again in 1927. During this time the minutes of smoke per week increased from 1050 to 2482. Police-court prosecution is found to produce but temporary results, and should be avoided if possible.

THE smoke department in Salt Lake City has conducted an active campaign against the making of smoke by industrial plants, larger heating plants, railroads, and residences. Observations are taken from an observation tower, of all dense-smoke periods (No. 3 or greater density of Ringelmann chart) of one or more minutes' duration. The observation tower is located centrally in the downtown district, 267 ft. above the street (on the top of the Walker Bank building).

It provides a view of the stacks of about 1300 heating and industrial plants. Many of these are operated smokelessly, so that actual contact and careful supervision from the tower are restricted to about 700 plants (not including railroads) at the present time. Telephone service enables the observer to communicate with any plant. Smoke violations observed are checked up as soon as possible by inspectors. If it is the fault of a new man firing at the place, instructions in proper firing methods are given. In case the plant is not in proper condition, pressure is brought to bear upon the responsible party to have the plant repaired. The primary duty of inspectors is the instruction of all firemen in the districts assigned to them. They caution firemen who show carelessness and inefficiency, and give any assistance that may be necessary when smoke occurs from breakdowns or other unexpected conditions.

Elimination of smoke from railroads consisted at first in equipping roundhouses and locomotives with smoke-prevention appliances, such as blowers, side-induction tubes, arches, and ring blowers. At present more attention is paid toward vigilance work and seeing that the equipment is kept up to date and used properly.

By far the most difficult problem is with residences. Experimental work done in 1922 and 1923 in small areas in the instruction of householders in the proper method of firing domestic equipment and supervision of operation after instruction proved conclusively that residence smoke could be reduced quite materially. Last year for the first time this work was carried out on a large scale. Each instructor was assigned a territory, over which he was held responsible. Results have been unusually good. Of course it is realized that this is only temporary relief and that the final solution will depend upon the use of smokeless fuels.

Inspection and supervision of the installation of new heating and power plants, including furnaces installed in residences, are important. The office directs all work and maintains coordination between the different activities of the smoke department.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

EXPLANATION OF TABLES

Observation from the tower and the patrolling of heating and industrial plants are usually started some time during October, the date depending upon weather conditions. The average duration of a campaign is about 25 weeks, the maximum having been 28 weeks and the minimum 20 weeks. The first part of a campaign comprises the period from October to December 31, which has been marked B in the tables, while the period from January 1 to the end of a campaign has been denoted by A. In several tables, plants have been divided into five groups: Hotels, apartment houses, business houses, manufacturing, and miscellaneous plants.

TABLE 1 TOWER OBSERVATIONS

Year	Total minutes of smoke observed	No. of weeks	Minutes of smoke per week	Average length of smoke period	Mean temperature
1921 A	119,800	11	10,891	11.7	40
1921 B	40,588	9	4,510	7.3	41
1922 A	78,094	15	5,206	5.8	31
1922 B	31,771	10	3,177	4.0	38
1923 A	40,959	15	2,731	3.6	35
1923 B	28,491	13	2,192	3.8	42
1924 A	31,725	15	2,115	3.6	35
1924 B	12,912	10	1,291	3.8	35
1925 A	11,547	11	1,050	3.6	39
1927 B	34,742	14	2,482	4.9	43
1928 A	33,385	15	2,226	4.3	38
1928 B	13,691	12	1,141	4.1	40
1929 A	12,279	11	1,116	3.7	30

Table 1, prepared from data taken on the observation tower, shows the best means of determining the effectiveness of the work done. All sources of smoke except the railroads and residences are included. The smoke decreased from 10,891 min. per week in 1921 A to 1050 min. in 1925 A or 90 per cent. The decrease has been steady except in 1922 A, which shows an increase over 1921 B. Figures for 1921 B may be considerably low, since they are available for nine weeks only, or 72 per cent of the average period of a campaign.

TABLE 2 LIST OF WORST VIOLATORS

	Hotels		Apartments		Business		Manufacturing		Misc.	
Year	C[1]	D[1]	C	D	C	D	C	D	C	D
1922–1923	16	1670	31	5270	25	2739	11	2759	1	145
1923–1924	18	3005	34	3866	29	3318	8	752	2	79
1924–1925	14	1364	24	2232	32	2183	5	540	0	0
1925	4	270	10	715	3	210	5	296	0	0
1927–1928	19	2434	41	5320	23	2549	8	1042	1	40
Total	71	8743	140	17,473	112	10,999	37	5389	4	264
Average per plant	1	123	1	125	1	98	1	145	1	66

[1] C = No. of plants. D = Min. of smoke.

The average length of each smoke period in 1925 A was 3.6 min. This is a decrease from 11.7 minutes in 1921 A, or 60 per cent. For about two years the average length of each smoke period remained fairly constant, indicating that the lower limits had been reached.

In the year 1925 active smoke-abatement work was discontinued, finally resuming again in the fall of 1927. The retrogression during the time smoke-abatement work was stopped is surprising. Results show 1927 B was as bad as 1923 when compared with minutes of smoke observed, and as bad as 1922 when compared with the average length of each smoke period. However by 1928 B, within a year, the records of 1925 A were being approached. A further decrease in smoke, and particularly the shortening of the average smoke period, depends directly upon the elimination of habitual and continual smoke violators.

Table 2 gives a list of the worst violators, segregating them into five groups of hotels, apartments, business houses, manufacturing plants, and miscellaneous plants. The number of plants in each group and the amount of smoke observed are given. So far as smoke production is concerned, individual plants of different groups do not seem to differ materially one from another. This table, as well as all other tables, emphasizes the fact that much ground is lost when smoke-abatement work is discontinued for any great length of time. Considering the groups independently, apartments contribute the largest amount of smoke, business houses come second, hotels third, and manufacturing and miscellaneous plants fourth and fifth, respectively

Two separate lists of violating plants have been kept, one of plants that violate once during any one week, and the second of plants that violate two or more times during any one week. Table 3 has been prepared from the latter list, while the two figures with an asterisk are taken from the former. The sum of the two would equal the total number of violations observed.

TABLE 3 NUMBER OF VIOLATIONS RECORDED AT OBSERVATION TOWER

Year	No. of weeks	Total violations	Violations per week	
1923 B	13	587	45	*44
1924 A	16	599	37	
1924 B	1	366	33	
1925 A	14	519	37	*49
1925 B	12	306	26	
1927 B	13	516	40	
1928 A	15	709	47	
1928 B	5	145	29	
1929 A	10	200	20	

TABLE 4 VIOLATIONS DIVIDED AS TO GROUPS

		Violations in per cent				
Year	Total violations per week	Hotels	Apartments	Business	Manufacturing	Misc.
1923 B	45	22	38	35	4	1
1924 A	37	22	36	31	9	2
1924 B	33	23	24	37	3	3
1925 A	37	18	37	33	6	6
1925 B	26	20	43	27	5	5
1927 B	40	18	40	28	10	4
1928 A	47	22	45	22	6	5
1928 B	29	20	43	22	9	6
1929 A	20	19	43	21	9	8

TABLE 5 POLICE COURT PROSECUTIONS

Year	Hotels	Apartments	Business	Manufacturing	Total
1921–1922	4	1	6	2	13
1922–1923	3	7			10
1923–1924	1				1
1927–1928	10	22	12	7	51
1928–1929	2	1	3	2	8
Total	20	31	21	11	

In Table 4 violations observed per week (as shown in Table 3) have been separated into the five groups, and are expressed in percentages. It is evident that violations of business houses have been decreasing and those of apartment houses increasing, while those of hotels have remained practically the same. As would be expected, apartment houses lead in number of violations, business houses are second, hotels third, manufacturing and miscellaneous plants fourth and fifth, respectively, thus confirming the observations of Table 2.

Police prosecutions have been resorted to only in extreme cases when other means of getting cooperation have failed. Results, usually, are not permanent. The effectiveness of police-court prosecutions has generally been overestimated. (See Table 5.)

Inspection work and calls at heating plants are a necessary part of smoke-abatement activity. While the number of inspections may in time be decreased, a certain amount of this work will always be necessary. The immediate checking up of violations by inspectors has been one of the most effective means of stopping the smoke. (See Table 6.)

Real progress in smoke abatement is best measured by the permanent improvements made in plants. Due to their nature, they must be done during mild weather, or when plants are not in use. At first, improvements consisted mostly in the installation of arches and the getting of proper chimneys, while lately the installation of stokers and air-panel doors has predominated. The present ordinance requires all plants except in residences to be equipped with a smokeless device and connected to a properly built chimney. (See Table 7.)

TABLE 6 INDUSTRIAL INSPECTIONS

Year	Inspections per week	No. of inspectors	Calls per inspector per week
1922 B	330	4	82
1923 A	330	4.4	75
1924 A	253	4	63
1924 B	116	2	58
1925 A	126	2	63
1925 B	143	2	71
1927 B	260	3	87
1928 B	151	2	76
1929 A	237	2.6	91

TABLE 7 RECONSTRUCTION OF PLANTS

	1921	1922	1923	1928
Arches	66	67	51	16
Extensions and chimneys	14	29	23	8
Stokers	5	2	0	44
New plants	4	4	1	4
Air panels	...	...	..	14
Miscellaneous	11	14	1	12
Total	100	116	76	98

TABLE 8 SMOKE RECORDS OF PLANTS

	Hotels	Apts.	Bus.	Mfg.	Misc.	Total
Number of plants investigated	23	12	20	2	2	59
Minutes of smoke per plant—1 yr.	1382	1433	1097	1327	420	5659
Minutes of smoke per plant—2 yr.	190	285	166	11	84	736
Reduction in smoke, per cent	86	80	85	99	80	87

Table 8 shows the reduction of smoke through the installation of smokeless devices. All plants investigated were operated under identical conditions before and after the change. The figures include all types of equipment. With the predominance of stoker installations at present, still greater reduction in smoke may be expected.

Permits are required for all boiler and furnace installations, but it is useless to require permits unless ample means for enforcement are provided. During the first few years shown on Table 9, the per cent of unapproved installations was large. The excess of approvals over permits in 1926 and 1928 is due to the fact that permits were taken out one year and finally approved the following year. Since 1926 practically all installations have been approved.

TABLE 9 PERMITS AND APPROVALS

	1922	1923	1924	1925	1926	1927	1928
Permits	272	701	726	913	929	801	953
Approvals	220	444	595	838	1011	655	1028
Difference	−52	−257	−131	−75	+82	−146	+75
Difference per cent	−19	−38	−18	−8	+9	−18	+10

TABLE 10 RESIDENCE INSPECTIONS

Year	Inspections per week	No. of inspectors	Calls per inspector per week
1922 B	151	4	38
1923 A	130	3½	37
1924 A	129	3⅛	41
1928 A*	317	3½	91
1928 A	360	12	30
1928 B*	137	2	69
1928 B	287	9½	30
1929 A*	406	7.7	52
1929 A	366	9	41

* Full-time men.

During the winters of 1922–1923 and 1923–1924, some experimental work was done in giving proper instructions to operators of furnaces in residences. Part-time men (university stu-

dents) were employed, working about 2½ hours each morning beginning at 6:30 a.m. In 1928 this work was carried out on a large scale with good results. Besides university students, several full-time instructors were employed, working in two shifts, four hours in the morning beginning at about 6:30 a.m. and four hours beginning about 2:00 p.m. During the first part of 1929 university students were replaced by men from the engineering department such as surveyors, paving inspectors, etc. who have little work at that time of the year. It is believed that this method will not only provide them with additional work, but also insure that the smoke department has competent instructors and inspectors with a minimum amount of loss from labor turnover. This method may also help in securing the budget money needed to carry on this work.

TABLE 11 RESIDENCE INSPECTIONS

	1928 A	1928 B	1929 A
Cooperation calls	929	1352	3492
Friendly calls	1285	856	1051
Indifferent calls	107	143	253
Miscellaneous calls	312	194	321
Total	2633	2545	5115

The public is cooperating whole heartedly as seen from Table 11. Miscellaneous calls are those made to homes where a chimney is smoking and nobody is home, or other cases where instructions cannot be given. Out of 2351 instruction calls in 1928 B and 4794 instruction calls in 1929 A, 94 per cent and 95 per cent, respectively, were friendly and cooperative, while only 6 per cent and 5 per cent, respectively, were indifferent. Those actually hostile are included in the figures for "indifferent" and amount to less than 1 per cent in each case.

Three Stages of Smoke Abatement

An important part in any smoke campaign is the creation of a favorable attitude of citizens and officials toward smoke elimination. Public support is essential for success in any public undertaking, especially in smoke-abatement work.

Smoke abatement in its broadest term includes elimination of smoke as, (1) smoke cloud, especially during early morning hours, also toward evening, (2) visible smoke from all plants, and (3) invisible smoke.

Smoke Cloud. This is a mixture of smoke, carbon, and condensation or fog. The only means of judging the proportion of smoke and fog in the atmosphere is the color of the cloud. Smoke appears dark and black, while fog has a light grayish color. The shade of the color of the cloud obviously depends upon the relative proportion of smoke and fog.

The formation of the "cloud" usually takes place in the morning, and less frequently toward evening. The building of fires produces a large amount of smoke, while atmospheric conditions are favorable for fog formation. Salt Lake City is surrounded by mountainous regions covered with snow continually during the winter months. Due to large radiation the air is cooled below its dew point, separating the vapor into visible water drops. This condition is most pronounced in the morning when new fires are being built.

Smoke increases the tendency of fog formation, while fog in turn is favorable to smoke-cloud formation. Air movements are unusually low, and the direction of the wind changes constantly, creating an "eddy-current" effect, with the result that all smoke is collected into a dense cloud. This cloud hovers over a certain section of the city for some time, and later is blown to other sections with great rapidity. The density of such a cloud is almost impossible to determine. It varies in different parts of the cloud at different times and never remains constant. As the distance to the cloud increases, the apparent density increases. Several city blocks from the cloud, its density appears much greater than it really is. While it may be impossible to eliminate this cloud entirely at the present time, its size, density, and duration can and have been greatly decreased.

The public judges success in smoke abatement by the decrease of the "smoke cloud" more than anything else. The average person is not accustomed to watching smoke, and notices only general conditions of the atmosphere at the time he leaves home. By decreasing the morning cloud, his attention is gradually directed toward observations of smoke conditions generally, which later leads to observation of individual smokestacks. This process is slow, but nevertheless effective.

Visible Smoke. With the observation of smoking chimneys on the part of the public, a desire grows to eliminate that smoke. It is realized that the expense of keeping clean and healthy is unusually high and can be cut materially.

The most widely adapted means for smoke elimination is the use of proper methods of building and replenishing fires. Even with poor equipment, proper firing improves smoky conditions.

There are many useful smokeless devices on the market which help eliminate smoke from the larger hand-fired boilers. The use of smokeless fuels and automatic appliances has greatly increased. This factor has contributed more than any other toward permanent smokelessness. There are in Salt Lake City at present 275 small and 175 large stokers, 390 oil burners, and 128 gas installations.

Invisible Smoke. Invisible smoke includes undesirable, practically colorless gases, very small soot particles, dust, and other impurities. It constitutes a further step after elimination of visible smoke. Very little work has been done up to the present, except in industrial plants. This phase of smoke abatement is not included in the public's conception of smoke elimination. It has not been considered in this discussion.

Conclusions

1 With present equipment the amount of smoke made from all plants, except residences, can be kept low enough so as not to create a smoke cloud.

2 Elimination of the smoke cloud depends upon the elimination of a few heavy smokers, rather than many light smokers.

3 Most of the visible smoke is produced by a minority of plants which are continual and persistent in creating smoke.

4 Plants in different groups produce the same amount of smoke (except some industrial plants).

5 The percentage of cooperation from the different groups is the same.

6 Smoke performance of plants is not necessarily measured by the number of violations.

7 Results obtained from employing good firemen are only temporary, unless the plants are improved.

8 Police-court prosecution usually brings only temporary results, and should be avoided whenever possible.

9 Permanent smoke elimination may be obtained by installation of smokeless devices, especially stokers, and by burning smokeless fuel.

10 With proper care and attention, stoker fires can be cleaned and flues blown without making any smoke.

11 Smoke from the downtown district, due to better supervision, has been diminished greatly.

12 Violations from apartment houses have increased. This is due to the rapid increase in apartment houses, their scattered locations, and the difficulty of adequate supervision.

13 Violations from hotels have remained practically constant.

14 Effective smoke-abatement work requires a certain minimum number of inspectors over a long period of years. (The number of inspectors depends upon size and population of city.)

15 Smoke from residences can be reduced materially by giving instructions in the proper methods of building and replenishing fires.

16 Residence furnace operators are willing to cooperate if approached properly.

17 There is no relation between the mean monthly temperature and the amount of smoke made.

18 Smoke hours observed by the U. S. Weather Bureau apply to changes of the smoke cloud, and do not indicate the amount of visible smoke eliminated.

The Preparation of Anthracite

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This paper covers the development and present practice of the methods used in the manufacturing process called "preparation," necessary to the conversion of the raw anthracite produced from the mines into the finished products consumed. The paper is divided into three main sections covering the crushing, sizing, and cleaning of the coal and into supplementary ones dealing with breaker design, operation, and efficiency.

THE TERM "preparation" is used in the anthracite industry to define the complete conversion of the raw run-of-mine product into the various sizes commercially consumed. It covers a complicated manufacturing process requiring the following steps: (1) Crushing, (2) sizing, and (3) cleaning.

Anthracite as mined is a heterogeneous mixture of pieces of coal, intergrown coal and refuse, and refuse, ranging in size from lumps of a hundred pounds or more to the finest dust. This material, containing from 12 to over 50 per cent of refuse, must be so crushed, sized, and cleaned as to meet the following general specifications:

PREPARATION STANDARDS

Size	Over hole, in.	Through hole, in.	Oversize, %	Undersize, %	Slate, %	Bone, %
Broken	3 7/16	4 7/16	5	10	2	2
Egg	2 8/16	3 7/16	5	15	2	3
Stove	1 9/16	2 8/16	5	15	3	4
Nut	11/16[3]	1 9/16	3	10[3]	4	5
Pea	9/16	11/16	5	10	7.5	7.5
Buckwheat	5/16[1]	9/16[1]	5[1]	20[1]	...[1]	...[1]
Rice	3/16[1]	5/16[1]	..[1]	20[1]	...[1]	...[1]
Barley	3/32[1]	3/16[1]	..[1]	20[1]	...[1]	...[1]
No. 4	3/64[1]	3/32[1]	..[1]	..[1]	...[1]	...[1]
Silt[2]	..	3/64[2]	..	..	...	...

[1] No common standard in use; size, etc., depends on contract requirements, most of these sizes being sold to users of large tonnage. Ash is reduced to from 12 per cent to 18 per cent depending on size.
[2] Generally a waste product.
[3] Nut, if made over 3/4-in. screen, can carry 15 per cent undersize.

Sizes larger than broken, known as lump and steamboat, are shipped but rarely, and broken itself is not produced at many collieries, the general trend in domestic demand being toward the smaller prepared sizes.

Those sizes larger than pea are termed "prepared sizes;" pea and smaller, "steam sizes;" and the material passing a 1/16-in. or 3/64-in. round-hole screen, "silt." Pea is sometimes included with the larger sizes, those above buckwheat then being called "domestic sizes." Silt is generally a waste product. There is, however, a limited use of it, both for sintering iron ore and for briquetting, but no extensive demand as a powdered-fuel source has yet developed.

The ratio of prepared to steam sizes is of vital importance to the anthracite producer as this determines the average selling price ("realization") for the shipped product. The prepared sizes used for domestic purposes naturally command very much higher prices than the steam sizes, and therefore the average realization from the coals of the Northern Field, running from 65 to 75 per cent in prepared sizes, is higher than in some parts of the Southern Field, in which less than 50 per cent of prepared sizes can be expected.

These differences in possible yield of prepared sizes are due either to (*a*) natural crushing of the coal underground by flexures or faults in the enclosing strata, (*b*) inherent weakness in physical structure of the coal itself, or (*c*) a combination of these two causes.

[1] Consulting Mining Engineer.

It must not be concluded from the foregoing that all coals of low prepared yield are necessarily of less intrinsic value, because it frequently happens that the total cost of production of coals of a smaller average sales realization is sufficiently low to make up for the handicap in selling price.

Whenever coal is handled, a certain amount of breakage, or degradation in size, is unavoidable. This occurs in every department of the preparation process, and the aim of the entire plant design should be to reduce it to the minimum possible.

That breakage which results in the production of silt constitutes a direct loss in shipped product, or yield, but in many cases the degradation of prepared sizes to those of steam grade is of equal economic importance. Thus, if the average difference in selling price between the prepared sizes and steam coal is taken at $4.50, then each per cent of loss in prepared means a direct loss of $0.055 per ton of shipped coal, and a plant increasing the prepared sizes from 60 to 65 per cent will show a direct increase in average sales price of $0.275 per ton.

The reduction of impurities below those at which the coal can be passed as merchantable would appear to constitute a direct economic loss, and this view was long held by the industry in general. The experience of the last few years has demonstrated the fallacy of this viewpoint, and it is now the aim of some of the largest producers to clean the coal to the highest standard possible. The gain from this policy is either the direct one of selling the product at a premium or the indirect one of maintaining output through periods of poor market demand.

Crushing

History. Anthracite was originally shipped in large sizes, lump and steamboat coal being used as a blast-furnace fuel and the chestnut and smaller sizes being left in the mines, loading being accomplished by tine forks. Such crushing as was practiced was done by hand, at the surface.

Rolls were introduced at an early date, and various types of smooth, corrugated, and toothed-roll faces were developed. It was soon found that the toothed roll most nearly approximated the efficient result that can be obtained manually with a pointed pick, and this type of roll came into general use.

Present Practice. Rolls were originally operated at peripheral speeds of 1000 ft. per min. or more, corresponding to the trajectory velocity of the falling material. These machines are directly belted to the lineshaft and are known as "high-speed rolls."

At present, geared rolls, operating at speeds of 250 to 300 ft., are generally used for crushing lump coal to broken and smaller. These machines are called "compound-geared rolls," and it is believed that their use has resulted in measurably improved crushing efficiency. In rolls of this type but one roll is power driven, the other being geared from it. This second driven roll is held up to its work by springs that provide a release should uncrushable material occur in the feed.

The favorite type of tooth is a curved pyramidal one with the hook facing forward—one such shape in common use being known as the "hawk's bill." The teeth are generally made of manganese steel and are either inserted in the roll shell or cast in segments attached thereto.

Fluted-face or corrugated high-speed rolls are generally used for crushing bone and refuse to free intergrown pure coal prior to rewashing. Both jaw and gyratory rock crushers are used

for breaking down large refuse, and high-speed hammer mills are used for comminuting the final breaker refuse when the latter is to be used for mine filling.

It is the present practice to crush the feed in two or three stages, as this results in larger yield of prepared sizes. Thus the lump may be crushed to steamboat, the latter to broken, and the broken to egg, the material smaller than each of these sizes being removed by screening ahead of each set of rolls. When no coal larger than stove is to be shipped, it is doubtful whether the reduced degradation by installing a fourth set of rolls would cover the increased cost of such extended crushing and screening.

With modern slow-speed rolls the material after crushing should contain at least 90 per cent prepared sizes, if the usual reduction per roll is adhered to and rolls of proper diameter are used. Lump rolls should be not less than 42 in. in diameter, and a diameter of 54 in. has been found to give results justifying the increased cost. Rolls crushing to egg should not be less than 36 in., and 42 or 46 in. diameters are efficient in service. The width varies with the tonnage to be handled, ranging from 34 to 50 in.

Modern roll frames and gears are generally made of cast steel, this material well withstanding the heavy shocks to which the rolls are subjected. The drive is by belt or rope, either of these providing the necessary slip when choking occurs.

Sizing

History. The first sizing of anthracite was carried out on stationary bar screens. At an early date, revolving screens, similar to stone trommels, were introduced, and these persisted in practice down to the last decade. It was found that serious breakage was caused by this type of screen due to the rolling of the coal. Further, blinding of the holes occurred if much clay were present in the coal; but a small area of the screen was available for actual use; and the wear per ton of coal was excessive when acid water was present.

These defects led to the adoption of the shaking type of screen, and today this is generally used, although stationary-bar lump screens are still retained at a few of the older collieries. Eckley B. Coxe, the greatest contributor to the mechanical side of anthracite preparation, experimented with various types of gyrating, traveling-bar, and other screens, but these have all been abandoned.

Present Practice. The most commonly used shaking screen is of the wood side-type suspended by flexible wood hangers and driven by wooden driving arms fitted with spring pieces to provide flexibility. It has the advantage of cheapness, low weight, and a snap at the end of the stroke tending to clear the mesh. Supporting legs at times replace the hanger boards. The capacity varies with the size of screen and character of feed, 125 tons per hour being a fair figure for a four-deck shaker 5 ft. wide and 17 ft. 3 in. long.

The following data cover most of the structural features in common use:

Shaker sides—3-in. or 4-in. by 6-in. oak or longleaf pine; 6-in. ship channels; various sizes of steel angles.

Hanger boards or supporting legs—1-in. by 8-in. oak or hickory spring boards; pin-connected wood hangers; pin-connected steel members; steel chains.

Segment side support—segment flange itself most desirable; side angles sometimes used, but objectionable due to loss of screen area.

Segment cross support—$2^1/_2$-in. and larger steel angles.

Screen segments—punched plate of steel, copper, Monel metal, bronze, and/or iron-chromium alloy, cast segments being at times used for lump coal; wire mesh is at times used for fine sizes, but not suited to coarse sizes due to blinding and wear.

Driving arms (usually in pairs)—3-in. or 4-in. by 6-in. wood arms, with wood spring pieces; wood, bar steel, or steel pipe with wristpin connections to shaker sides.

Driving shaft—$3^{15}/_{16}$-in. to 6-in. steel shaft fitted with cast-iron plain, babbitted, or bronze-bushed eccentrics or provided with forged crank throws.

Drive—belt, rope, or special drive, such as Texrope.

Horsepower—3 to 12 hp. per deck, depending on length and width.

Decks per driving arm—one, two, or, rarely, three.

Weight per deck—1000 to 6000 lb.

Crank or eccentric layout—two arms at 180 deg.; three arms at 120 deg.

Shaft speed and stroke—lump shaker, 120 to 150 r.p.m., 5 to 6 in.; prepared shakers, 145 to 160 r.p.m., 5 to 6 in.; steam shakers, 150 to 200 r.p.m., 3 to 6 in.

Screen dimensions—4 to 7 ft. inside width; 12 to 70 ft. length.

In general, the latest practice involves shorter stroke and higher speed for the steam sizes, and a preference is given by many to crankarm drives, although eccentrics predominate.

The screens are set at a pitch varying from $^1/_2$ to $1^1/_2$ in. per ft., this being largely a matter of individual judgment. Increased pitch decreases screening efficiency, but increases conveying capacity, while lump coal will travel forward faster than fine coal on a light pitch. A $^3/_4$-in. pitch is frequently used for prepared coal, with the hanger boards inclined forward perpendicular to the screen pitch. With this arrangement, a conveying speed of 60 ft. per min. is reached at 160 r.p.m.

Screen decks are usually superposed, each finer size descending to the next lower deck. Screening is also at times carried out on a single deck, the sizes being removed in series with the finest first.

Some designers incline the hanger boards to the rear at about $1^1/_2$-in. pitch, giving a positive vertical arc of travel. This tends to jerk the material forward at the end of each stroke, increasing capacity and cleaning the mesh. The screen can be set level or even slightly up-grade, which is advantageous in saving breaker height. The practice is not general, as there is a divided opinion as to its relative merits.

High-speed vibrating screens (familiar to the bituminous industry) set on a pitch at which the material will flow have not yet given the accurate sizing demanded by the anthracite industry. It is believed that these may be applied to sizing and dewatering fine coal and in a more nearly horizontal form to the sizing of larger coal.

Hydraulic classifiers, of the Dorr rake type, are successfully in use for sizing and dewatering silt, the operation being a combined cleaning and sizing one.

Cleaning

History. The first cleaning of anthracite was confined to underground hand-picking at the working "face." This is still practiced in flat dipping measures, but is impossible where the coal is loaded by gravity on steep pitches. The efficiency of such "face" preparation has steadily decreased with the working of thinner, dirtier coal beds and with the desired increased output per loader, causing more and more work to be done outside the mine.

When surface preparation came into practice, it was at first confined to hand-picking. Later, various types of mechanical separators were used, hand-picking being retained for the larger sizes and for recleaning the separator products.

Mechanical separators were generally called slaters or pickers and involved the use of: (*a*) Shape—coal, cubical; slate, flat. (*b*) Friction—coal, low coefficient of friction; slate, high coefficient of friction.

The simplest form of slater consists of a slot too narrow to pass the more or less cubical coal, but wide enough to admit the slate. This device is sometimes incorporated in the shaker screen itself. It is also built as a series of small-diameter rolls

revolving in the same direction, the slate passing through the spaces between the rolls and the coal traveling over them, a device of this latter type now being built by one manufacturer.

The frictional type of slater functions by causing the coal to jump a gap after passing over a frictional surface that retards the slate more than the refuse, the separating effect being produced by the greater kinetic energy acquired by the coal.

The latest and most successful type of frictional slater is known as the "spiral separator." It consists of a chute wound as a vertical helix, the coal and slate running down this spiral chute receiving differential centrifugal accelerations by reason of their different coefficients of friction and thus being in effect centrifugally separated. This type of machine may be operated either on wet or dry coal, but it does not do the best work unless the operation is a wholly wet or dry one.

Another late development is the "Ayres picker," using the frictional characteristics of a moving belt for separating the larger sizes.

Mechanical separators do not approach a metallurgically perfect result because:

1 Some coal is flat and some refuse cubical
2 Pieces of laminated coal and slate will go with coal or refuse in a frictional machine, depending on which side is under. Thus a thick piece of refuse with a thin surface of coal may be retained with the coal or a thin piece with a thick layer of coal sent to the refuse
3 Screening methods do not permit a sufficiently close approximation in size.

Hand-picking of the separated products has been generally required to produce a finally satisfactory separation. However, these mechanical devices served their purpose very well in the old dry-preparation breakers of the Northern Field. Their relative importance has steadily diminished with the increasing severity of the preparation problem, and there is little hope that they can be so improved as to become an important factor in future anthracite preparation.

In the Southern Field, the high proportion of refuse, coupled with an exceedingly wet product, caused an early adoption of wet cleaning methods. Hand-picking was retained for the larger sizes, but the hydraulic method of jigging was used for the smaller prepared sizes. There was thus a division of preparation practice into the dry breakers of the Northern Field, combinations of dry and wet breakers in that and other fields, and purely wet breakers in some parts of the Southern Field.

At the present time plants employing dry preparation are gradually being displaced by those using the various methods of wet cleaning, and in the latest installations in every district the newest plants are being equipped with wet cleaning devices throughout. This is due to the increased purity demanded in the product, the higher efficiency of the wet breaker, and the wet character of many breaker feeds. It is therefore thought unnecessary further to describe the essentially dry methods of operation formerly in favor, and the following notes on present practice and flow sheets in use will be confined to wet methods, even though some future breakers may be built to operate without wet cleaning of the crushed pure lump coal.

It is not thought that pneumatic tables of the type used for bituminous-coal cleaning will be adopted by the anthracite industry because these as now built require a dry feed, a condition not generally present in anthracite practice.

Present Practice—Preliminary Preparation. Lump coal is generally hand-picked to remove the large refuse. Picking is carried out on stationary platforms or on moving picking tables of the apron-conveyor or shaking type. From 2 to 16 men are usually required for this work, and 25 per cent of the raw feed sent to the breaker is frequently removed as refuse. In many cases 6-in. lump and 6 in. by $4^7/_{16}$ in. are separately picked, and at times the 4 in. by $3^7/_{16}$ in. broken is also cleaned. At a few collieries the mine rock is sent to the breaker and the good coal picked before transport to the refuse bank. If a large amount of bone or laminated coal and refuse occurs, this may be removed separately for further treatment.

All new collieries are being provided with moving picking tables, the platform being obsolete. When these tables are of the shaking type, they are either operated at long stroke (6 in.) and low speed (90 r.p.m.) or at short (2 in.) stroke and high speed (160 r.p.m.). The shaking table is supported on flexible, or pin-connected, legs and is vibrated by mechanism similar to that used for screens. This type appears to be in greatest favor where large pieces of refuse are to be removed. It is from 4 to $5^1/_2$ ft. wide and from 12 to 24 ft. long.

Principles of Cleaning by Hydraulic Methods. All wet methods now in use, with the exception of the sand-flotation process, depend on differences in the falling velocity in water of particles of coal and refuse. These differences may be utilized in either free-falling or hindered-settling apparatus, the jig and concentrating table being examples of the latter, the launder method a combination of the two, and the rake classifier an example of the former. Rittinger's formula for velocity of fall in water may be reduced to the following simple ratio for particles of like shape but unlike specific gravity:

$$V_c = \sqrt{D_c(\phi_c - 1)}$$
$$V_s = \sqrt{D_s(\phi_s - 1)}$$

When V_c = velocity of fall of coal in feet per second
V_s = velocity of fall of slate in feet per second
D_c = diameter of coal in feet
D_s = diameter of slate in feet
ϕ_c = specific gravity of coal
ϕ_s = specific gravity of slate

Hence if we wish to group particles of like falling velocity:

$$D_c(\phi_c - 1) = D_s(\phi_s - 1)$$

and for anthracite of 1.6 and slate of 2.4 specific gravity:

$$D_c = 2.333\ D_s$$

or a piece of coal 2.333 times as large as a piece of slate will fall with equal velocity if of the same shape. Hence, a smaller piece of coal will fall at less velocity and position itself above the slate, so that if the coal and slate are sized to within this ratio, a separation can be effected. A comparison of these ratios with those obtaining for bituminous coal of 1.4 and refuse of 2.4 specific gravity gives:

For anthracite: $D_c = 2.33\ D_s$
For bituminous: $D_c = 3.5\ D_s$

It will therefore be understood why apparatus satisfactory for the washing of bituminous coal may fail to meet the more rigorous requirements of anthracite preparation—a fact not always appreciated when apparatus from bituminous practice is installed in an anthracite plant.

The foregoing ratios apply to the free and unchecked fall of particles in what is termed a "free falling" process. As already stated, this process has had but a limited application in anthracite practice, a further development of this method known as "hindered settling" being the controlling principle in the apparatus used for hydraulic cleaning.

In hindered settling, the particles are permitted to approach each other until they touch, forming a heterogeneous mixture, or "bed," with the interstices filled with water, the hindered settling effect being produced by:

(*a*) Upward currents through the beds, as in jigs, classifiers, and trough washers

(*b*) Horizontal agitation, as in tables and trough washers.

The relative falling velocities of different-sized pieces of refuse cannot be calculated by a simple formula for hindered-settling conditions. However, the ratio of size of particles that can be separated by this method is considerably larger than that possible in free settling and, further, much lower velocities are needed in the upward-current water.

Jigs. The anthracite jig is a development of machines of similar character used in ore dressing, consisting of a screen supporting the bed of coal and refuse in a wooden or steel tank provided with means for introducing the raw feed and for removing the separated products, including the slush passing through the jig screen. In operation, the refuse is stratified above the coal by water pulsations passing up through the jig screen and collects thereon. The refuse is periodically or continuously withdrawn from this screen and the coal continuously removed from the upper portion of the bed. Anthracite jigs are of two generic types—plunger jigs and pan jigs. In the former, pulsations of the water are produced by a reciprocating plunger operated by eccentrics, while in the latter the jig screen is itself vertically reciprocated.

The coal is sometimes discharged by a water overflow, but in most cases a scraper conveyor is used for such discharge, overflow jigs operating with less breakage of the cleaned coal. Both plunger and pan jigs may or may not produce suction in the jig bed. The suction jig separates the materials by a series of harmonic pulsations of equal amount, the heavy material being elevated less rapidly and falling more rapidly than the light material. The non-suction jig utilizes a series of unidirectional pulsations to secure the separating result, the fall between pulsations occurring in comparatively still liquid. Both types have their adherents, and no marked advantages appear to be present in either system. The non-suction effect may be produced by introducing the overflow water under the plunger by operating the plunger or pan by a quick-return motion, or by fitting the plunger or pan with non-return valves (producing the effect of a single-acting pump).

Special types of plunger, pulsator, and pan jigs have been introduced for treating the various sizes of coal. Of late years the pan jig has come into extensive use and has been well liked by those using the jig method of preparation.

Various methods of mechanically removing the coal and refuse from the jig tank are in use, including conveyors, elevators, the Tench plunger, slate wheels, etc. The amount of refuse withdrawn is either automatically effected by a weighted pan, is manually regulated, or is controlled by gates in a manner similar to the cup-and-dam discharge common in ore-dressing practice.

Jigs treat each size in separate machines, the capacity of the units varying from 6 to 20 tons per hour, depending on the material treated and size of the jig.

Tables. Wet concentrating tables of the type used in ore-dressing plants are used at many collieries for the treatment of buckwheat and steam coal. The table consists of a plane rhomboid surface called the deck, adapted to be reciprocated rapidly by a quick-return motion and to be tilted both axially and transversely. The deck is covered with low steps or riffles running longitudinally in the direction of motion and is provided with a feed box at its upper inner corner and a wash-water launder along its upper edge. A fixed clean-coal-collecting chute runs along the lower longitudinal edge and a fixed refuse chute under the upper transverse edge.

The deck is supported on flexible, or hinged, legs and moves faster on its in-stroke than on its out-stroke, propelling the materials in the riffles toward the outer discharge end. The reciprocation produces a vibratory movement that agitates the material in the riffles and permits it to stratify under hindered settling conditions. The coal thus rises to the top and is sluiced down from the riffles by the wash water while the refuse travels longitudinally in the riffles toward the refuse discharge.

The zone of separation between the pure refuse and pure coal consists of a "twilight middlings zone," containing good coal, refuse, and intergrown coal and slate. The location of this zone may be controlled by the inclination of the table, the quantity of wash water, and the character of the table vibrations.

Tables have not come into extensive use for sizes larger than buckwheat coal, and their principal field appears to be in the cleaning of rice and smaller sizes. They treat from 3 to 15 tons per hour, dependent on the size of the material and character of the feed.

Trough Washers. The concentration of heavy minerals in hydraulic sluices has been practiced since the beginning of the ore-dressing art, and the same methods have been long employed in coal washing. In the simplest form, a long inclined sluice, provided with dams or riffles, is used, the coal washing down over these riffles and the refuse collecting therein. To avoid the manual cleaning of these riffles, moving dams consisting of flights carried on conveyor chains, or in some cases taking the form of a spiral flight within an inclined revolving drum, have been used continuously to remove the refuse. In other types classifier pockets are located along the bottom of the trough, the refuse falling in these pockets either unretarded or against an upward hydraulic current.

One of the latest types of trough washers of the last-mentioned type is the Rheolaveur, a machine that has been widely introduced on the Continent, especially in France and Belgium, where various types of trough washers have long been in favor. In this machine the difficulty of making a satisfactory separation, without either troughs of great length or the shipment of a middling product, is met by:

1 Continuously recirculating the middlings (consisting of pure coal, intergrown coal and refuse, and pure refuse) in a section of the trough between a primary refuse discharge and the middlings discharge. This builds up a thick bed between the pure coal and refuse similar to a jig bed and permits the discharge of the purer coal from the top of this bed as it passes over the middlings discharge.

2 Rewashing the refuse in additional troughs to free it of mechanically entrapped pure coal and middlings.

3 Rewashing the fine coal in separate troughs designed for this particular purpose.

In practice the unsized feed is sent to the primary washing trough. The larger refuse and middlings are removed from this primary trough by water-sealed elevators, and the clean coal is discharged from the end, after which it is screened into the various market sizes of coal. The steam coal passing through the screen is sent to the steam-coal-washing troughs for further treatment. The refuse from the primary trough is re-elevated for further treatment in the refuse-rewash trough.

The Rheolaveur would appear to use a combination of hindered-settling and frictional methods of separation, as the material rolling down in the trough is assisted in its stratification by differences in frictional resistance, while in its passage over the refuse and middlings discharges an upward current may be utilized for producing hindered-settling conditions. Further, in some of the discharges from the fine-coal trough free-settling conditions may be allowed to exist. Shape also plays an important part, the flat slate not being as readily transported through the trough by the washing water as the more cubical coal.

Classifiers and Settlers. Hydraulic classifiers are used to some

extent in the preparation of anthracite coal. These include series machines, designed to retreat the refuse—such as the "Hydrotator;" direct-flow classifiers of the Menzies, Grant, and other types; and free-settling rake classifiers such as are built by the Dorr company. The various types of classifiers are generally confined to operation on buckwheat and finer coal and function in a manner similar to that of their ore-dressing prototypes. Rake classifiers have been principally used to dewater tabled coal, and incidentally to secure a concentration by washing out the very fine high-ash material, this latter being carried to waste with the overflow water.

Settlers are either rectangular or conical settling tanks. They may or may not be provided with conveyors or other means for continuously removing the settled product. They serve to clarify the washery water either for reuse or before discharging it to waste. Of late years the Dorr type of circular thickener has come into considerable use for this purpose, thickeners upward of 80 ft. in diameter having been installed.

Résumé of Hydraulic-Cleaning Methods. The present separating methods successfully prepare anthracite to the normal slate and bone percentages with reasonable losses of coal in the refuse. If an improved standard is required, it can be attained only with the loss of excessive amounts of coal to the refuse, or if a better refuse is demanded, this is gained at the expense of decreased purity of the cleaned coal. Tabling is limited to the smaller sizes of coal, requires large building volume, but on the finer sizes does better work than most jigs. Jigging requires constant supervision. If the refuse gate draws slate too rapidly, good coal passes to the refuse, and if the slate is drawn too slowly, refuse is removed with the coal. Neither jigging nor tabling is an exact process; hence flat slate caught broadside in the separating currents may go with the coal and flat coal caught on edge go to the refuse. This particular defect is so troublesome at some Schuylkill collieries that the flat material (coal and slate) is removed by slot slaters and treated separately in "flat coal" jigs.

The existing methods of hydraulic cleaning all require storage of the feed for efficient operation. In addition, they demand either close sizing of the raw coal or recirculation of a middlings product to secure the desired results. Storage, close sizing of the raw coal, and middlings circulation all contribute to breakage of the coal, resulting in the degradation of the valuable prepared sizes to the cheaper steam coals and practically worthless silt. In addition, considerable breakage is caused within the separating machines themselves due to the attrition caused by the particles rubbing against each other in the relatively compact separating bed.

In considering the necessity of close sizing in the operation of anthracite jigs as generally installed, it should be noticed that, even with the small differences in screen opening allowed in the present anthracite-sizing scale, the largest pieces of coal will be three or more times that of the smallest pieces in any given size, because the coal breaks into particles of almost every conceivable shape. In practice the largest pieces in any one particular size weigh from two to five times as much as the smallest pieces. This great variation in size, shape, and weight causes the principal difficulty in the jigging of anthracite.

The Rheolaveur presents the great advantage of treating an unsized feed. It, however, requires a continuous recirculation of a part of the product under process, this middling treatment inherently producing breakage that, partially, at least, offsets that produced by close sizing of the raw coal. If the shipped product and refuse are of higher grade than that produced by jigging with an equivalent retreatment of the refuse, then this process should be a successful competitor because of the large tonnage which may be treated in a single unit. As it has already been used at five collieries on mixed sizes, one on steam coal, and two on silt, its future should naturally depend upon its success in meeting the rigorous standards now required by the anthracite industry.

Sand Flotation. The sand-flotation process is a commercial application of the float-and-sink method of testing coals in which a solution of density sufficient to float the coal, but permitting the refuse to sink, is used. It has been developed in the belief that it would overcome some of the difficulties inherent in the older methods of anthracite preparation.

Solutions of solids in liquid are so expensive and so difficult to recover from the separated products that attempts commercially to use them have not been successful. In sand flotation, the heavy solution is replaced by a high-density mechanical mixture of a comminuted solid (such as sand) and water, this mixture being termed a "fluid mass." It is maintained in a highly mobile condition by hydraulic agitation, and is thus caused to assume the general physical characteristics of a liquid of high density. The densities used in anthracite operation vary from 1.6 to above 1.75.

A steel-plate cylindro-conical type of separator has been generally used in commercial plants, this providing an efficient method of withdrawing the refuse from the high-density fluid mass. In this type of apparatus the lower part of the fluid mass is supported on an upwardly rising column of the hydraulic water required for agitation.

The conduit containing this column is termed the "classifier column," and the agitation water rises therethrough at a velocity permitting the refuse to fall, but sustaining most of the sand. The coal is discharged by causing some of the fluid mass to overflow, and the refuse is withdrawn through a water seal by interlocked slide valves at the base of the classifier column. A mechanical agitator is used with machines operating at the higher densities (this effecting a diffusion of the very small quantities of agitation water required for these densities), and additional agitation water may be introduced through the walls of the cone. The sand discharged with the separated products is reclaimed by screening and returned to the separating tank.

Sand flotation has been introduced at 31 anthracite collieries. When properly operated it can practically duplicate the results of the float-and-sink test. As it separates the coal by a purely flotational method, particles of any desired size, down to the smallest that can be separated from the sand, may be treated. Size, shape, and frictional characteristics therefore play no part in the separating process, the only property employed being that of specific gravity. The ash-content curve closely follows the specific-gravity curve in any given coal; therefore, such a specific-gravity separation affords close control of the ash in the washed coal and refuse.

The operating results of the last eight years have indicated that the use of sand flotation permits, among other things:

Shipment of pure coal
Minimum breakage
Low attendance
Plants of low first cost
Intermittent feed
Radical changes in quality of feed
Treatment of unsized feed.

Sand-flotation cones have been built to ship from 20 tons per hour to 200 tons per hour, ranging from 7 ft. 8 in. to 15 ft. in diameter, and 18-ft. diameter cones are now under construction of greater capacity.

Breaker Construction

The entire preparation plant is called a "breaker," and it is

built either as an integral unit, from mine-car dump to loading pockets, or as two detached structures—a rock house for preliminary treatment of lump and steamboat and the breaker proper handling a feed crushed to broken or egg.

Breakers are built to ship from 300 to 7000 tons per 8-hour day, and single-shift operation is the rule, although one company has operated some double-shift plants and another is contemplating an extensive two-shift program.

Construction is of:

Wood throughout
Steel frame with wood floors
Steel frame and concrete
Reinforced concrete (rare).

Sheathing is of wood, corrugated steel, asbestos-covered steel, or corrugated asbestos. Concrete and gypsum are at times used for roofing. Considerable areas of the siding are now glazed, metal sash being much used.

Breaker heights range from 60 to 200 ft., dependent on the flow sheet used. Gravity handling of all materials is sought for, and this increases the height. In all breakers the vibratory stresses due to the shaking screens is excessive, requiring much bracing, and high-concentrated loads are caused by pockets and tanks.

The present trend is toward fire-resistant construction, with steel frame, concrete floors, fireproof siding and roofing, and steel or concrete pockets. Chutes and hoppers in the newer breakers are of steel or concrete and are sometimes lined with tile.

The clean coal is either loaded from pockets into the railroad cars or storage of the prepared sizes is eliminated by the use of loading booms, pockets being retained, however, for the steam sizes. Loading booms reduce the breakage, as they do away with crushing in the pockets. Their use on all prepared sizes has been confined to four sand-flotation breakers, the Laflin colliery of the Hudson Coal Company being the first to adopt this practce.

Materials are handled by:

Scraper conveyors
Belt conveyors
Chain bucket elevators
Shaking chutes
Gravity chutes.

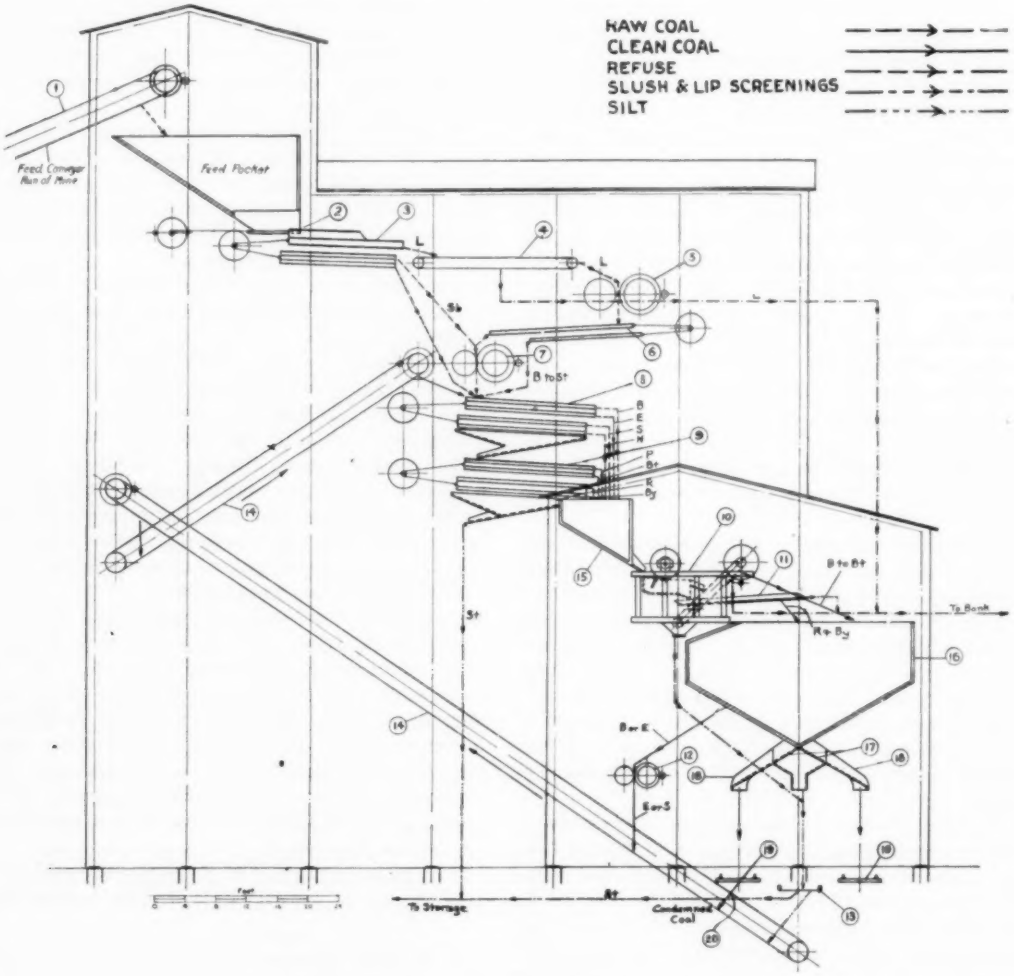


FIG. 1 TYPICAL FLOW SHEET FOR THE JIGGING SYSTEM

Scraper conveyors are in greatest favor, the breakage from the discharge of belt conveyors somewhat offsetting their other advantages. Bucket elevators should be used only for refuse and bone, because of breakage, but they are sometimes used on coal where room is limited. Elevation by scraper conveyors is common, a pitch as high as 37 deg. being used in some cases. Shaking chutes, designed similarly to shaking screens, are

efficient, but are confined to horizontal or lowering transport.

The feed to the breaker is by:

(*a*) Hoisting mine car to breaker dump—minimum coal breakage
(*b*) Inclined scraper conveyor—high maintenance
(*c*) Inclined belt conveyor—not suited to lump coal
(*d*) Self-dumping skip—increased breakage over (*a*).

Breaker Flow Sheets and Operation

The drawings, Figs. 1 and 2, illustrate typical flow sheets for the two systems now most widely used—the jigging system, requiring screening before cleaning, and the sand-flotation system, employing screening after cleaning. A Rheolaveur breaker could employ a flow sheet generally similar to the sand-flotation plant, but without the removal of silt before washing.

The flow sheets, with their accompanying legends, are self-explanatory, but it will be understood that the jigs are intended to represent any hydraulic cleaning device requiring screening before washing, such as classifiers and the like, and that various forms of refuse treatment for the recovery of laminated coal and bone may be employed.

The drawings have been prepared as diagrammatic vertical elevations of simple forms of breakers of the two general types. In Fig. 1, a scraper feed conveyor, apron conveyor, picking table, final sizing by removing the large sizes first, and pocket loading are illustrated. Fig. 2 shows a direct mine-car dump, a shaking picking table, final sizing by removing the large sizes last, and loading booms for the prepared sizes. It will of course be understood that all of these individually different devices could be, and in many cases are, used in either type of breaker. Further, hand picking of the steamboat, washing the steam sizes in separate sand-flotation units, special treatment of the bone, etc., while frequently practiced have not been shown in these simplified layouts.

The coal and refuse produced by the breaker are continuously inspected, and the operation of the plant is governed accordingly. In the past such inspection has been generally confined to hand picking and screening, but the float-and-sink method of testing is now coming into use at some collieries.

Every car of prepared coal that does not meet the standard of impurities and sizing is condemned and retreated. The cost of such retreatment (including that due to breakage) will run from $0.60 to upward of $1 per ton, and for this reason every effort is made to reduce condemnation to a minimum.

The labor employed naturally varies with the tonnage, from 70 to 125 men being used at many collieries. The number of men in some of the older breakers has increased in the last few years because of the higher standards to which the coal is being prepared.

That improved methods of preparation can result in marked savings is shown by the operating results of the new plant of the

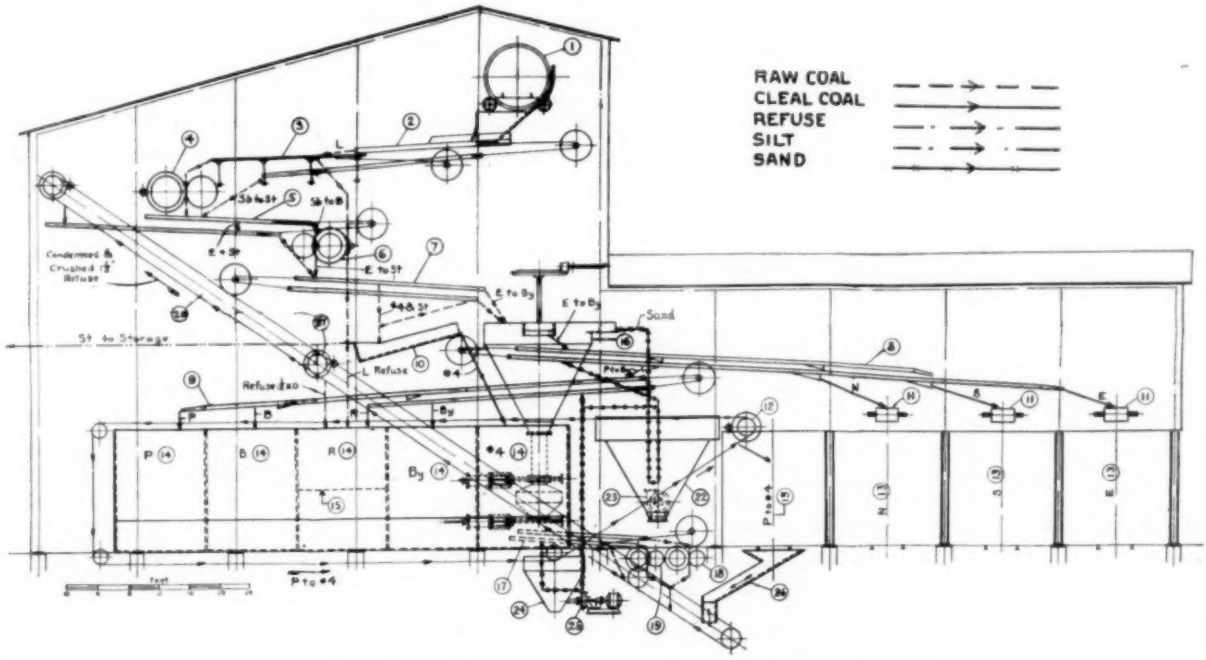


Fig. 2 Typical Flow Sheet for the Sand-Flotation System

St. Clair Coal Company, which replaced an old jig breaker designed some years ago. These are as follows:

Per cent of prepared sizes—increase of 6 per cent
Prepared yield—stationary, but with less refuse in coal
Total yield—stationary, but with less refuse in coal
Steam sizes ash—decrease of 10 per cent
Labor—decrease of 53 men.

The old breaker had a building volume of 1,260,000 cu. ft. and employed 80 to 90 men, depending on the preparation required. The new one has a volume of 492,000 cu. ft. and employs 27 men.

Water Supply

The water requirement of anthracite breakers is about 1 gal. per min. per ton shipped per day, although this quantity may be exceeded if care is not exercised in operating car-washing pits, lip screens, etc.

This water is used for shaking-screen sprays, operating wet cleaners, lip screens, and the like, and as it frequently consists of acid mine water, the wear on machinery is high. In the latest breakers, neutralization of the water, with adequate water recovery, is being introduced, with corresponding savings in first

cost of equipment (it being possible to eliminate much of the acid-resisting equipment now in use) and in renewals.

Breaker Efficiency

If the per cent of merchantable coal, of refuse, and of silt in the feed are known, then the ratio of coal shipped to coal fed would represent the efficiency of the plant as a whole. Such a figure is rarely obtainable because of the lack of accurate tests of the feed, but to indicate what may be expected of a modern well-designed plant, the figures of a 1100-ton test recently conducted are given.

CRUSHING, SCREENING, AND CLEANING EFFICIENCY OF BREAKER AND ROCK HOUSE

Feed of Steamboat (Lump Refuse Previously Removed)

$$\text{Efficiency} = \frac{\text{Shipped free coal on } 1/16 \text{ in.}}{\text{Net feed free coal on } 1/16 \text{ in.}}$$

$$\text{Efficiency} = \frac{81.245}{84.161}$$

Efficiency = 96.5 per cent

Loss in Per Cent of Feed

	Per cent
Silt produced in breaker	2.850
Coal sent to refuse bank	0.066
Total loss	2.916

Weight Reconciliation

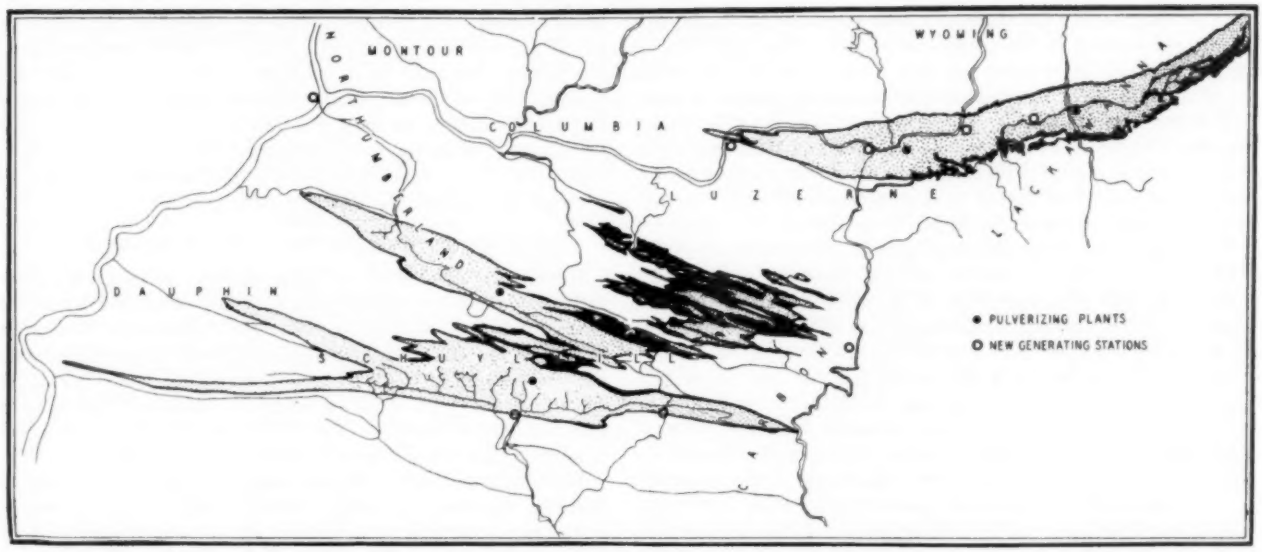
	Per cent
Free coal shipped (egg and smaller)	81.245
Bone shipped with coal	1.309
Slate shipped with coal	1.336
Slate and bone sent to bank[1]	10.894
Coal sent to bank	0.066
Silt in feed	2.300
Silt made in breaker[1]	2.850
Total	100.000

[1] Lump refuse previously removed and feed crushed to steamboat before sending to rock house. All refuse crushed to nut before final rejection, to recover laminated coal and bone.

Discussion

H. E. Walter.[2] Improvements have been made in anthracite preparation during the last 15 years. In 1914 the average breaker had a production capacity of 500 to 700 tons per day. At present a number of breakers will produce in excess of 5000 tons per day, with fewer employees than were required in the old 500-ton breaker. The ash content in domestic sizes of anthracite varies between 8 and 12 per cent as against the old figure of from 12 to 15 per cent, and it is not unusual to find anthracite today with a heating value of 13,500 B.t.u. per lb. dry.

[2] Sales Agent, The Hudson Coal Co., Scranton, Pa.



MAP OF ANTHRACITE COAL REGIONS OF PENNSYLVANIA AND ADJOINING COUNTIES

Economics of Reclamation of Anthracite Silt Coal and Culm

BY FREDERICK H. DECHANT,[1] READING, PA.

The paper deals generally with the production of fine-sized anthracite coal, its preparation for market, and its present uses. It defines several of the most important present markets, and the ultimate essential market to be created. It develops that anthracite can be most cheaply prepared for market by the coal companies themselves, taking in all sizes down to and including pulverized fuel. It recommends complete preparation by the coal-producing companies as the method of attaining the best economic position.

FINE anthracite coal or silt coal is the product of the wet preparation of standard market sizes of anthracite coal. It amounts to from 7 per cent to 13 per cent of the entire production of anthracite depending upon the particular field under consideration; being less in the northern, and greater in percentage of output in the southern fields. Its size ranges from $^{3}/_{32}$ in. diameter to dust, and it ordinarily is mixed with about 25 per cent of non-combustible foreign matter such as slate, rock, etc. as it is discharged from the colliery breaker. About 9,000,000 tons of fine anthracite coal are produced annually. Culm is a mixture of rock and coal, and is the product of the dry preparation of anthracite coal. Culm banks are of early origin, and may or may not contain valuable amounts of coal depending upon the methods employed at the mining operation where they are located.

According to statistics of the U. S. Bureau of Mines, Table 1 gives the quantities of culm, silt, and mixed materials by fields, in long tons.

The large deposits of culm are the result of market conditions which confronted the coal-mining industry of earlier days while their counterpart is now occupied by the deposits of silt; both the result of fundamental economics. This statement is more readily understood when it is realized that within a comparatively few years there has been created a market for anthracite coal in sizes smaller than pea coal, while prior to this time so-called undersized coal was relegated to the dump or culm bank. Since the creation and development of active markets for steam-sized anthracite coals down to No. 4 buckwheat (bird-eye) in size, the silt coal is now being deposited on silt banks at the rate of about 9,000,000 tons annually. Inasmuch as from 60 per cent to 80 per cent of this quantity of material is combustible, it represents an important item in the economics of anthracite-coal production both presently and in the future.

The solution of the problem lies in the development of uses for this fuel, and the resultant creation of markets which will relieve the mine-production costs of the burden of this waste, in proper ratio with its fuel value.

PRODUCTION AND CONSERVATION OF FINE ANTHRACITE COAL

Anthracite silt coal and coal from old culm banks is reclaimed by hydraulic processes of various nature. Briefly these processes depend upon the difference in specific gravity between the coal itself and the fluid in which it is in suspension. The fluid may be water, water mixed with rock dust and coal, water and sand, or water and oil. Various mechanical devices are employed in the operation of the processes, but the result achieved

TABLE 1 QUANTITIES OF CULM, SILT, AND MIXED MATERIALS BY FIELDS, IN LONG TONS

Field	Culm	Silt	Mixed	Total
Southern	37,745,000	36,815,000	10,000,000	84,560,000
Western Middle	43,785,000	40,735,000	17,175,000	101,695,000
Eastern Middle	2,430,000	6,200,000	1,385,000	10,015,000
Northern	8,125,000	8,035,000	1,795,000	17,955,000
Total for all fields	92,085,000	91,785,000	30,355,000	214,225,000

[1] W. H. Dechant & Sons, Consulting Engineers. Mem. A.S.M.E.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

by them is generally the same, although some are better adapted to the reclamation of the finest sizes of coal. The trade names of the processes in common use are: Chance process, the principle of which is the flotation of coal on a fluid mixture of sand and water in which slate and other refuse sinks; the Rheolaveur process using water as the carriage medium, and the Rheolaveur box for coal and slate separation; the Hydrotator using a fluid mixture of coal and water, and sometimes oil with counter water currents in sequential stages of separation.

Combined with these processes of separation there are three well-known thickening devices for the concentration of the coal slush, and the separation of it from its carriage water, namely, the Hydrotator, Hardinge, and the Dorr thickeners. Undoubtedly, one of the most difficult problems in the preparation of fine anthracite is its dewatering, but the simple piling of the prepared silt in protected conical banks as it is produced, and then the using of the material from such storage piles will cause a sufficient reduction in moisture content by gravitational action to justify the rehandling of the material.

It is reasonable to conclude therefore that there are today well-known methods, and the machinery for the accomplishment of the methods has been sufficiently developed to make practical the recovery of 95 per cent of the combustible material produced in the mining of anthracite coal, and for reclaiming the combustible material contained in existing culm banks.

Present Uses of Anthracite Silt Coal and Culm

Statistics reveal the fact that of the 9,000,000 tons of fine anthracite produced annually only 14 per cent is now being used, while the balance is being dumped, used for mine filling or permitted to flow directly to the streams draining the coal regions.

The uses now made of fine anthracite coals can be grouped into: (*a*) Direct-stoker firing under boilers; (*b*) pulverized and used as boiler fuel; (*c*) agglomerated into fuel briquettes with a binder; and (*d*) as fuel in the metal trades for such processes as sintering, and for the washing of foundry molds. These four groups combined show a present consumption of fine-anthracite coal of about 1,200,000 tons per year divided approximately as shown in Table 2.

TABLE 2 COAL CONSUMPTION IN TONS PER YEAR DIVIDED INTO FOUR GROUPS

Group (*a*)	375,000
Group (*b*)	500,000
Group (*c*)	300,000
Group (*d*)	25,000
Total	1,200,000

Screen tests indicate that an average of 30 per cent of all silt coal produced and in silt banks in the anthracite region will be retained on a $^{3}/_{64}$-in. mesh screen, and will contain particles ranging from $^{3}/_{16}$ in. down to $^{2}/_{64}$ in. in size. Silt coal of this sizing is suitable for direct firing on traveling-grate stokers such as the Coxe at combustion rates up to 24 lb. per sq. ft. of effective grate surface per hour.

As nearly as can be ascertained, pulverized anthracite coal is always produced from fine silt coal or so-called river coal, none of the larger sizes being pulverized. Several large and efficient pulverized-anthracite power-generating plants are in operation on this fuel giving excellent operating results. Pulverizing is best accomplished by means of the Hardinge type of mill, and combustion rates of 12,000 B.t.u. per cu. ft. of combustion volume are sustained.

Fuel briquettes made of anthracite silt have been more successfully produced since the war as a substitute for fresh-mined anthracite. Much care is needed in their preparation, and at best they constitute a substitute fuel to which the public must be educated gradually, and at great expense. While they may be developed to a greater degree of perfection for domestic use, it is believed they still will occupy a minor position among the uses for this coal, if for no other reason than that they are a substitute, which is valued by the producers at their plants in Pennsylvania at an average of $6.75 per net ton, and costs the user in Pennsylvania an average of $8 per net ton. Comparatively No. 1 buckwheat and pea coal mixed in equal proportions can be purchased for the same price.

Economic Position of Silt Coal as Boiler Fuel

Based upon the results of investigations at many localities in Pennsylvania and New England, it may be stated that with all relative plant cost, labor costs, and combustion efficiencies accounted for on a comparative basis of plant output, anthracite silt coal refined to 15 per cent ash, and sized from $^{3}/_{16}$ in. to $^{3}/_{64}$ in. inclusive at $3.27 per gross ton compares equally with bituminous low-volatile coal at $3.92 per net ton, at the point of consumption when burned as boiler fuel on stokers.

Pulverized-anthracite silt coal used as boiler fuel in furnaces designed especially for using it indicates that it can be burned at the rate of 20,000 B.t.u. per cu. ft. of combustion space, so that in comparison with stoker-fired silt coal its cost at the point of consumption may be about $1.20 per ton more than the cost of silt coal for direct stoker firing. Under average conditions within the range of freight limitations then, the adoption of pulverized anthracite coal as boiler fuel is more advantageous to both the power producer and the coal producer.

Anthracite silt coal if divided into two grades by screening, after cleaning for reduction in ash content, can be classified as boiler-stoker fuel and boiler fuel for pulverizing. It appears from Table 2 that approximately 7,800,000 tons per year out of the total production of silt is yet to find a market. Manifestly the natural market for this fuel is in the generation of electric power at large central stations, located near the sources of production of silt coal. Assuming that the producers do classify silt coal as suggested, it would then be possible to ship via rail for comparatively long hauls for direct-stoker firing the first grade of silt-coal fuel, or about 2,300,000 tons per year. The cost of producing this grade of fuel should not exceed 35 cents per gross ton at the point of origin, and could be sold for $1 to $1.25 per gross ton at the mines in competition with bituminous low-volatile coal in localities where the differential at the point of consumption was 65 cents or less per ton.

The balance or second grade of anthracite silt coal, i.e., under $^{3}/_{64}$ in. size, constitutes 70 per cent of all silt production, and after the deduction of the groups *a*, *c*, and *d* still leaves 5,400,000 tons per year for which there is now a very small market. A satisfactory market created for this slush, now waste fuel, would to a large measure solve the general anthracite problem, and to this end the author submits the following, in the belief that if carefully and thoroughly studied and practised, it will help the economic position of the industry. It is well known that the fine sizes of anthracite alone cannot be made to pay for themselves, but the greater the yield of marketable coal the less will ultimately be the cost of each ton of production.

A Suggested Program

The great objection to the use of pulverized anthracite coal even in moderately sized boiler plants is the first cost and the operating cost of the drying and grinding machinery. In large installations the cost of these elements is also a big factor. In few cases of the installation of drying and pulverizing equipment does the load factor on the entire operating and standby units exceed 50 per cent, so that the cost of pulverizing and drying is generally abnormal when considered as an element in the cost of operating at a single plant.

It appears to be a practical and economical thing to establish at suitable geographical centers with reference to the mining fields, central-station coal-pulverizing plants for converting the slush coal into marketable pulverized coal. A concept of such a plant might comprise works capable of pulverizing 250 tons of coal per hour. Drying would be accomplished by heat furnished through the combustion of a portion of the fuel. The pulverized coal could be shipped in specially constructed multi-hopper cars of air-tight construction similar to those in use in England, and hauled long distances to points of consumption.

Considering the factors in favor of such a program it is believed that coal, so prepared for use as pulverized fuel, could command a mine price of $2.20 per ton while its cost of preparation should not exceed 60 cents per ton. Prepared pulverized coal would widen the market for its use industrially. Central power stations as well as others could contract for pulverized fuel without the investment in pulverizing plant, and its consequent operation and maintenance. This should be attractive because it would relieve the power-generating organization from the problem of fuel preparation without the loss of its benefits as a fuel.

Centralized pulverized-coal preparation plants can be operated at a higher load factor than individual pulverizing plants, so that the cost of pulverizing and preparation will be reduced below what they now are at individual preparation plants.

The cost of boiler and combustion equipment for burning prepared pulverized anthracite coal is less than for the necessary equipment to burn anthracite fuel in any other way. Such equipment would comprise a storage bin with a capacity equal to the plant consumption over a period of several days or a week, coal conveyors, feeders, and burners; all relatively inexpensive parts of the equipment for burning pulverized fuel.

Pulverized anthracite coal is not subject to spontaneous combustion in storage so that the fire hazard usually present in the pulverizing plant is absent in a boiler plant fired with prepared pulverized anthracite coal.

Considered from the coal producers' standpoint it is not unreasonable to place the burden of complete anthracite-coal preparation from the broken-coal size down to pulverized coal on the mining companies, because by so doing they can effect a higher yield per ton of mine output, stabilize their output, and reduce the burden of expense generally on sized anthracite coals.

It is believed that a product such as silt coal is so different from any of the coals now handled by the railroad companies, also since it represents an additional tonnage, that it would be reasonable to expect that an adjustment should be made in the freight charges on this type of fuel, which would widen its market.

Conclusion

This paper has not been intended to deal with either the chemistry or thermodynamics of the combustion of anthracite silt coal, but rather its purpose is to define a means of converting the present average 10 per cent waste anthracite mining into marketable fuel.

It is the author's belief that this problem is fundamentally one of coal preparation, which should properly be assumed by the coal-producing companies primarily for the benefits to be derived.

Tentative locations are given on the attached map of the anthracite coal regions for proposed central pulverizing plants, and also there are indicated locations for new central power stations owned and operated by the power-generating companies having jurisdiction in the regions, which could be economically operated on prepared pulverized anthracite coal.

Generally the process that will convert fine anthracite coal into a marketable product at the least cost to the producer will ultimately be the most profitable.

Discussion

Frederick L. Moore.[2] There is without question an increasing demand for fine-size anthracite coal, and one source of supplying this demand is from the accumulated silt and culm banks and waste accumulations from the coal fields. This demand for anthracite fine sizes is due partly to the economic aspects of a cheaper fuel. The finer sizes of anthracite originally went on the market as a by-product without preparation, whereas today all sizes are prepared with equal care, and therefore contain equal heat value per pound of product. Therefore the consumer in a sense is getting more than he is paying for as the highest price of any of the anthracite fines is below the cost of production. This demand for the smaller sizes is also based on the possibilities of better combustion control and automatic features. It is true that the operator will have to produce what the consumer demands.

Some years ago lump coal was sold. Today no one seeks lump coal and none is sold. Likewise the demand for grate size is falling off, and there is practically no market for it. Egg size will soon follow the others. It becomes necessary therefore to crush these to finer sizes and produce a larger percentage of buckwheat and silt.

There seems to be considerable education required to teach the public how to burn anthracite properly. The public today is demanding close sizing of anthracite when such is not necessary nor economical and efficient. Anthracite can be burned successfully in all sizes, and the writer does not believe that briquetting will be practiced to any marked extent. Up to today it has not reached very large proportions.

That size designated as silt between $^3/_{16}$ and $^3/_{64}$ in. is now prepared and marketed as buckwheat No. 3 and No. 4, and not strictly silt. Correctly speaking silt is that below $^3/_{64}$ in. size mesh, and is being prepared today down to 60 mesh. The oversize in the present silt banks would be due to spillage from a broken or worn jacket or screen, and in that case would be but a small percentage of the reclaim. It is likely also to be the refuse from the larger sizes, and in that case would possibly be of low heat value. The finer the size the greater is the ash content, and below 60 mesh it does not at present seem economical to attempt to reclaim such fines. The use of such fines in all probabilities should be confined to or near the field of production due to the added cost of transportation. The writer does not believe that the very fine coal will ever be transported to distant points in great amounts.

Pulverized anthracite is being successfully produced and consumed in the southern field, but there is some question as to successfully and economically pulverizing the coal in the middle and northern fields due to its harder nature.

This is an age of great changes, and great changes are facing the anthracite industry. Just what the ultimate situation will be no one can tell exactly. Some of the trends are visible today, and it is well to follow these leads and prepare for the future. He who looks the furthest ahead will be the wisest.

The tendency today is toward greater use of fine sizes both by the commercial and household user due to the economical automatic feature of combustion control.

There is a certain amount of evolutionary process involved which is slow in progress and must be allowed to take its time and course. Education can hasten or deter this action. It has even been mentioned that coal be sacked the same as ce-

[2] Scranton Coal Co., Scranton, Pa. Mem. A.S.M.E.

ment. Just how and when this will be done no one can tell.

In all probability coal will be used more at or near the mine for the generation of power, possibly along the Susquehanna and Delaware rivers, and the power transmitted to the metropolitan areas. There should be no fear of a degenerated product in reclaimed fuel from waste banks as there is practically no deterioration in stored anthracite.

In one case the reclaimed fuel from an old bank tested over 14,000 B.t.u. while the fresh mine fuel from the same mine but different veins tested but over 12,000 B.t.u. This test was checked due to its apparent contradiction.

James A. Powell.[3] Much credit is due Mr. Dechant for the keen interest that he has manifested generally in the question of waste anthracite fuel. The writer is in accord with Mr. Dechant that fine sizes of anthracite coal should be prepared at the point of shipment, but questions just how far this preparation should be carried.

It is well known that silt and culm have from 10 to 15 per cent moisture which must be driven off before pulverizing, and it is certainly uneconomical to pay freight on this amount of water. Dryers can be installed at the mine, and operated cheaper than at the power plant. The writer therefore agrees entirely that the drying should be done at the point of shipment. It is perfectly feasible to pulverize the coal after drying at the point of shipment, and transport it in special cars, but this is not the most economical method. Pulverized-fuel bunkers large enough to take care of varying-load demands and irregularities in deliveries would be very expensive.

A suggested method which appears to be more economical is: (1) To dry the coal before shipment down to $1^1/_2$ to 2 per cent moisture, and then to ship in special hopper-bottom cars with a roof for keeping out moisture. (2) Coal should be stored in closed-type conventional raw-coal bunkers with capacity large enough to take care of irregularities in shipment. (3) Mills should take coal from raw-fuel bunkers and discharge to powdered-fuel bins as desired.

The pulverized-fuel installation would be no more expensive at the point of use, and as the power cost is of considerable importance, this would favor the milling at the point of use due to cheaper power cost.

A twelve-ton mill is now in use with river anthracite coal, moisture approximately 10 per cent, sand 10 to 12 per cent, with a power consumption of 30 kw-hr. per ton, and a maintenance of approximately two cents per ton which is less than other mills in this same plant using bituminous coal. This installation has demonstrated that anthracite can be dried and milled at a cost which is not prohibitive.

In regard to combustion equipment, this is a special study, which may be summarized as follows:

Today boiler-room equipment is more expensive for anthracite fuel than for bituminous, whether fired by stokers or in a pulverized form.

If a satisfactory and economical manner of preparing pulverized refuse anthracite is available, certainly development in the art of burning with a reduction in boiler cost will follow, such as higher rates of combustion by the combination of a small stoker for ignition with variable loads with very low volatile coal, or some other method.

Mr. Dechant's paper should stir up the operators to see that the silt and culm banks are not at present satisfactory as fuel for pulverized-fuel plants, and when a finished or semi-finished prepared coal can be offered to the utility field at a known price, then the power-plant designing engineer can work out the possibilities of such a fuel.

The writer understands from Mr. Dechant's paper that there are 5,400,000 tons annually available for pulverized fuel for electric generation. Assuming that a kilowatt-hour can be generated on two pounds of coal, this would give an output of 5,400,000,000 kw-hr. yearly, and with an average load factor of 50 per cent would require an installed capacity of 1,230,000 kw. to consume this fuel.

The author has selected eight station locations to consume this coal. This would mean an average of more than 150,000 kw. capacity stations. The writer does not believe that the territory shown on Mr. Dechant's map could absorb more than 300,000 kw. capacity today, so that it would appear that it will be many years before the annual electric generation would equal the silt and culm production unless all of the mines and railroads would electrify.

It is interesting to note that the public utilities for the entire state of Pennsylvania only used the equivalent of 6,700,000 tons of anthracite coal during the year of 1927 for fuel to produce electric generation.

Assuming a station cost of $100 per kw. the utilities would have to invest more than $120,000,000, and the coal operators between $7,500,000 and $15,000,000, depending upon whether the coal is simply dried or dried and pulverized. No estimate has been made of the expense necessary for special cars for transporting either dried or pulverized fuel.

The writer hopes that Mr. Dechant's paper will be an incentive for further study and work on this important subject so as to provide an economical solution for the important subject of refuse anthracite coal.

Author's Closure

It should be understood by all readers of this paper and the comments thereon that the incentive to prepare the fine sizes of anthracite for market lies in the possibility of converting this present-day loss to the mining companies into a highly profitable product with indications that the production of pulverized fuel presents the most logical course of accomplishment.

It is interesting to note that the utility companies will apparently be interested in prepared fuel of this nature so that there is a definite market awaiting developments along the line suggested.

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[3] W. S. Bartow Management Assoc., Inc., Reading, Pa. Mem. A.S.M.E.



FIG. 1 TYPICAL RECORD BY THE LEEDS AND NORTHRUP DAYLIGHT RECORDER, USING THE CASE PHOTOELECTRIC CELL
(The record, which reads from right to left, was taken on October 25, 1928; the weather was clear until about 11:30 a.m. From 11:30 a.m. to 3 p.m. there were occasional clouds; from then until dark the sky was somewhat overcast.)

Atmospheric Pollution and Sunlight

BY PHILIP DRINKER,[1] ROBERT M. THOMSON,[1] AND SARAH P. CHOATE,[1] BOSTON, MASS.

Records of daylight taken in Boston over a period of 16 months are correlated with dust counts, wind velocity, and per cent relative humidity. Results indicate a slight inverse relationship between dust counts and daylight (coefficient = −0.1861) and dust counts and wind velocity (coefficient = −0.1501). At the recording station on the southern edge of the city 68 per cent of the maximum daylight possible was received. Although conclusive proofs are lacking it seems probable that natural causes more than smoke pollution account for this transmission figure. Measurements of solar ultraviolet light by the acetone-methylene-blue method are plotted for six stations in Massachusetts. The degree of atmospheric pollution in the vicinity of the Boston recording station is not sufficient to destroy the therapeutic value of sunshine, even during the winter months.

DURING the past two years certain of the therapeutic and physical aspects of heliotherapy have been studied by the Department of Ventilation and Illumination, Harvard School of Public Health, in cooperation with the Department of Pediatrics of the Harvard Medical School and the Children's Hospital, Boston. The authors' part in the study has consisted in securing continuous records of visible daylight, records of atmospheric dust from air samples taken on the roof of the Harvard School of Public Health, and intermittent records of solar ultra-violet light inside and outside the neighboring Children's Hospital building, and at five other stations in Massachusetts.

Both ultra-violet and visible light, whether from the sun or from artificial sources, are known to be of importance in clinical medicine. Industry has been accused of so polluting the atmosphere with smoke that an appreciable part of the solar radiation to which we are entitled by nature never reaches us. If this is true, it is an important argument in favor of smoke abatement; if it cannot be demonstrated, efforts toward smoke abatement must be based in more general terms on the greater economy, convenience, and beauty of clean cities.

RECORDS OF VISIBLE DAYLIGHT

Records of visible daylight were taken by means of a Case photoelectric cell connected to a Leeds and Northrup recording potentiometer. This daylight recorder is similar to the one used by Ives in his investigations and has been fully described by him.[2]

The spectrophotoelectric curve of the Case cell agrees closely with the visibility curve of the human eye. The manufacturer claims that the photoelectric current generated in the cell is proportional to the intensity of the light incident to the cell.

The recording potentiometer, to quote Ives, is "merely a mechanism for making a chart record of a difference of potential between two points in a current circuit." The cell was connected across the potentiometer with a variable resistance in shunt so that the magnitude of the response of the recorder might be varied as desired. Fig. 1 shows a typical record for a bright October day with occasional clouds. Records of the short days in December and the long days in June are similar in type but differ in the area enclosed by the curve. By comparing points on these records for both sunny and cloudy days with actual determinations of daylight measured by the Macbeth illuminometer, a curve such as that of Fig. 2 is obtained.

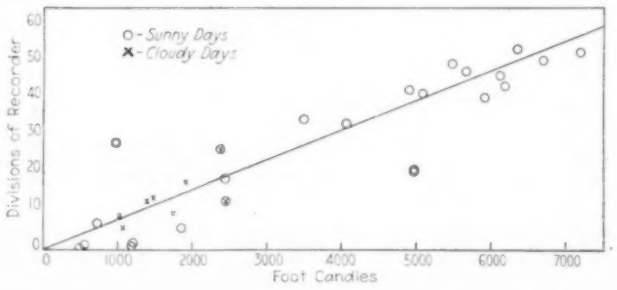


FIG. 2 RELATIONSHIP BETWEEN ACTUAL FOOT-CANDLES OF DAYLIGHT ILLUMINATION, TAKEN BY MACBETH ILLUMINOMETER WITH DAYLIGHT FILTER, AND RESPONSE OF THE DAYLIGHT RECORDER
(From the slope of the plotted line, the foot-candle-hour values of Fig. 3 were computed.)

The Leeds and Northrup recorder is driven by a very accurate constant-speed motor, so that the time intervals, abscissas in Fig. 1, are uniform and reliable. Consequently, if these records are integrated with a planimeter, the areas may be expressed in "foot-candle hours," a term devised for this particular study, whose meaning is obvious. Fig. 3 shows the results of records taken over some 16 months and calculated in this way.

The seasonal character of daylight variation is apparent on both the maximum and average curves. The maximum may be regarded as the theoretical curve obtained if precipitation (rain, snow, and fog) occurred only at night, and the days were all clear and sunny with occasional fleeting clouds and a 15- to 20-mile breeze. The average curve represents a mathematical

[1] Department of Ventilation and Illumination, Harvard School of Public Health.

[2] Records of Daylight by the Photoelectric Cell, by J. E. Ives, Transactions of the Illuminating Engineering Society, 1925, vol. 20, p. 498.

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moving average, in three-week periods, of the actual figures obtained.[3]

Atmospheric Dust Counts

By means of Owens' Jet Dust Counter[4], the dust particles in small measured volumes of air were collected twice daily on the roof of the Harvard School of Public Health, at 10:30 a.m. and 3:30 p.m., throughout the period covered by the daylight records. Owens' instrument was described at the first national meeting of the Fuels Division of this Society,[5] and is well known in this country and in England. It is especially well adapted to the estimation of particles of the order of 0.5 to 5 microns in size, particles small enough to float about in the air.

The actual points, or counts, and the three weeks' moving average values, calculated in the same way as the daylight averages in Fig. 3, are shown in Fig. 4. The seasonal character of this curve is apparent, and also certain cyclic phenomena not seen in the daylight curves. These cycles, appearing as humps in the curve, reoccur in an extension of the curve not shown in the present paper. It should be remembered that these data, also, are true only for the particular place where the observations were made.

Records of Solar Ultra-Violet Light

Solar ultra-violet radiation can be satisfactorily measured, but the methods now available are, for the most part, too elaborate and the equipment too expensive for general use.[6] There is,

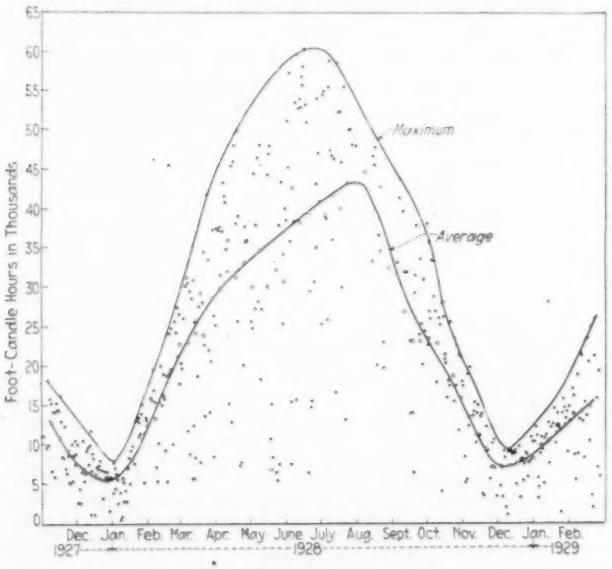


Fig. 3 Curves Showing Maximum Theoretical and Actual Average Daylight Illumination in Foot-Candle-Hours Measured From the Roof of the Harvard School of Public Health, Boston

unfortunately, no constant relationship between visible and ultra-violet solar radiation, so that ultra-violet cannot be estimated from the curves in Figs. 1 and 3. For practical reasons, therefore, a simple photochemical reaction was adopted, the bleaching of aqueous-acetone-methylene blue, suggested by Webster, Hill, and Eidinow,[7] and used by several investigators in this country.[8] A small clear quartz tube is filled with solution of methylene blue and exposed to the sun. After a suitable time interval, the tube is replaced by a fresh one, and the old tube matched against a set of fixed standards. The standards are graded in "units of methylene blue decomposed." Although this scale cannot be translated into any standard physical or

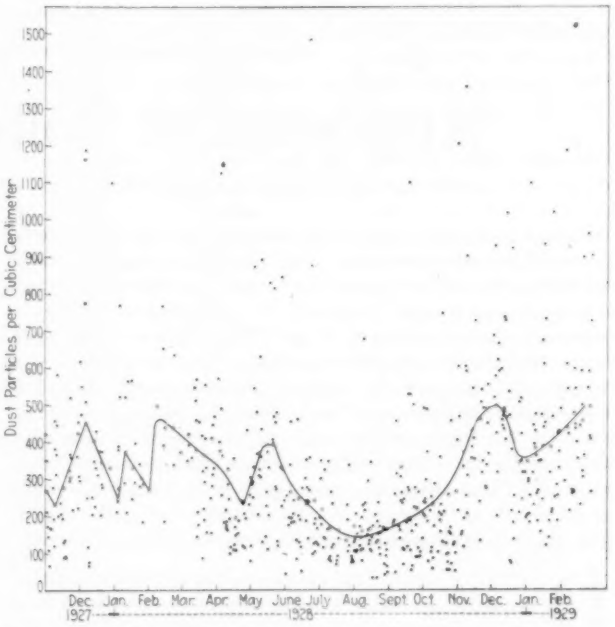


Fig. 4 Dust Particles Per Cubic Centimeter of Air as Shown by Owens' Jet Dust Counter

(Samples were taken twice daily over the periods covered from the roof of the same building in which the daylight recorder was installed.)

photometric units, it is useful in grading exposure of patients to solar ultra-violet light.

The averaged results of observations made at five stations in different parts of Massachusetts, as compared with the results from Rutland alone, are shown in Fig. 5. It would appear that Rutland, at an altitude of 1200 ft., is the station best adapted for ultra-violet treatments.

This method of estimated solar ultra-violet radiation is open to serious criticism on several grounds, chief of which is the fact that the reaction is, to some extent, caused by wave lengths other than those which are physiologically significant. The reaction, moreover, is somewhat affected by temperature. In spite of these objections it is felt that, for the purpose of the present study, the use of the method was justified.

Standard Meteorological Data

From the Boston station of the U. S. Weather Bureau, daily

[3] It would be unwarranted to assume that these figures are applicable to other cities, or even to the downtown districts of Boston. The Harvard School of Public Health is on the outskirts of Boston, and the prevailing wind is favorable to the locality.

[4] Jet Dust Counting Apparatus, by J. S. Owens, *Journal of Industrial Hygiene*, 1922–23, vol. 4, p. 522.

[5] The Measurement of Atmospheric Pollution, Visible and Invisible, by G. T. Moore, *Mechanical Engineering*, Oct., 1927, vol. 49.

[6] Ultra-Violet Solar Radiation, by Edison Pettit, Proceedings of the National Academy of Sciences, 1927, vol. 13, p. 380.

[7] Measurement of Ultra-Violet Light by Means of an Acetone-Methylene-Blue Solution, by A. Webster, L. Hill, and A. Eidinow, *Lancet*, Apr. 12, 1924, vol. 1, p. 745.

[8] A Preliminary Report of the Measurement of Variation of Energy in the "Vita Spectrum" of the Sunshine in Kansas, by J. S. Hughes and R. L. Pycha, Transactions of the Illuminating Engineering Society, 1928, vol. 23, p. 233.

See also: "A Simple Method of Measuring the Intensity of Ultra-Violet Light With Comparative Results on a Number of Physiological Reactions," by J. H. Clark, *American Journal of Physiology*, 1924, vol. 69, p. 200; and "The Zinc-Sulphide Method of Measuring Ultra-Violet Radiation and the Results of a Year's Observations on Baltimore Sunshine," by J. H. Clark, *American Journal of Hygiene*, 1929, vol. 9, p. 646.

records were obtained of wind velocity, temperature, per cent relative humidity, and barometric pressure at times corresponding approximately to the hours when the dust counts were taken. The building where the counts were taken is about two miles southwest of the Weather Bureau station and at the same general altitude. It is reasonable to assume, therefore, that the Weather Bureau data can be applied without introducing any appreciable error.

Relation Between Daylight, Dust Counts, and Meteorological Data

Although this study was actually carried out over a two-year period, continuous data for 16 months only are used here. These data are sufficient to permit a statistical interpretation of the results and to indicate which variations are due simply to chance and which to definite physical causes.

The amount of daylight recorded depends, of course, on several factors other than dust counts. Of these, wind velocity and per cent relative humidity seem to be the most important. The effect of dust counts on daylight is of especial interest. Do these factors show a definite inverse dependence or relationship, or are other factors like relative humidity and wind velocity of greater importance?

It is possible by statistical methods[9] to determine the coefficient of partial correlation between daylight and dust counts by holding constant the variables, wind velocity and per cent relative humidity, thus eliminating the variability of wind velocity and relative humidity without actually ignoring these factors themselves. This mathematical procedure may be approximated by selecting from the tabulated data a series in which humidity and wind velocity are approximately constant, but dust counts and daylight vary. This course, however, although the more obvious one, is less accurate than the method used and equally laborious.

For daylight data the ratio of the actual values obtained on the recorder to the maximum curve of Fig. 3 was used instead of the average curve. On a bright day this ratio would be close to unity, on a dark day it would be very low regardless of the season of the year. This ratio was used in order to minimize the seasonal character of daylight variation, which otherwise might have had a quite misleading effect. The results are shown in Table 1.

TABLE 1 COEFFICIENTS OF PARTIAL CORRELATION

$r_{12.34} = -0.1861$ (count and per cent maximum daylight)
$r_{13.24} = -0.1501$ (count and wind velocity)
$r_{14.23} = -0.0357$ (count and per cent relative humidity)

The subscripts of r indicate the variable involved, those to the right of the period being held constant in accordance with the following: Count = 1, sunlight = 2, wind velocity = 3, and per cent relative humidity = 4.

According to standard statistical practice, these coefficients show a slight inverse relationship in the first two cases, whereas the relationship in the third is too low to be worth considering. Perfect correlation would, of course, give a coefficient of 1. The results indicate, therefore, that as dust counts increase daylight decreases somewhat, and that a high wind reduces dust counts to some extent.

Per Cent Transmission of Daylight

The ratio of the areas under the average and maximum curves of Fig. 3 is 0.84. Instead of using the average data, however, it is more accurate to use the summation of the ratios between the actual points and the maximum curve, as they were used in obtaining the correlation ratios. This summation gives a ratio of 0.68; that is, at this particular station, 68 per cent of the maximum daylight theoretically obtainable was reached. If a daylight recorder were installed 10 or 15 miles outside the city limits and another in the downtown section, values between the two would probably be recorded at the station where the tests were made. The differences then might be ascribed to man-made pollution and not to natural causes.

[9] An Introduction to the Theory of Statistics, by G. U. Yule. Philadelphia: J. B. Lippincott Co., 1922.

Observations

Any dense cloud of smoke, either black or white, can shut off an appreciable amount of daylight, both visible and ultra-violet radiations. Theoretically, the smaller particles (0.5μ) are the more effective in shutting off the short wave lengths of solar ultra-violet (0.35μ to 0.4μ). Particles from volcanic eruptions or explosions, like that of Krakatoa[10] in 1883, are of the order of 1.85μ, particle size being determined by the effect upon daylight.

A few years ago measurements were made by this laboratory of the size of the particles emitted from the stacks of a large power plant burning powdered coal. The samples were collected from the top of the stack at about the middle point. Under the microscope these particles were found to be approximately spherical, of glassy appearance, and somewhat pitted; among them, an occasional black particle, presumably of coke or unburned coal, occurred. These particles compared in size with the volcanic ash, most of them being less than one micron in diameter. The temperature and velocity of their carrying gases at emission from the top of the stack were, however, in no way comparable to those of a volcanic explosion. Under present methods of power-plant operation it is quite impossible for the particles to be carried for great distances.

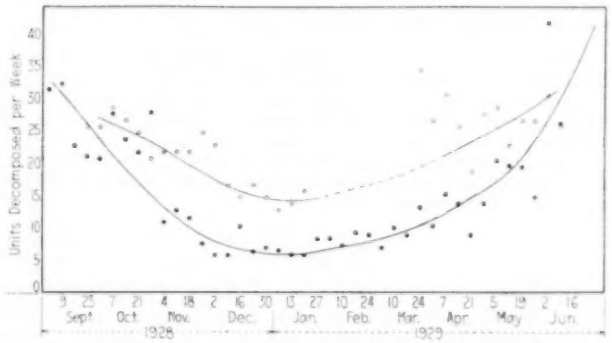


FIG. 5 RECORDS OF SOLAR ULTRA-VIOLET LIGHT BY THE METHYLENE BLUE GAGE

(Data from five stations in Massachusetts which showed values of the same order were averaged and plotted in one curve; data from Rutland, which showed consistently higher values—probably due to altitude—were plotted separately.)

Nevertheless, the ash emitted from the chimneys of some plants burning powdered coal is reaching such amounts that it must be measured in tons rather than pounds per day. The color of this ash is so light that a dense cloud of it does not necessarily appear offensive in terms of the Ringelmann chart. Obviously, then, a different method of judging atmospheric pollution from powdered-coal ash is desirable.

In a recent paper, Shrader, Coblentz, and Korff[11] state: "The belief that atmospheric pollution actually occludes ultra-violet light to a very great extent is well founded." They found that the ultra-violet transmission was greater at the tops of tall buildings than at street level, that it was greater in the country than

[10] Physics of the Air, by W. J. Humphreys, p. 572. Philadelphia: The Franklin Institute, 1920.

[11] Effect of Atmospheric Pollution Upon Incidence of Solar Ultra-Violet Light, by J. H. Shrader, M. H. Coblentz, and F. A. Korff, *American Journal of Public Health and the Nation's Health*, 1929, vol. 19, p. 717.

in the city, and that it was, of course, greater on bright sunny days than on dark cloudy days. They do not attempt to show whether health is affected by the amount of pollution observed in Baltimore.

Dr. Edwin T. Wyman[12] has demonstrated the effectiveness of applying solar ultra-violet to infants throughout the winter months in the Boston Children's Hospital. In December a much longer exposure is necessary than in September or March, but the treatment has been found to be so practical a therapeutic measure that a large solarium equipped with "Cel-O-Glass" windows[13] is being installed on the top floor of one of the new hospital buildings. The degree of atmospheric pollution occurring in this vicinity, therefore, is not sufficient to destroy the therapeutic value of sunshine. To what extent it reduces the effectiveness of sunshine and makes longer exposures necessary cannot be definitely determined from the facts now available.

Tisdall and Brown[14] believe that rickets occur rarely in localities where the height of the sun at noon exceeds 35 deg. For approximately 12 weeks, in November, December, and January, the noon sun in Boston is below 35 deg., and it is during that period that the Children's Hospital will be prepared to supplement natural with artificial solar ultra-violet radiation. Even were there no atmospheric pollution, this condition would probably not be significantly altered.

[12] The Prevention and Treatment of Rickets, by E. T. Wyman, *Boston Medical and Surgical Journal*, 1927, vol. 197, p. 376.

[13] Cel-O-Glass is cheaper than other quartz substitutes; it transmits solar ultra-violet almost as effectively (about half as well as pure quartz), and it has a low deterioration factor.

[14] Relation of the Altitude of the Sun to its Antirachitic Effect, by F. F. Tisdall and Alan Brown, *Journal of the American Medical Association*, 1928, vol. 92, p. 860.

Kimball and Hand,[15] and Owens [16] have demonstrated that an inverse relation, non-linear in character, exists between dust counts and wind velocity, the dust counts being highest on still days. The data used in both these investigations were comparatively few and represent extreme rather than routine conditions. In the present study some 550 pairs of values for dust count and wind velocity were secured. The day to day relationship is expressed by the low partial correlation coefficient, —0.1501. The difference between this value and those indicated by the previous investigators are probably explained by these differences in procedure. Had selected data been used the results given in this paper would have been more nearly like those cited.

It is clear that we suffer in this country from lack of adequate comparative data, routinely taken under similar conditions, and by similar methods in our leading cities. The Advisory Committee on Atmospheric Pollution, first organized under the Air Ministry and later under the Department of Scientific and Industrial Research, has provided an efficient service for the British Isles. The Weather Bureau would be well equipped to make similar observations at its different stations throughout the United States, and could easily undertake the work if the various groups and associations interested in atmospheric pollution would unite to initiate the necessary legislation in Congress.

The authors wish to acknowledge their obligation to Laura A. Cauble of the National Conference Board on Sanitation for her generosity in supplying the daylight recorder used in this study.

[15] Investigation of the Dust Content of the Atmosphere, by H. H. Kimball and I. F. Hand, *Monthly Weather Review*, 1925, vol. 53, p. 243.

[16] Eleventh Report of the Advisory Committee on Atmospheric Pollution, p. 42. London: H. M. Stationery Office, 1925.

FSP-52-9

Economic Status of Anthracite and Factors Affecting Its Use as a Domestic-Heating Fuel

By A. F. DUEMLER,[1] NEW YORK, N. Y.

Pennsylvania anthracite as a smokeless fuel for domestic-heating purposes occupies an important economic position, and the producing fields are located central to the most densely populated area of the United States. The anthracite industry has for the past several years been engaged in extensive research pertaining to the utilization of this fuel for domestic heating. This work is being carried on in three stages, as follows:

1 By direct service contact with the consumer in an extensive survey to determine and classify factors affecting efficient utilization.

2 By laboratory research to determine relative efficiencies of various sizes of anthracite as one size or varying mixtures of sizes.

3 By laboratory research for the development and testing of equipment which will provide greatest convenience and efficiency in utilization.

The research work in the first stage has been particularly valuable in that it brings out the factors to be shared and considered by interests other than the anthracite industry.

TO CONVEY an idea of the important economic status of anthracite as a smokeless domestic fuel one needs only to refer to tabulations available through the U. S. Census Bureau and the U. S. Department of Commerce, which show the population of the country by states and the land area occupied by that population. This information can be reduced to density of population per square mile of land area. Then compare this information, as it applies to the eleven New England and Middle Atlantic States and District of Columbia, with the country as a whole, and then locate the geographic position of the anthracite fields in northeastern Pennsylvania.

TABLE 1—NET LAND AREAS AND POPULATION OF NEW ENGLAND AND MIDDLE ATLANTIC STATES

(U. S. Department of Commerce estimate as of July 1, 1928)

State	Rank in net land area	Net land area, square miles	Population	Population per square mile
New York	29	47,654	11,550,000	242.3
Pennsylvania	32	44,832	9,854,000	219.7
Maine	38	29,895	795,000	26.6
Maryland	41	9,941	1,616,000	162.5
Vermont	42	9,124	352,428	38.6
New Hampshire	43	9,031	456,000	50.4
Massachusetts	44	8,093	4,290,000	530.0
New Jersey	45	7,514	3,821,000	508.5
Connecticut	46	4,820	1,667,000	345.8
Delaware	47	1,965	244,000	124.1
Rhode Island	48	1,067	716,000	671.0
District of Columbia	49	62	552,000	8903.2
Total		174,037	35,913,428	

Population per square mile (35,913,428 ÷ 174,037).....206.3

Compare this figure of 206.3 persons per square mile in the district covered in Table 1 with the figure of 35.5 per square mile for the entire country.

Until five years ago anthracite as a domestic fuel in the area outlined was without competition, and the questions of smoke and air pollution due to domestic fuel consumption were negligible ones. However, the entrance of other fuels into this market during the last few years has developed a new era in the preparation, marketing, and utilization of anthracite which has proved profitable to the public as well as to the anthracite industry. The quality of the product being sent to market has been improved greatly through the extensive development of mechanical-cleaning methods.

Service to the consumer has been extended widely, not only in the Middle Atlantic and New England states, but as far west as Milwaukee, St. Paul, and Minneapolis. This service from the anthracite producers is being carried on principally through several thousand retail dealers as a part of their present merchandising methods.

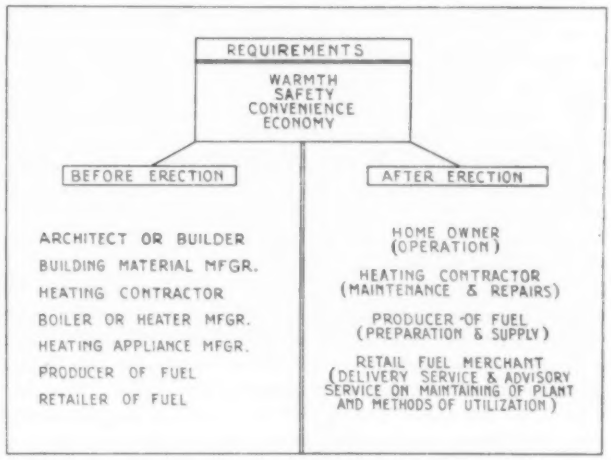


FIG. 1 FACTORS INVOLVED IN DOMESTIC-HEATING COMFORT

Utilization by the consumer has been improved greatly through this service, and other educational methods and records show that the anthracite industry was first as a fuel industry to develop a coordinated plan of operation covering the phases outlined.

Being first in planning such an operation meant that there were very few facts available to use as a guide. Therefore much had to be done in the nature of research and experiment, and the development of this work again brings to light the fact that well-prepared anthracite and a willingness to serve are not the only factors effecting efficient utilization or complete satisfaction to the home owner.

Some idea of this last statement may be gained through Fig. 1, charting the factors involved in the business of domestic-heating comfort. This indicates that the problem is not altogether unlike that encountered by the consumer of fuel for commercial purposes.

However, the consumer of fuel for commercial or industrial purposes has made it his business, through the aid of the engineering profession, to reap every possible benefit from the research and development of methods and equipment whereby the greatest return is realized for every fuel dollar expended.

Much credit is due the manufacturers of industrial boilers, furnaces, and auxiliary equipment in that every step taken by them seems to lead to higher efficiencies, resulting in conserva-

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Presented at the Third National Fuels Meeting, Philadelphia, Pa., Oct. 7 to 10, 1929, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

tion of fuel and labor. Likewise these equipment manufacturers are to be commended for their merchandising methods by which the industrial consumer, again through the services of the engineer, is able to purchase equipment of proper design and rating to perform efficiently a given duty.

On the other hand, the consumer of fuel for domestic-heating purposes is seriously handicapped in making decisions in the purchase and installation of heating equipment, a part of the home having much to do with the joy of living. It is not meant to convey the idea that the heating and ventilating business has existed without engineering research and development. Much in that respect has been done, but most of this research and development has until the past few years been largely confined to details of the heating system other than those effecting the actual burning of the fuel and utilization of heat in the combustion chamber of the boiler or heater. Also it must be discouraging to the profession to see the benefits of research and development almost entirely cancelled through misapplication due to competitive merchandising methods whereby the dollar mark of first cost very often clouds all other considerations.

Viewed strictly from an economic standpoint, the home-owner rarely is justified in going to the expense of calling in a consulting or heating engineer to solve his problem, and the result is that an ill-purchased heating plant refuses to heat the home regardless of how much or what kind of fuel is consumed. The plant might furnish heat after a fashion, but a few dollars saved in first cost generally results in what can be fairly called an annual heat tax in the form of excessive fuel requirements plus unsatisfactory operation. This condition in turn causes dissatisfaction and inconvenience, with the final result that changes to other fuels are often made in an attempt to accomplish what might well be termed the impossible.

If it is reasonable to assume that the problem is outside the range of the engineer to deal direct with the individual consumer, then it might be a reasonable suggestion that cooperative educational methods could be as successfully applied in this field as they have in other matters pertaining to the American home.

It would be rather imaginative to assume or suggest at this time that home construction, as regards weather insulation or type and size of installation of heating plants, should be regulated by municipal or state ordinances as is the case, for example, in matters of sanitation, plumbing, electrical wiring, etc. Rather, experiences in direct contact with the domestic consumer during the past few years lead to the conclusion that educational methods based on thorough research and development can be applied in making the public aware of the benefits to be enjoyed from the economical burning of fuel in well-designed and properly operated heating equipment.

As pointed out in Fig. 1, the ideal home, from the heat standpoint, involves many factors other than just fuel. While the anthracite industry has for the past few years been actively engaged in problems of service and utilization, and has only scratched the surface of possibilities, its experiences have been sufficient to prove that the consumer is willing to be advised and to make use of the advice.

Laboratory research by the anthracite industry on domestic utilization was put into operation on December 1, 1928, but in order to concentrate on the work accomplished in the field with the consumer, a summary on the work accomplished in the laboratory to date will be covered later.

Service Contact With the Consumer

In searching for a method of procedure to extend service to the consumer, it was found that all complaints from the customer to the retailer were based on quality of coal. This condition did not check with conditions at the mines where the installation of improved methods of preparation was resulting in the shipment of coal of high quality.

The retail dealer at that time was seriously handicapped in dealing with the situation because his knowledge of what constituted a good heating plant in good repair was practically nil. He generally handled such complaints with the explanation that the coal delivered was just as he had received it from the producer and he could do nothing about it. It was seen that such a condition must be corrected, and it was decided to make a very careful survey of some 5000 such complaints.

Plan of the Heating-Plant Survey

In order that the results of this survey would be of a nature permitting general application later, the following definite plan was arranged and strictly adhered to in every case investigated:

(1) The complaint must be from the consumer to the retailer delivering the coal.

(2) Investigations were not to be limited to one city or section, but carried on in all centers of the anthracite-consuming territories, as follows:

Portland, Me.	Albany, N. Y.
Boston, Mass. (Metropolitan area)	Syracuse, N. Y.
Providence, R. I.	Rochester, N. Y.
New York City (Metropolitan area)	Detroit, Mich.
Philadelphia, Pa. (Metropolitan area)	Chicago, Ill.
Baltimore, Md.	Milwaukee, Wis.
Washington, D. C.	Minneapolis, Minn.
St. Paul, Minn.	

(3) All investigations were to be made by engineers representing the anthracite producers, in the presence of the customer and retailer who delivered the coal.

(4) Investigations were to be carried on until sufficient proof and cause of trouble would be to the mutual satisfaction of the three parties.

With very few exceptions there was a cooperative spirit with the customer, and the findings shown in Table 2 formed a real basis for the work of educating the dealer to qualify himself as a source of heat service, rather than be on the obsolete basis of operating merely as a source of supply. These investigations

TABLE 2 FINDINGS OF DOMESTIC HEATING PLANT SURVEY BY ANTHRACITE COAL SERVICE

(Service Bureau of Anthracite Industry. Number of homes visited, 5000)

Classification		Causes	Occurrences	Per cent of total
1 Faulty equipment 62 per cent of total	(a)	Smokepipe, leaks, dirt, construction	1145	12
	(b)	Chimney, construction, obstructions, leaks, size	572	6
	(c)	Boilers and heaters, undersize, erection	668	7
	(d)	Soot on heating surfaces of boilers and furnaces	1526	16
	(e)	Oil in water, steam boilers	191	2
	(f)	Radiators, air-bound or otherwise obstructed	382	4
	(g)	Hot water and steam piping, layout, pitching, etc	286	3
	(h)	Warm-air systems, general arrangement, including warm-air flues and cold-air supply	573	6
	(i)	Thermostats	191	2
	(j)	Grates	191	2
	(k)	Miscellaneous	191	2
2 Method of firing and general operation 36 per cent of total	(a)	Disturbing fuel bed	191	2
	(b)	Carrying fire at wrong depth	477	5
	(c)	Manipulation of dampers	1145	12
	(d)	Operation of grates	191	2
	(e)	Size of fuel being used	954	10
	(f)	Not maintaining proper water line in steam boilers	191	2
	(g)	Accumulation of ashes in ashpit	191	2
	(h)	Miscellaneous	95	1
3 Quality of coal 2 per cent of total	(a)	Excessive degradation	95	1
	(b)	Excessive slate and bone or other foreign matter	95	1
		Total number of occurrences	9541	100

in the field also provided an excellent laboratory in which the consumer has assisted in no small way in the development of improved methods of utilization.

Faulty Equipment

Of all the difficulties, 62 per cent was due to faulty heating equipment, 56 per cent of which was found to be caused by the condition of the chimney, flues leading from furnace to chimney, and soot lodged on the heating surfaces of the furnaces. This latter condition not only greatly reduced the thermal efficiency of the furnace but obstructed draft.

Very satisfactory progress to correct these conditions has been made all over the anthracite consuming territory. This has been accomplished largely through educational methods, first from the service department of the anthracite industry to the coal dealer, and then from the coal dealer to the consumer. From the educational state it developed into a practical working plan of service which is quickly becoming a very important and economic feature in home heating, and there are now but few sections of the territory where such service is not available.

When the tonnage sold by a dealer so warrants, the individual dealer has established his own service department, and where dealers handle comparatively smaller tonnages, they have grouped into cooperative service organizations.

A practical example of the latter plan is that covering the operation in and around Elizabeth, New Jersey, where sixty-nine dealers are carrying on this service and maintenance work through a central office and in conjunction with a registered plumbing and heating contractor.

A representative example of this service and maintenance work being done by an individual company located in Brooklyn, N. Y., can be described as follows:

All heating surfaces of the furnace are cleaned and scraped thoroughly by means of the electric vacuum machine as illustrated in Fig. 2. The flue leading from furnace to chimney is taken down, cleaned, and replaced after necessary repairs are made. The dampers are then checked over, water glass cleaned (if a steam boiler), asbestos covering repaired, and the general condition of the furnace checked over. If major repairs are necessary, an advisory report as shown in Fig. 3 is left with the customer.

The cleaning and servicing operation is completed by white painting the asbestos covering of the furnace and black painting the doors, etc. A nominal charge is made for this work, and the net result has been valuable. After finding thousands of furnaces uncleaned for several years and in bad state of repair, the consumer has not only taken a new interest in the heating plant, but utilization is on a more efficient basis.

Until this class of work was first taken up by the coal dealer, it was always the heating contractor who was called in for such cleaning work, and there was some question about the coal dealer annexing this work. However, much cooperative work has been done in this respect, and the heating contractor now finds the coal dealer a source of maintenance and repair work which greatly overbalances any other prejudice.

Overloaded Furnaces

Next in order causing operating difficulties is that of overloaded furnaces or boilers. In drawing conclusions in any investigation it is the practice to see that all other conditions are right before stating that the boiler was of inadequate capacity. Too many overloaded boilers are in operation today, and as a matter of fact are still being installed. This is particularly true in connection with houses on large contract operations built largely for speculation, and where a few dollars saved in first cost is important to the contractor but very unfortunate for the ultimate purchaser. How to solve this problem completely is always one that promotes ready discussion.

It is at this point, as referred to before, that boiler-rating codes drawn up and adopted by the various societies and manufacturers associations become ineffective, and lead to the belief that a full realization of properly selected equipment will not be present until the ultimate consumer is given more light on the subject.

An outstanding example of this was in connection with a group of 200 two-family houses involving the operation of 400 steam-heating boilers. The occupants of these houses had gone through two heating seasons under most unsatisfactory conditions when at a meeting of the owners they were told something about heating their homes. The meeting was arranged through a coal dealer supplying a majority of the fuel, and the facts given at the meeting were based on surveys made of a number of



Fig. 2 The Electric Vacuum Machine Provides a Convenient Means for Cleaning Furnaces

the heating plants which were installed uniformly in all the houses.

There were 320 sq. ft. of standing direct radiation, and an indirect hot-water supply coil which, with allowances for lines and risers, showed a total net load of 450 sq. ft. connected to a boiler which, under accepted codes, would not be installed to carry more than a 290 sq. ft. net load. With hot-water supply included, the annual anthracite requirements should not have been over $11\frac{1}{2}$ net tons, but a survey showed an average of 14 net tons per boiler, representing an average excessive cost of $35 annually. To make matters worse, all of these boilers were connected to a 6 in. × 6 in. × 28 ft. chimney whereas the manufacturer specified a 10 in. × 10 in. × 35 ft. chimney.

First cost was undoubtedly responsible for this combination of conditions. The present owners were not in position to scrutinize the plans before erection, but it is reasonable to assume that fitting publicity and educational methods would show some way for a prospective purchaser of a home to know more about the heating plant he is about to acquire. When this is done one great handicap will be removed in that the speculative builder will be forced to give proper consideration to a very important part of the home.

Work has been done with some 6000 anthracite retail dealers, giving them facts on this phase of house heating, and also urging them to make contact with prospective builders or owners of homes, giving them information on the subject. The rating code adopted by the American Society of Heating and Venti-

lating Engineers and the National Boiler and Radiator Manufacturers' code in boiler testing are fully subscribed to. Effective work is expected to be done by the Heating and Piping Contractors National Association with its certified heating campaign.

Repair Recommendation Slip

Our Engineering Service Crew have completed Cleaning and Inspecting your Furnace and find the following Repairs Necessary:

_____

_____

_____

_____

We advise consulting your Plumber or Steamfitter at once.

If difficulty arises during the Burning season request our Free Advisory Service.

Signed_____

Date_____

HENRY HENJES, Inc.
COAL

BENsonhurst 0002-0003 **Service Dept.**

FIG. 3 FORM FOR RECOMMENDED REPAIRS

ARRANGEMENT OF DAMPERS AND THEIR CONTROL

From Table 2 it will be seen that 36 per cent of the causes of unsatisfactory results was due to methods of firing and general operation. Damper control offered the most opportunities for observation and suggestions to consumers. In checking over damper arrangements offered by the numerous boiler manufacturers, a rather wide variance of ideas is found. Those engaging greatest attention during the survey are outlined in diagrams A and B in Fig. 4.

Diagram A shows a counterweighted combination check and flue damper which, while entirely safe from a gassing standpoint, is inefficient as a means to regulate draft. When the check damper is closed the fire is subjected to a full chimney draft regardless of the size of anthracite being burned. In many instances this condition causes unnecessarily overheated fuel beds, resulting in preventable chimney losses and clinkering.

Diagram B shows an arrangement which is not only inefficient but decidedly unsafe, and it was usually encountered when customers were complaining about "coal gas," especially at night after the fire had been banked. First, it is inefficient, as regulation or setting of the flue turn damper in any position other than wide open reduces the efficiency of the check damper in checking the rate of combustion when there is no demand for heat. Secondly, when the fire is banked at night it was found to be the rule to close the flue turn damper as well as opening the check damper, and this permitted the gases to pass from the fuel bed up through the furnace and then out into the basement through the open check damper. In all cases of this kind encountered it was urged that the check damper be sealed closed permanently and a new check damper installed between the flue turn damper and the chimney, as shown in diagram C.

Diagram D outlines a damper arrangement which is efficient and safe, but is found in the minority. This gives independent flue turn damper control, and the check damper is in the correct relative location for efficient and safe checking of the fire.

It would be undoubtedly another step in the right direction if this principle of operation would act as a guide to a uniform type of damper gear furnished by boiler manufacturers. It lends itself to various adaptations, and should be considered as an integral part of the boiler or furnace as furnished by the manufacturer. All check dampers should provide a free opening of not less than 80 per cent of the area of the flue leading from the boiler to the chimney. Check and turn dampers should not be designed to operate in tandem when a change of draft condition is required. This does not permit the regulation which is desirable for different sizes of anthracite.

It was found that better results were the rule where damper

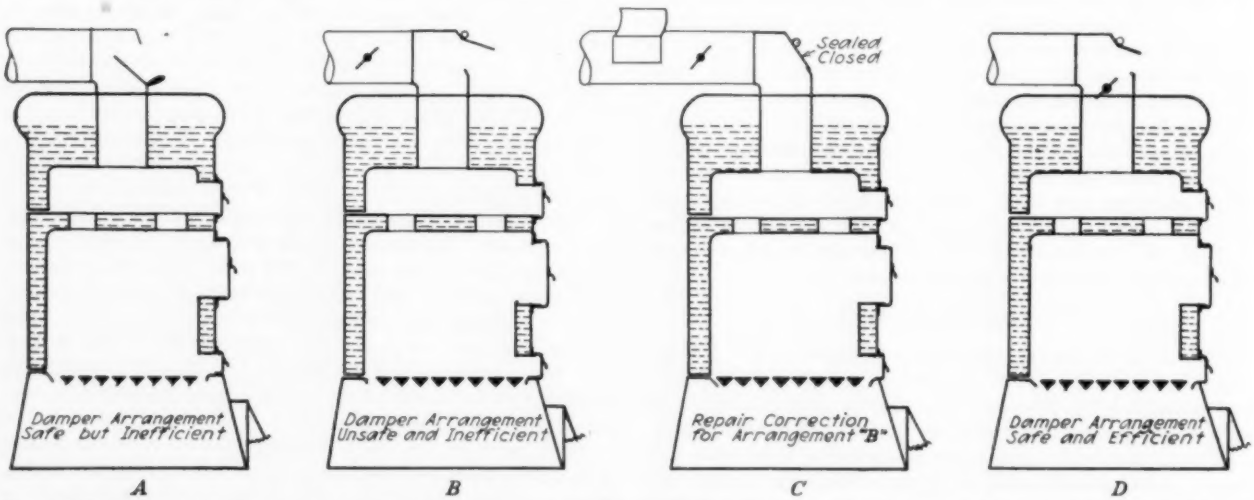


FIG. 4 DAMPER ARRANGEMENTS. A—COUNTERWEIGHTED COMBINATION CHECK AND FLUE DAMPER. B—INEFFICIENT AND UNSAFE ARRANGEMENT. C—CHECK DAMPER INSTALLED BETWEEN FLUE TURN DAMPER AND CHIMNEY. D—FLUE DAMPER INDEPENDENT OF CHECK DAMPER

arrangements, as shown in diagram D, permitted operation of the flue turn damper independent from the check damper. The flue turn damper should be set to meet the draft requirements of the particular size of anthracite being used. The check damper should be used to start or to check the draft to the furnace.

The Ashpit Draft Door

Practically all steam and hot-water heating boilers are fitted with diaphragm or sylphon damper regulators which operate the check dampers in combination with the ashpit draft door; that is, when the regulator operates to check the fire, the ashpit draft door is closed and the check damper in the flue is opened.

Compare this plan of operation with that practiced in large steam boilers where natural-draft control is not obtained by opening or closing of ashpit doors. This point should be given consideration. To date, this has not been investigated in the laboratory, but experiments in the field give considerable encouragement to the idea that improved and uniform damper arrangements between furnace and chimney, as shown in diagram D, would permit operation without the ashpit draft door and result in improved furnace operation.

This method of control is particularly beneficial when overloaded boilers are encountered in conjunction with clinker trouble. Forcing of the fire causes overheated fuel beds, and when the dampers finally operate, a sudden shutting off of air to the ashpit invariably results in ash fusion at the fuel and ash line. This formation of clinkers was practically eliminated by leaving the ashpit draft door open and limiting control to the other two dampers.

Selection of Size of Anthracite for Use

In treating this item it will be of interest to outline the standard sizing, also the yield of the various commercial sizes, as an average for the entire region based on 70,000,000 gross tons annual production, as given in Table 3.

TABLE 3 SIZING AND AVERAGE YIELD OF SIZES

Classification	Standard round-mesh sizes Through, in.	Over, in.	Yield Gross tons	Per cent of total
Broken	4 7/16	3 7/16	455,000	0.65
Egg	3 7/16	2 9/16	7,875,000	11.25
Stove	2 9/16	1 9/16	17,535,000	25.05
Chestnut	1 9/16	11/16	17,318,000	24.76
Pea	11/16	9/16	5,453,000	7.79
Buckwheat No. 1	9/16	5/16	9,240,000	13.20
Buckwheat No. 2	5/16	3/16	5,670,000	8.10
Buckwheat No. 3	3/16	3/32	5,530,000	7.90
Smaller	..	..	924,000	1.30
Total			70,000,000	100.00

Of these nine sizes, egg, stove, chestnut, and pea are classed as domestic sizes, in addition to which No. 1 buckwheat is gaining in popularity due to the development of automatic stoking equipment.

Some people are accustomed to burning a particular size of anthracite, and they are reluctant to be shown the advantages of using a size or combination of sizes which would result in better operation or greater convenience. There are others, however, who are anxious to know which will best suit the purpose.

In all the work of research and survey it has been found rather difficult to formulate a specific set of rules to govern the selection of size or combination of sizes, except that in the case of egg coal its use without combination with smaller sizes should be limited to furnaces where the depth from firedoor level to grate level is 14 in. or more and where the diameter of the grate is 22 in. or more. This is necessary in order to get volume to maintain ignition, but when furnaces not meeting these requirements are encountered it is good practice to use the egg size in combination with pea or buckwheat. The admixture of these smaller sizes acts as a baffle against excess air and "snuffing out" conditions encountered due to the low draft resistance of straight egg used in small volumes. Other than this the determination of size or combination of sizes to use is largely a matter of what is best adapted to the local conditions as regards draft available and design of the equipment to control the draft.

In connection with this question of selection of sizes probably no more interesting data could be presented than those compiled and recorded for the Anthracite Operators Conference in 1925–26 by R. H. Fernald, director, Mechanical Engineering Department, Towne Scientific School, University of Pennsylvania, Philadelphia, Pa. A great number of tests were made to determine the relative economic values of the several sizes of anthracite with the mixture of two or more sizes in varying percentages. Among the most important conclusions resulting from this series of studies were the following:

(*a*) From the standpoint of excellence of general performance, uniformity of combustion, ease of fire control, freedom from clinkering, ease of fire cleaning, and general high efficiency, the anthracite mixture of 80 per cent chestnut and 20 per cent pea took first rank of the solid fuels tested.

(*b*) The operating characteristics of the mixture of 40 per cent egg and 60 per cent stove were similar to those of the chestnut-pea mixture, thus giving it a high rating from the standpoint of operation performance.

(*c*) The excellent operating behavior and high efficiencies secured from the mixture of chestnut and pea and the corresponding ease of handling the mixture of egg and stove fully warranted further investigations of the exact proportions that will result in the highest efficiencies for mixtures of these sizes.

It was decided to devote the year 1926–27 to a careful investigation of the efficiencies and economics of different sizes and mixtures of sizes of anthracite for house heating. The tests were made in three series based on present-day commercial anthracite sizes, as shown in Table 4.

As this investigation was a study of the relative economic values of different mixture proportions of anthracite sizes and not a test of equipment, no effort was made to determine the relative efficiencies of the different heaters or the variation in efficiency of the individual heater under different load conditions. The efficiencies shown are based on the amount of heat transferred to the water.

While similar tests were conducted on various sizes of equipment the data for this discussion will be limited to that giving relative efficiencies in connection with a house-heating boiler

TABLE 4 SIZES OF EGG, CHESTNUT, AND PEA COAL

1 Egg Series

	Per cent oversize	Per cent egg	Per cent stove	Per cent chestnut	Per cent pea	Per cent slate	Per cent bone
(a)	5	80.0	15.0	0.0	0.0	1.50	3.00
(b)	5	55.0	40.0	0.0	0.0	2.00	3.00
(c)	5	30.0	65.0	0.0	0.0	2.50	3.00
(d)	5	15.0	38.0	42.0	0.0	3.00	2.50
(e)	5	12.0	32.0	36.0	15.0	3.00	2.50

2 Chestnut Series

	Per cent oversize	Per cent chestnut	Per cent pea	Per cent undersize buckwheat	Per cent slate	Per cent bone
(a)	5	80.0	10.0	5.0	4.5	2.0
(b)	5	75.0	15.0	5.0	4.5	2.0
(c)	5	65.0	25.0	5.0	4.5	2.0
(d)	5	60.0	30.0	5.0	4.5	2.0
(e)	5	55.0	35.0	5.0	4.5	2.0
(f)	5	45.0	45.0	5.0	4.5	2.0
(g)	5	40.0	45.0	10.0	4.5	2.0
(h)	5	35.0	45.0	15.0	4.5	2.0
(i)	5	30.0	45.0	20.0	4.5	2.0
(j)	5	45.0	20.0	30.0	4.5	2.0

3 Pea Series

	Per cent oversize	Per cent pea	Per cent undersize buckwheat	Per cent slate
(a)	0	85.0	15.0	7.0
(b)	0	70.0	30.0	7.0
(c)	0	55.0	45.0	7.0
(d)	0	40.0	60.0	7.0

having a manufacturer's rating of 1025 sq. ft. steam radiation with 3.53 sq. ft. grate area. Tests on each mixture were run for seven consecutive days, and for 15 hours out of each 24 the boiler was required to maintain a uniform load equivalent to 60 per cent of the manufacturer's rated capacity. For the other 9 hours out of each day the boiler was operated under banked conditions with a load equivalent to 25 per cent of the manufacturer's rating.

The operation at 60 per cent of manufacturer's rating was chosen because such heaters maintained at rating would mean flue-gas temperatures in the neighborhood of 1000 deg. fahr. or higher. This condition would of course destroy the object of the tests. The relative efficiencies of the sizes tested in the boiler described are shown graphically in Figs. 5, 6, and 7.

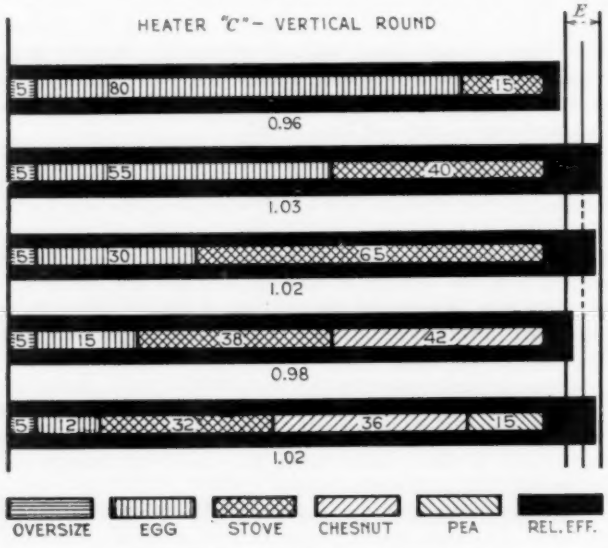


FIG. 5 RELATIVE EFFICIENCIES OF EGG SERIES

GENERAL CONCLUSIONS FROM TESTS

In connection with the vertical round boiler specified, the ease of fire control and excellence of general performance were less satisfactory with standard commercial egg coal (carrying 5 per cent oversize and 15 per cent undersize) than with any of the mixtures of egg, stove, chestnut, and pea, that is to say, egg coal carrying from 40 to 83 per cent undersize.

The average efficiency for the same boiler with egg coal carrying from 40 to 83 per cent undersize was higher than the corresponding average efficiency with standard commercial egg (carrying 5 per cent oversize and 15 per cent undersize). The average efficiency was practically constant for the egg mixtures carrying from 40 to 83 per cent undersize.

RESEARCH COVERING ANTHRACITE-BURNING EQUIPMENT

The third phase of the anthracite industry's research work is that started in December, 1928, for the purpose of developing and testing equipment which will provide the greatest convenience in utilization. This work is being carried on at the Frost research laboratory, at Norristown, Pennsylvania, under the direction of R. V. Frost, and for the present special attention is being given to the testing and development of thermostatic controls and automatic stokers for the burning of anthracite. As this development progresses and tests show that necessary requirements of convenience and economy have been met, the results will be published in bulletin form.

Fig. 8 shows a typical arrangement of a domestic-heating stoker for the burning of No. 1 buckwheat or No. 2 buckwheat anthracite under a round cast-iron boiler. This particular arrangement provides also for the continuous removal of ashes to a receptacle outside of the furnace. Fig. 9 shows a phantom view

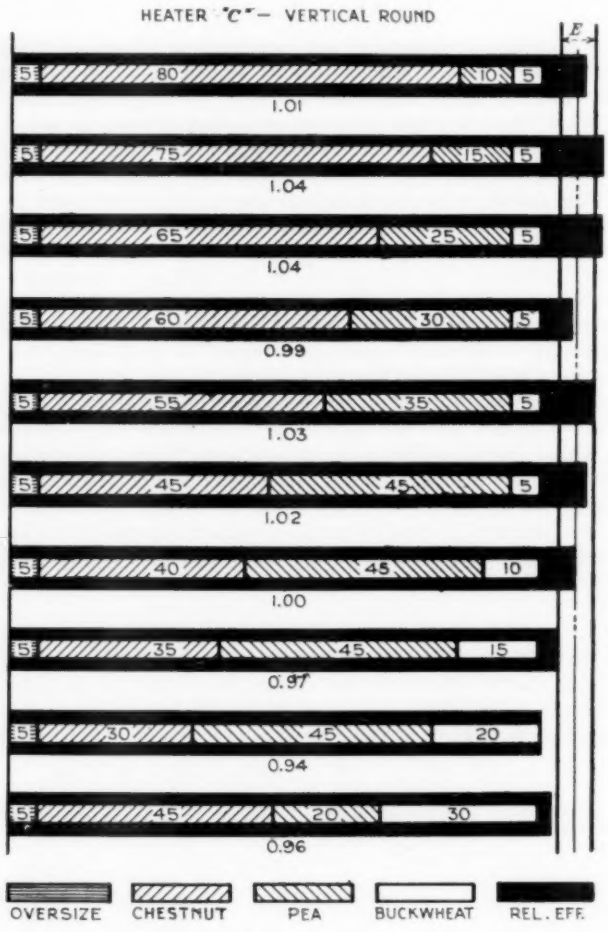


FIG. 6 RELATIVE EFFICIENCIES OF CHESTNUT SERIES

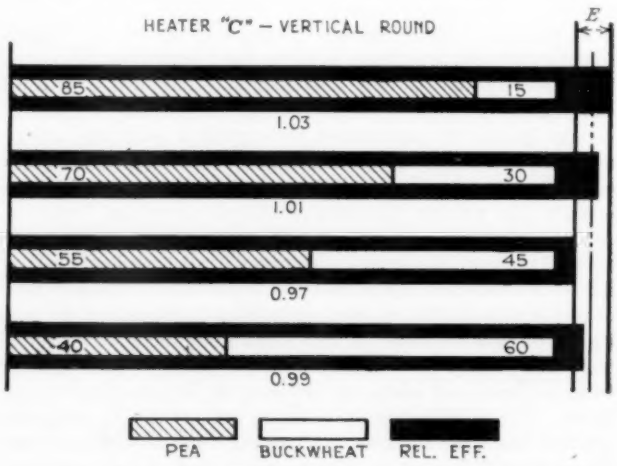


FIG. 7 RELATIVE EFFICIENCIES OF PEA SERIES

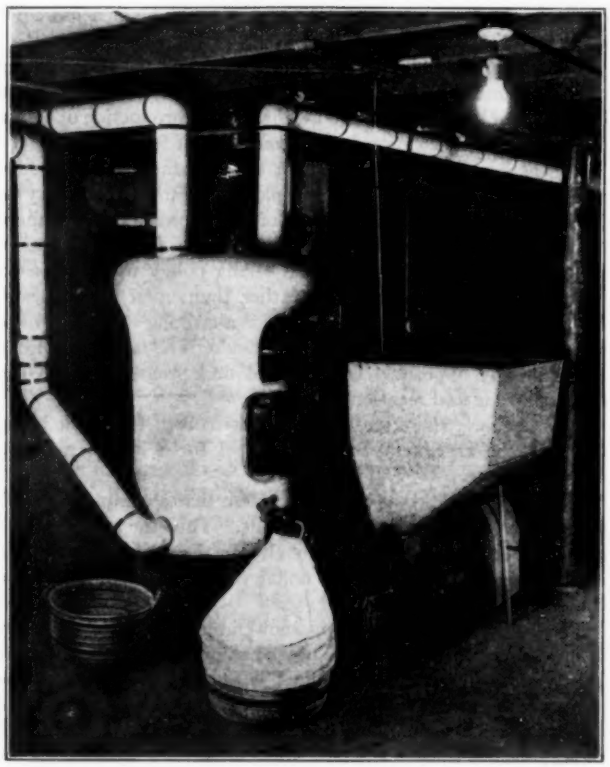


FIG. 8 DOMESTIC-HEATING STOKER ARRANGEMENT

of the same stoker applied to a sectional cast-iron boiler. That increased efficiencies may be expected from this type of equipment is brought out in Table 5, which gives results in tests where hourly coal consumption rates were 2.83, 5.93, 9.71, 16.6, and 21.4 lb.

THERMOSTATIC CONTROL OF DRAFTS

Much attention will be given to perfecting the utility of automatic draft controls which in turn will add greatly to convenience in the use of the larger sizes of anthracite in surface-fired boilers and furnaces. Investigations to date, however, indicate that damper design and arrangement will play an important part in this work. It is understood that one large producer of thermostatic controls finds it necessary to make many changes of dampers when installations are made. This is particularly true where arrangements outlined in diagrams *A* and *B* in Fig. 6 are encountered. The usual correction is outlined in diagram *C*.

TABLE 5 STOKER OPERATION

Test number	166	167	168	169	170
Date of test	5-23-1927	5-24-1927	5-25-1927	5-26-1927	5-27-1927
Duration of test	6 hours	7 hours	7 hours	7 hours	7 hours
Stoker, type	Domestic underfeed				
Boiler	Fitzgibbons H-16 (covered)				
TEMPERATURES					
Air in room, deg. fahr.	70.0	67.0	69.0	....	76.0
Flue gases, deg. fahr.	215.0	250.0	270.0	....	320.0
Water to boiler, deg. fahr.	55.3	53.1	52.7	52.1	52.0
Water from boiler, deg. fahr.	120.7	149.8	159.5	179.3	171.0
PRESSURES					
Air in pipe, inches water	0.10	0.09	0.19	0.36	0.48
Air in pot, inches water	0.03	0.23	0.51	1.01	1.29
Gas analysis, CO_2 per cent	6.0	9.0	11.5	....	....
Power required, watts	73-77	75-80	76-82	....	80-85
Stoker feed, notches	1	2	4	6	8
Air setting	1¾	2¼	3⅜	4½	5¼
Coal, kind and size	No. 1 buck.	No. 1 buck.	No. 1 buck.	No. 1 buck.	No. 1 buck.
Coal, amount burned, lb.	17.0	41.5	68.0	116.0	151.0
Coal, amt. per hr. burned, lb.	2.83	5.93	9.71	16.6	21.4
Coal, B.t.u. per lb. as fired	11,850	11,925	11,590	11,604	12,050
B.t.u. liberated	201,450	494,888	788,120	1,346,064	1,819,550
Water fed to boiler	2079.4	3377.5	5370	7891.4	10,606
Rise in temperature, deg. fahr.	65.4	96.7	106.8	127.2	119.0
B.t.u. absorbed by boiler	135,993	326,594	573,516	982,786	1,262,114
Efficiency developed, per cent	67.5	65.9	72.7	73.0	69.4

ASH REMOVAL

Development of equipment or systems to reduce to a minimum or to eliminate completely the necessity of ash removal by the householder is receiving serious attention. Devices on the order of that shown in Fig. 7 are steps in the right direction, but the goal is a practical system that will go further in that the ash and refuse will be removed from the premises without any labor whatsoever on the part of the consumer.

Vacuum ash removal bids fair to meet this requirement and progress is being made. Such a system for removing ashes from the boiler operating in a large apartment house was inspected recently. This equipment is really a portable adaptation of the standard vacuum ash-removal idea, being entirely self-contained on a large motor truck chassis fitted with pneumatic tires. A gas-electric set provides power for the operation of the exhauster blower, also for the belt conveyor which discharges the ashes from the truck hopper to the standard ash truck. An 8-in. universal-joint fitting connects the blower intake line to a standard 8-in. pipe running from sidewalk level to the boiler room. The advantages of such a system are readily appreciated when compared with the common present-day procedure requiring ashcans, ash hoists, general upkeep, and the attendant noise and dust.

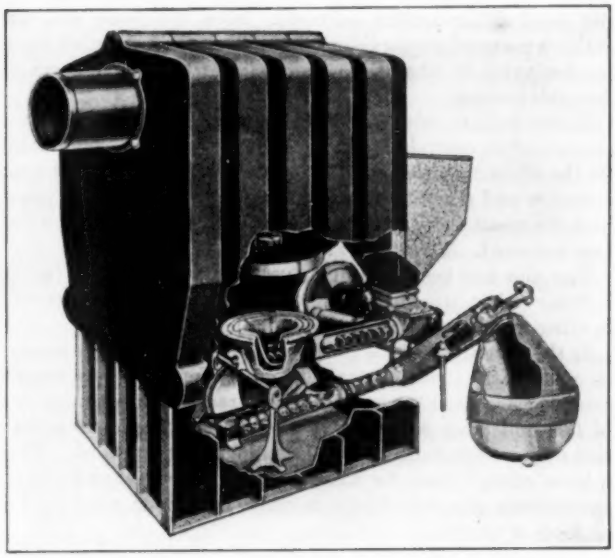


FIG. 9 PHANTOM VIEW OF STOKER APPLIED TO SECTIONAL CAST-IRON BOILER

It is proposed to design smaller units of this system which will make it adaptable for service to the domestic consumer. This is an excellent prospect to make manual ash removal a thing of the past in that it will be entirely practical to have ashes removed directly from the ashpit of the furnace to the service truck. Although the plans for de-

signing this apparatus are under way, it is not yet known to how small a size heating plant it may be applied.

Heat balances on some of these tests have been computed to check up the reliability of the results obtained. They show that the results are entirely reasonable. A typical heat balance (referring to test No. 168) is as follows:

	Per cent
Loss due to combustible in refuse in ashpit	5.3
Loss due to heat carried away in dry chimney gases	8.0
Loss due to moisture in coal and moisture formed by burning hydrogen	3.1
Loss due to radiation and unaccounted for	5.0
Total losses	21.4
Possible efficiency (by difference, 100 — 21.4)	78.6

Discussion

DAVID MOFFAT MYERS.[2] The paper is a worthy contribution in the field of fuel engineering, as pertaining specifically to anthracite for building-heating purposes. Both the paper and the author's personal accomplishment as a leader and organizer mark the beginning of what the writer believes to be a new era in domestic heating.

In the field of relatively large units, as in industrial power plants and in central stations, the art of economic combustion for the efficient generation of heat has been undergoing the most intensive and scientific development for the last fifteen years, with the result that today practically the utmost attainable has been achieved.

The aim has been maximum useful heat per dollar (which is first cousin to thermal efficiency) and reduced labor required; in other words, a minimum in fuel and labor.

In the domestic field the aim is exactly the same. If we assume as granted the items of "warmth" and "safety," there remain "convenience" and "economy" which translate into a minimum of fuel and labor costs. It is true that in the domestic field cost may include more than dollars apparently expended. Thus a home owner's time, for which he does not pay himself wages, nevertheless counts decidedly as cost. So also do soiled clothes and loss of temper.

Now when it comes to a question of reducing fuel and labor costs in domestic heating as compared to the same question in the power field, we have two sets of conditions so different as to be comparable chiefly by contrast. To be sure, fuel is fuel and combustion is combustion, but there the similarity pretty much ends.

The present state of the art in domestic heating may be said to exist chiefly in the pioneer stage, and the most hopeful sign for its future development is the fact that it is being dealt with by trained engineers.

One of the principal jobs of these engineers is to educate the public, i.e., that particular public which will benefit the most by increased convenience and economy. For it is necessary to create a desire and demand for better things and to disseminate information as to how these better things are obtainable.

The various channels through which this public may be reached effectively must be utilized. There appear to be three commercial agencies which must come in contact with the ultimate consumer more or less directly—the heater companies, the fuel producers, and the specialty companies.

Without the services of the first two, no heating plant exists or operates. The specialty companies engage in much advertising and publicity concerning their temperature-control systems and patent furnaces, combustion devices, and ash-handling methods. But their contact is not so sure as that of the first two groups. And while it is true that each group is endeavoring to sell its product, their efforts do positively assist in creating a desire for better convenience and economy. This is a most desirable result.

The fuel producers (i.e., the anthracite group), as stated in the paper, are making the most advanced and effective efforts thus far exhibited by any of the groups toward the production of better convenience and economy for the consumer of fuel.

The heater companies, although their contact with the consumer is more sure and more completely organized than any of the three, are doing the least of all of them toward real advancement in economy and convenience in the use of fuel. The reason for this appears to be an adopted selling policy which is based primarily on reduced first cost rather than upon reduced cost of operation and convenience. This is a striking contrast to the way business is handled in the power field. The two conceptions are strikingly opposite. One is right and progressive and the other is wrong and reactionary.

All the more, therefore, should be welcomed the work being done by the anthracite industry as set forth in the author's excellent survey.

It may be set down as a modern axiom that a commodity with service is worth more than one without. Thus the service which has been organized largely through Mr. Duemler's efforts has already added real thermal and dollar value to a ton of anthracite coal.

The author discusses the apparent clinker-producing tendency of checking a hot fire by closing the ashpit door. Probably at least one chief cause of this effect is the sudden raising of the temperature in the fuel bed by the quick and drastic reduction of air flow when the ashpit door is closed. The CO_2 would mount rapidly with a consequent temperature rise to a point probably above the fusion point of the ash in the fuel, whereas control by check damper would still permit a moderate air flow through the fuel, thus tending to avoid the clinkering effect.

The tests conducted by Prof. R. H. Fernald on the relative efficiency of various mixtures of different sizes of anthracite constitute a fine and constructive contribution to the art. It is easy to imagine why it is that in general the results are as they are. But the writer should like to check his imagination to the extent of knowing definitely whether the better results in each case were accompanied by a higher percentage of CO_2 in the flue gases.

Also it would be interesting, if possible, to translate the thermal-efficiency results into commercial-efficiency results by introducing the element of comparative fuel cost of the various mixtures used. In other words, which mixture is the cheapest to burn?

In closing, the writer wishes to express the satisfaction of one keenly interested in fuels and their economic application in witnessing the beginning of a movement to improve domestic heating as evidenced by the author's excellent paper.

LESTER C. BOSLER.[3] The most striking thought brought out in the paper, and it is hoped that the anthracite-consuming public will appreciate it, is the fact that a mixture of the different sizes of anthracite is a more efficient fuel than any one size alone.

Fig. 5 shows that a mixture of 40 per cent of stove size with 55 per cent of egg size with a 5 per cent of allowable oversize gives a greater efficiency than a standard egg coal. In Fig. 6 it again is shown that a 15 to 25 per cent mixture of undersize to chestnut gives better results than a standard straight chestnut coal.

This information should be very valuable to the consuming public because it represents the work of Dr. Fernald's tests made at the University of Pennsylvania a year or so ago. The time will come when the producer will make just two sizes of anthracite

[2] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[3] Mechanical Engineer, Maderia, Hill & Co., Philadelphia, Pa. Mem. A.S.M.E.

for domestic use. One will be for the warm-air furnaces and large steam boilers and the other for the smaller heating devices and kitchen ranges. They will then burn a more efficient fuel, but one that will not look so good in the coal bin.

To those wanting cleanliness and safety and at the same time the convenience given by the liquid and gaseous fuels, the steam sizes of anthracite can now be burned automatically and mechanically by stokers in such a way that the so-called hard work of shoveling the coal and ashes is entirely eliminated. By this method the yearly heating cost is the lowest and the convenience is equal to any other type of fuel.

NOEL CUNNINGHAM.[4] The author should be congratulated on the thoughtful and constructive character of his paper and particularly on having confined himself to present-day means for meeting present-day heating needs.

My own work as well as that of Mr. Duemler's organization brings me into contact with groups of merchandisers of fuel and heating equipment as well as hundreds of householders throughout the Northeast, and discussion with them of house heating and house-heating ways and means. The merchandisers are chiefly of three general classes: (1) Coal merchants who are thinking of the future of the coal business, (2) plumbing and heating people who have superposed present-day merchandising on the traditional conservatism of prewar days, and (3) young men who have incorporated businesses purely for modern merchandising to the new home demands. The householder contacts are almost exclusively with the middle and upper middle class having incomes of $5000 up.

People are educated through reiteration in advertising and propaganda articles in the home-phase magazines, to the conclusion that house heating can be dissociated from actual physical discomfort, mess, and all work of an arduous nature. The features which a very great number of anthracite users are intent upon avoiding are uncertainty of getting heat as and when required and dirtiness. Mr. Duemler has indicated that automatic control and stoker firing with synchronized coal feed and ash removal overcome these difficulties, and I can say from personal experience that he is right.

At the present time approximately five million households are being heated with anthracite. Some speakers have left the impression that this is all to be changed. I will say to those who believe that the American householder and his wife do not count the cost of home fixtures and go rushing into debt to satisfy any happy little whim, that they greatly underrate the canny thriftiness of their fellow countrymen. Normally every dollar of household expenditure is very thoroughly scrutinized; this applies to all classes, but becomes progressively more pronounced as real wealth increases. Those who casually drop into a show room and sign on the dotted line are almost invariably people of small means.

Two things will maintain anthracite in use: first, the investment which individual anthracite users have in good anthracite-burning equipment, and, secondly, the available accessories which fulfil their requirement that mess, arduous effort and uncertainty shall be eliminated, with the very important proviso that there must be proved savings which will largely or entirely repay the investment in added plant. There appears to be no foundation for an anticipation that any large proportion of householders will spend money for non-income bearing household features as long as the objections evident to their own common sense can be overcome at practically no expense.

CHARLES C. TRUMP.[5] The paper shows the importance of chimney draft for anthracite. It is equally important to regulate draft on domestic heaters with any fuel.

In fact, the difference between uncontrolled draft and properly balanced draft may mean as much as 30 per cent fuel economy.

A cold chimney does not draw. A hot one draws too much. Draft is also affected by wind and other atmospheric conditions.

Every heater using fuel therefore should be equipped with a simple but effective automatic draft control. A simple draft gage is also an advantage to show results of control.

A balanced check damper does more to give heater fuel satisfaction than any other one thing. Let us use more of them.

F. C. RUSSELL.[6] The entire question of home insulation and domestic-fuel conservation has been receiving much more attention recently than it ever has in the past, and home owners, as well as architects and builders, have shown keen interest in the application of insulation to the domestic heating plant.

The primary interest of course centers in the almost incredible economies which real insulation gives. Heating plants, in years past, have been well designed, and for the most part well installed, but the responsibility for insulation was not directly felt by the furnace manufacturer nor by the heating contractor who installed it. The result has been that the splendid results which were possible with these heating plants have been defeated by neglecting this extremely important point.

It is hardly conceivable that the home owner with his limited knowledge of heating and the complicated questions of fuel conservation would directly demand insulation, as he was ignorant of the real value and of the losses that follow the absence of efficient insulation.

But rising fuel costs have brought the possible economies of home heating to the attention of the home owners, and competition has stimulated the heating contractor and the furnace manufacturer. So today the owner of the house is insisting upon economy, and the furnace contractor is offering economy in his solicitation for business.

A great many different kinds and grades of fuel are being used today in the American home, and for the most part all of them will provide satisfactory heat if given a fair opportunity to do so. The fuel dealer is often unjustly blamed and accused of selling an inferior quality of fuel, whereas actually the cause for the dissatisfaction is due to heat losses from the boiler and piping through lack of adequate insulation. It therefore appears that it is well worth the while for the fuel dealers to recommend insulation whenever possible, as a close survey of the conditions existing in 75 per cent of the homes will show that most of the complaints will be overcome after the heating plant has been insulated.

There are a great many different insulating materials available, and while many of them are not without value, there are but few that are sufficiently efficient to produce the desired results. The thermal efficiency tests made by the U. S. Bureau of Standards may be helpful to those who desire to know the relative value of the most commonly used insulating materials. The comparisons show the heat loss in B.t.u. through various materials, and are as follows:

Air (if convection currents could be prevented), 4.2; mineral felt (mineral wool felted in blocks), 6.9; 85 per cent magnesia (magnesia and asbestos), 12.2; fire felt, 14.0; asbestos air cell, (corrugated asbestos paper), 12.2; asbestos cement (very hard and rigid), 40.0. The heat loss is the thermal conductivity expressed in B.t.u. per day per degree fahrenheit temperature difference per square foot per inch thickness.

[4] Domestic Stoker Company, New York, N. Y.

[5] President, Trump Corporation, Philadelphia, Pa. Assoc-Mem. A.S.M.E.

[6] President, Mineral Felt Insulating Company, Toledo, Ohio.

D. H. FAULKNER.[7] One cannot help being impressed by the vision of the author of this paper. Every one in the heating industry that knows Mr. Duemler and is familiar with the good work that he has accomplished in the past few years will realize that there is food for thought in this survey.

Table 2 is of special interest and might well serve as a page for a service manual in the heating industry. There is one item that might be included in this table, namely, lack of air in boiler rooms. Many times the writer has made a long trip to survey a faulty heating job only to find that lack of proper air supply for combustion was the trouble and that merely opening a window was the remedy. This is especially important on larger work.

The paragraph on overloaded furnaces is something of vital importance to any one in the heating industry. We have been faced with a confusion in boiler ratings for years. The efforts being made by the A.S.H.&V.E., National Boiler and Radiator Manufacturers organization, etc., are steps in the right direction. This, however, is going to take time, and anything that will serve as a yardstick for boiler comparison is worth considering.

If a boiler is considered as a unit, so much in the form of iron or weight is required for the base, grates, and other accessories, and the balance in the form of weight goes into heating surface. By this token weight is a pretty fair measure of boiler capacities.

In a table here listing sectional boilers, it will be noticed that they all weigh about the same, and it so happens that they all sell at the same trade price, within $20. Boiler *C*, put in on the basis of 50 per cent allowance for piping losses, starting load, etc., will give a good job. If this is so, there is something radically wrong about the other ratings. Of course, there will be some variation in the efficiency of these boilers, depending on the way the heating surfaces are exposed to the fire, but by and large they are of equal capacity.

This makes a pretty fair method of comparing boilers, especially in competitive work, as data on weights are published by all manufacturers.

Another important point that the author stresses is the matter of adequate flues. This problem is serious, especially in suburban development work where operations involving 500 houses are not uncommon. If the mortgage and loan engineers were approached on this subject, they would realize the importance of good flues and would compel the contractors to install them.

The table listing sectional boilers is as follows:

Boiler	Rating	Weight
A	5480	6680
B	5750	6940
C	3800	6920
D	5500	6670
E	5200	6440

HORACE C. PORTER.[8] The author has given an excellent presentation of the very helpful and effective service that is now rendered to the domestic consumer by the Anthracite Service Bureau in many of the large consuming centers.

The title of his paper, however, would lead one to hope that he would present the "economic status of anthracite" more in the light of its future possibilities as a competitor with other fuels, the markets it can be expected to retain, and the basis of relative price, quality, and service that will be required to insure its success in competition with other fuels. It is true, as he says, that the quality of the product has been greatly improved of late, through development of mechanical cleaning methods, but these operations add to the cost and they tend also to add to the amount of coal discarded with the slate and other impurities on the waste piles.

The anthracite industry is confronted with serious economic difficulty in putting out a product clean enough to meet present-day demands at a cost enabling it to compete with other clean and more convenient fuels. Unless a broad research program is undertaken looking toward more advantageous use of the very fine sizes, including those not now marketed at all, so as to develop applications bringing higher returns on them and thus permitting lower prices on the domestic sizes, the prospect is not a promising one for successful competition in the domestic field.

Research should be started looking toward a growing application of more and more of the finer sizes of anthracite (and, in fact, of all solid fuel including coke and bituminous coal) for the manufacture of gas for public distribution. It may be a considerable number of years before any large proportion of the anthracite production is devoted to the making of gas, but the industry, it is believed, will show a short-sighted policy if it does not see the certain future growth of gas utilization in domestic heating to the replacement of solid fuel, and take steps accordingly to enter the field even in apparent competition with its own principal product.

Mr. Nicholls' paper on coke, following Mr. Duemler's, put in convincing fashion the excellent results that can be obtained with well-prepared and well-sized coke in domestic furnaces, when properly controlled. The Bureau of Mines in its tests has made it clear that coke can be used successfully in household furnaces and requires no very troublesome attention or control. The great drawback lies in its bulk and the extra storage and handling required, and the somewhat finer control of the fire that is necessary. In making these tests of domestic fuels, such as the bureau has many times reported, it would seem that certain optimum conditions might be chosen in each case, suited especially to the fuel under test, and a valuable practical comparison thus made.

The important factors, however, in the whole matter of domestic-heating fuels are those of cost, convenience, and cleanliness. In considering cost, also, the relative fixed charges on the investment enter, often, as a deciding factor. Both anthracite and coke are clean, smokeless, and efficient, but there will come a day not far distant when both are looked upon as old-fashioned by reason of the cumbersome handling required the dust and dirt of the ashes, the labor and attention, and the lesser degree of adaptability that they offer to automatic control.

Relative prices and the ability of the purchaser to pay for added convenience and comfort will be the principal factors in the domestic-fuel situation for many years to come.

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[7] Sales Engineer, H. B. Smith Co., New York, N. Y. Assoc-Mem. A.S.M.E.

[8] Philadelphia, Pa.

FSP-52-10

Mechanical and Human Elements in Smoke Abatement

By GEORGE C. FISHER,[1] NASHVILLE, TENN.

A brief description is given of the history of the work of smoke abatement in Nashville, with the local conditions and the technical details and the general policies followed. The greater part of the paper deals with the human elements that entered into the problem of enforcement and the educational features that were planned and carried out to put smoke abatement into effect.

THE statement often is made by those connected with smoke-abatement that the work consists of 20 per cent mechanical engineering and 80 per cent of what one of our past-presidents calls human engineering.

While many engineering problems arise in promoting the work which, when judged apart, may appear difficult, yet when compared with the tremendous task of leading a city into new channels of thought and action, their solution seems comparatively easy. If smoke abatement were only an engineering problem with one visit sufficing to convince the smoke offender that he ought to follow out the proffered recommendations, one might go up one street and down another abating smoke with the greatest dispatch, and in most cases with a financial benefit to the individual owner of the stack. However, because of human frailty, prejudice, habit, inertia, and distrust, such occurrences are rare, and often many visits must be made, not to check the accuracy of the original investigations, but patiently to lead the mind of the offender through the gradual evolution of ideas to the acceptance of these first recommendations. The greatest hindrance to smoke abatement does not lie in its economics and its engineering but in the minds and the customs of the people.

In this paper some space will be given to the history of the work in Nashville and to the local conditions, the technical details, and the general policies in connection with the work. The remainder of the paper will deal with the human elements and the educational features relating to smoke abatement.

Referring to the history of smoke abatement in Nashville it is found that a municipal smoke department existed for a few years previous to the World War. Some constructive work was accomplished during this period, but due to the war and to other contingencies the department was abolished. Because of the fact that the organization was broken up many people gained the impression that this constituted a failure and that smoke abatement was hopeless. This served as a most noticeable handicap in the most recent smoke-abatement campaign.

The present Smoke Department had its origin in 1926. On June 16, of that year, a new smoke-abatement ordinance was adopted by the city. A short time later a Smoke Abatement Commission, composed of representative business men, was appointed. The commission felt the need of a competent and forceful man to organize the work and arouse the interest of the people. The services of John Hunter, of St. Louis, were enlisted, and he started a smoke-abatement campaign on July 1, 1926, and remained with the city for a year. On November 8, 1926, the author began his work as smoke inspector. During this year Mr. Hunter accomplished a great amount of work in the way of mechanical alterations to existing equipment and in instructing the smoke inspector in the proper methods of procedure. It is quite probable that his greatest accomplishment was in the molding of the minds of the public through various means of publicity and through his many practical applications made in the boiler room to a recognition of the fact that it could be done and that it was worth while. His policies of attacking the problem have been consistently followed out.

[1] Chief Smoke Inspector, City of Nashville. Assoc-Mem. A.S.M.E.

Presented at the Third National Fuels Meeting, Philadelphia, Pa., October 7 to 10, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Topography Made City a Smoke Trap

The local conditions found in Nashville at the beginning of the campaign will be briefly noted.

The coal used is all high-volatile, averaging close to 40 per cent. No low-volatile coal is used, for the very good reason that none is to be had at a competitively marketable price.

The topography of the city is concave, every road out of town leading up a hill. This bowl tends to trap the smoke and hold it for long periods of time. Fogs are prevalent throughout the heating season, being especially heavy from the middle of November to the first of February. Low wind velocities are the rule during the winter.

A great conglomeration of combustion equipment was found, and it was apparent that in most cases cheapness had been the prime requisite for its purchase. Low settings, insufficient draft, run-down plants, boiler rooms lacking light and ventilation and overflowing with dirt and discarded material were the rule rather than the exception.

The A.S.M.E. Boiler Code had been in operation in the city for some time and was having an indirect bearing for good in smoke abatement.

There was no license law for engineers or firemen and no active organization for their educational and social betterment. A local chapter of the National Association of Power Engineers has since been revived and is proving to be of great benefit. Capable and alert engineers and firemen were to be found, but were not in the majority.

Managers, as a rule, took very little interest in their boiler rooms, but considered them merely as necessary nuisances and sources of expense.

The heating season has a few days comparable to a Northern winter, thus requiring plenty of capacity. On the other hand, many of the days are typically Southern, which means low ratings, low furnace temperatures, and incomplete combustion.

There was too much of a tendency among architects to lean heavily on sales advice for their heating layouts.

There was no local chapter of the Heating and Piping Contractors National Association. In the author's opinion this fact, along with the lack of "certified heating," has subjected smoke abatement, the heating trade, and the public in general to many ills.

A strong, fighting minority of the populace were confident that smoke could be abated and were insisting that it should be done, while a majority of the people were indifferent, dubious as to the value of the work, or outrightly opposed to its institution.

The local skirmish between coal and other fuels existed.

The preceding statements may give the appearance of a very dismal outlook for smoke abatement, but special stress has been given to the hazards that had to be overcome, and it is supposed that most of them are not confined to Nashville. Many favorable items might be mentioned, such as an excellent and unusual response from managers down to firemen to alter equipment and improve conditions when requested to do so.

The general policy in carrying on the work, as adopted by Mr. Hunter, was to unite an extensive program of educational publicity with a great deal of individual inspection, instruction, and recommendations.

As the work was begun in the summer time, attention was given first to the manufacturing plants. Being aware of the fact that the average owner or manager is giving most attention to his production and sales and knows very little about his steam plant, a great effort was made to impress on him the value of smoke abatement and the possible savings that could be made in connection therewith. Inspections were made of each individual plant, and in most cases recommendations were made for a complete overhauling. Engineers and firemen were instructed in the best methods of abating smoke and saving fuel. Besides general repairs and alterations to existing equipment, suggestions were made as to specific apparatus that might be added to improve conditions and lessen the smoke. Where at all practicable underfeed stokers were recommended.

Air Jets and Underfeed Stokers Were Put In

At first, managers of the plants considered the prices of stokers prohibitive and they could not be interested. However, they would give consideration to air jets, including a special type of jet designed by Mr. Hunter, and 131 sets of these were installed the first year. After two or three months several underfeed stoker installations were made, and from that time the number of stoker installations has rapidly increased, until now there are 80 new underfeed stokers in the city.

During the following heating season time was given to the inspection of heating plants. Fully a thousand plants were visited, minor changes and repairs to boilers were advocated, recommendations were made as to the best fuel to use, and instructions were given to firemen and janitors. Near the end of the heating season the motor-driven underfeed stoker was introduced and the use since that time has become general. In many of the older types of boilers there was but one solution and that was the use of a smokeless fuel such as coke. Among all the various designs of the hand-fired heating boilers there was only one that was considered, under average conditions, to be an inherently smokeless boiler and that was the downdraft.

Railroads Made Many Helpful Changes

The railroads came in for their share of consideration. Conferences were held with the officials of the three railroads entering Nashville. As a result schools were organized for the instruction of engineers and firemen for smokeless operation of their engines; the number of traveling engineers and firemen was increased; completion of the installation of ring blowers and arches was accomplished; many improvements were made in the roundhouses; and the equipment of all yard engines with air-induction tubes was undertaken, this being now nearing completion.

There is a collection of miscellaneous fuel-burning equipment such as small vertical boilers, foundries, type furnaces, tar kettles, excavating and road-building machinery, and locomotive cranes. Of these the small vertical boilers, used for a great variety of purposes, are the most numerous. These boilers have many attractive features for the purchaser, and in the smaller sizes no very practical substitute is available. We require the burning of coke in these boilers, but are recommending in sizes of 15 hp. and over that they install either horizontal downdraft boilers or updraft boilers with small mechanical stokers. Coke, oil, gas, or electricity has been used in solving the problem with many of the other types of miscellaneous equipment. Gasoline drive is proving an excellent remedy in construction equipment.

Owing to lack of funds and personnel, the residence problem can only be taken up in a general way. Coke, oil, and domestic stokers are very gradually improving this condition.

Another important phase of the work that needs to be mentioned is the issuance of permits on new installations. The ordinance requires that smokeless equipment be installed, but does not go into detail as to what constitutes smokeless equipment. This left the decision in the matter to be made by the Smoke Department. Upon investigation of the problem it was soon apparent that the extreme smoke condition would require a radical procedure. Much of the so-called "smokeless" equipment sold would hinder rather than help smoke abatement. There would not only be the work of correcting old equipment, and there was enough of it to keep the department busy for several years, but there would be also the added burden and responsibility of inventing a collection of accessories, such as jets and combustion arches, to make a smokeless installation out of the equipment that was being advertised and installed as smokeless, and then in addition be required to follow it up with constant policing. The market was littered with "pain killers" when what was needed was a cure. If there is anything that makes smoke abatement ridiculous and hopeless in the eyes of a none too optimistic or favorable public it is the permission of new installations that are chronic smokers. Quack remedies invite lack of confidence, breed discontent, depress and discourage smoke-abatement officials, and lead to ultimate disaster.

Specific Types of Equipment Described

A little space will now be given to the discussion of specific types of equipment.

The Scotch marine boiler is considered by the author the least desirable of all. It generally comes in sizes too large to induce the owner to burn coke without continuous pressure, and as a rule it is used in sizes too small for the insertion of an underfeed stoker. A Dutch oven and stoker can be used in conjunction with it, but the first cost and maintenance do not warrant it. Jets help some, but that means another stack that requires continuous observation.

The so-called "smokeless" updraft heating boilers of the drop-arch type have been a constant source of trouble. Under ideal laboratory conditions these boilers approach smokeless combustion, but any equipment will do that; and, besides, the smoke inspector has to deal with average conditions and not ideal ones. Where possible it often has been suggested to the fireman that he knock the arch out so that he could see what he was doing. The words of an old engineer and fireman come to mind; he said that you cannot handle a fire when you cannot see it.

Overfeed stokers of the various designs, both automatic and hand-operated, have subjected the Smoke Department to many additional worries. They belong to the classes of equipment previously mentioned in that if all the conditions, including stack, setting, fuel, rating, and the many human qualifications of a good fireman, are just right, reasonably smokeless operation may be expected. However, these conditions seldom exist.

The underfeed stoker and the downdraft boiler have been found uniformly good. This does not imply that they are always smokeless, but it does mean that they require a minimum of policing.

The thought that presented itself to the author was that if they are the best, why not use them. Energy that is being spent to make other equipment smokeless might be spent in

keeping it out of the city and in promoting the installation of underfeed stokers and downdraft boilers. Needless to say this proved a difficult undertaking. Besides, it worked a hardship on the local representative of other equipment and to a lesser extent on the manufacturer. However, the fact remains that the first duty of a smoke inspector is to abate smoke.

The law was not used, but an appeal to the public was made, and the response was excellent. Unsatisfactory equipment was gradually reduced, until this year there has not been a single coal-burning installation above 1200 sq. ft. of radiation other than underfeed stoker or down-draft boiler.

The severe criticism and the opposition induced have been balanced by the simplification of permits and the calculation of construction dimensions and the relief from smoke violation that only these two types of equipment can give.

Human Element Has Vital Effect

In the remainder of the paper the human elements underlying smoke abatement will be discussed.

The human problem connected with smoke abatement involves an enormous task. The large number of difficult and varied duties to be performed and the many qualifications required of a smoke inspector to successfully perform them would warrant the employment of much higher salaried men than is usually the case. Many problems which arise might tax to the limit even the well-developed ingenuity, diplomacy, courage, and all-around leadership of a ten to twenty thousand dollar a year man.

Progress in general has that difficult task of overcoming the inertia of long-established custom and of controverting reactionary policies and misinformation. To the author the most interesting fact learned in connection with this work has been the realization of the intricacies of human nature and complexities of the human mind. To most of us, habit, whether good or bad, is an all-powerful taskmaster. Some one has said that the customs of a people are much more powerful, being the habits of the individual reenforced to utter complacency and self-satisfaction by knowledge of the fact that all around him are confirmed addicts of the same habits. The individual or the firm that undertakes to sell something new to the public which will radically alter the customs of many years' standing has a problem to meet. Stimulating the demand of the public for something new requires patience and resourcefulness. However, it is being constantly done in business and commerce.

In the work accomplished in Nashville, smoke abatement has been considered as a business and has been taken up in a business and engineering manner.

As in any business that is likely to succeed, honesty and sincerity of purpose have been outstanding factors. It is needless to emphasize that political methods are not used. No favors are accepted from any one. The weak and lowly are not oppressed nor the wealthy and influential sidestepped. Everything which aids smoke abatement is actively and openly favored, and everything which hinders smoke abatement is vigorously opposed. Particular makes of equipment are not recommended, but types are.

As in any business, smoke abatement must be built on a solid foundation. Foolish expenditure of money is not recommended and, in the case of many cure-alls, is not permitted. An unsatisfactory installation based on the recommendation of the Smoke Department does more damage than good.

Smoke abatement is sold on its merits. An attempt is made to engender among the people confidence in the work being done and in the recommendations offered. A firm belief is held that smoke abatement does not generally bring oppression; it renders cheap and valuable aid. Money spent on abating smoke is not charity, but a wise investment. A smokeless stack denotes an economical plant and an enterprising owner. Smoke abatement is an asset to a community, not a liability.

In making many contacts with the public the smoke inspector has much in common with the salesman in that he is selling his product, in this case smokeless combustion, to the people. One important difference is that he has the law behind him and is always assured an entrance and hearing. Besides, the threat of prosecution always lies in the background. In other respects he meets resistance in the way of prejudice, objections, alibis, faithless promises, procrastination, suspicion, inertia, pretended or actual financial difficulties, etc. This opposition takes on many forms, but a few are predominant and ever-recurring and will be mentioned.

The most common form of resistance is the pair of violators living close together who may be called Mr. A and Mr. B. It is quite a human tendency of the individual to be intolerant of the faults of others and entirely oblivious to his own. It is also a mathematical axiom that a business man sitting in his office with a large flat roof over his head and the top of his stack above it cannot see his own smoke, while the smoking stack of his neighbor is very often at such an angle that it can be constantly observed from his office window and serves as an ever-present annoyance. When Mr. A is interviewed, the most prolific advice is forthcoming from Mr. A as to the severe tactics that the smoke inspector should adopt toward Mr. B. This is followed by a multitude of reasons why he himself should be left alone. The same dialogue occurs when interviewing Mr. B. This attitude has been dispelled with good results by ridicule through the press and other means of publicity.

A variation of the previous condition is a group of several smoking plants in the same neighborhood with the owner of each one objecting to all the others. In meeting this situation the best results have been obtained by concentrating on the weakest party and breaking him down and then using that as a leverage on the next.

Difficulties in Securing Proper Equipment

Another well-defined custom of the people is the almost universal tendency to install the cheapest equipment that can be found on the market. They may be lavish in spending money on other things that they know something about, but seldom on combustion equipment. Where there is no municipal supervision the cheap-equipment representative is assuredly resting in clover. Due to severe competition the architect and heating contractor often have been found to accentuate this condition. It is being continuously called to the attention of prospective builders that the size of the boilers should not be determined by the proposed dimensions of the boiler room, but that the dimensions of the boiler room should be determined by the size of the equipment necessary to heat the building. The cost, quality, and capacity of the boilers should not be determined by the subtraction of the money spent on other parts of the building from the money available, but by their economy of operation and the service they render. In purchasing combustion equipment first cost should receive least consideration, while operating cost and relative service should receive the most.

Another condition closely related to that just mentioned concerns those who build to sell or build to rent. These prove difficult cases to handle as they are naturally confirmed buyers of cheap equipment. The best method to deal with this type of builder has been to strike at his weakest point, hinder the sale or rental. In one particular case the sale of a property was delayed for several months, numerous legal difficulties, extra expense, and no end of trouble being encountered. Warnings have been given through the press to prospective purchasers

or renters of property to get in touch with the Smoke Department and investigate the smoke situation of their heating plant the same as they would inquire into the title of the property. If the stack is a smoke violator, have it stated in the contract that the recommendations of the smoke inspector be carried out before the sale or rental is completed. These methods have brought results.

In view of the many philanthropies and other signs of a progressive age of civilization it may appear irrelevant to say that the most attractive inducement that can be made to the smoke violator is the dollar appeal, but such is the case. Work in this city has been based on the theory that everybody loves money. The savings possible from following the recommendations of the Smoke Department have always been kept in the foreground and the pleas for smoke abatement kept in the background. It is fortunate that mechanical and economic conditions today make this possible in practically all cases. Civilization has not reached the point where indefinite and long-postponed promises of group benefits strike a very responsive chord; the average individual is much more interested in himself and his own business. The average business establishment is overrun with people inspecting this and that, asking for donations, and selling tickets for everything under the sun. The business man has his own worries and is weary of all this. He wants a breathing spell by having some one come in to help him instead of asking for help. For this reason it has been customary with the author to emphasize personal savings on investment and not charity or civic duty.

The dollar appeal is most successful, but is not always effective. In cases where it fails the next best plan has been found to be the pressure of public opinion, and particularly so in a city of this size. Public opinion is a powerful leverage. Most individuals want to be well thought of by their fellow men. Most of us will do things or fail to do things when we think nobody will find us out that we would reverse if we thought the general public would be let in on it. Advantage is taken of this human weakness always to let the sunlight in, open it up to everybody, and give the individual unfavorable publicity. If a man is a leader in church or civic activities, and orating from the platform on moral obligations, law observance, and civic pride and then failing in the practical details, it has been customary to broadcast the facts. He is sure to have plenty of acquaintances who will enjoy hearing the news. This system has been used with many variations.

Prosecution has not been used as yet. Although the author is heartily in favor of it, the work is so new in this city that the methods previously mentioned are bringing the necessary changes as fast as they can be disposed of.

Methods of Handling the Violator

A few miscellaneous items in connection with human relations will be noted. It has been found that it never pays to stoop to bitter fighting and personalities. The best method is to use the qualities of a high-grade salesman. Courtesy and respect should be rendered every one and demanded in return. If the man is really busy or worried over other matters, it pays to excuse yourself and see him at another time. If he receives you, he should give you his undivided attention, and any effort on his part to do other work such as opening mail should not be permitted.

Criticism comes from all sources. If sincere and not too malignant, no attention is paid to it. Where underhanded methods are used and great enmity shown, especially by influential people, an immediate demand is made for a "showdown." The idea is always given that they are out of step, not the Smoke Department.

Ridicule, disparagement, and disrespect are sometimes shown. Many have the idea that politics plays a big part in the work. These are best overcome by keeping steadily at the work before you and refusing either to talk or to listen to politics.

There is a very natural opposition among equipment and fuel men who may nurse real or imaginary injuries. It is frankly admitted by the author that some damage has been done them, and also for the most part their opposition has been very fair and aboveboard. It is believed that no great personal enmity exists between them and the Smoke Department. Many of these have taken plausible steps toward meeting the requirements of smoke abatement. One thing that it is well to bear in mind in this respect is that, however much it might be desired to believe the contrary, the fact remains that they are in business to make profits and not to abate smoke. The only time that they will try to aid smoke abatement is when they think it will pay them to do it. It would be foolish to expect anything else.

One of the greatest factors for success is a force of salesmen with strong personalities and unlimited energy selling smokeless equipment. They tend to minimize the efforts of those desiring to sell unsatisfactory equipment. Often where the personality of the smoke inspector will make no impression in holding off opposition a good salesman of smokeless equipment will walk away with the order. Great interest is taken in encouraging local connections of smokeless-equipment manufacturers with powerful sales representatives. Nashville is full of underfeed-stoker and downdraft-boiler salesmen.

In order to meet and overcome the many obstacles, some of which have been mentioned, it was perfectly apparent that there would have to be a pooling of resources and that all that would aid the work would have to be gathered together and organized. Disorganized effort and lack of harmony would mean defeat. Those forces which were unfavorable could get around and tear down faster than the Smoke Department could build up. A tip was taken from successful business men. After assembling a good product their next step is to let the public know about it. Mass action is employed through publicity and advertising to educate the public to its many advantages. Due to the many hazards in the path it was deemed perfectly ethical to adopt the methods of a showman and "ballyhoo" smoke abatement. "Grandstand plays" were intentionally made. An example of this was to pick out occasional bad offenders who, due to their location, were prominently known. As soon as an underfeed stoker was purchased the fireman was induced to make all the smoke possible till the stoker was installed. This would invite complaints and attract interest and often ridicule, and then when the stoker was installed the change was so noticeable that it gave everybody a thrill.

Channels of Altering Public Opinion

The methods that have been used to change public sentiment will now be discussed. It is worthy of note that the public soon gets tired of one method and wants a change, so that something new must always be devised for them. It often pays to appeal to the feelings and imagination rather than to reason.

The press forms the foundation of all educational work. Without its complete support smoke abatement would be a failure. The inseparable twin pictures of stacks before and after with explanatory articles were used till they grew old. Photographs of smokeless equipment were inserted. Articles on the savings produced and copies of testimonial letters were printed. Communications, whether favorable or unfavorable, were encouraged. Cartoons proved very effective. A photograph of the first domestic stoker installed, with the owner's baby sitting on the hopper cover, had a human appeal. A news photographer took

a picture of a noted grand-opera star as she stepped off the train. A note under the picture said that she was holding her furs close around her neck to protect her million-dollar throat from the smoke of a shifting locomotive. A clipping of this was sent to the president of the railroad and it had a good effect. The effect of smoke on aviation was emphasized. Headlines such as, "Mail Ship Gropes in City's Smoke and Lands Over River" helped the work along. Rotogravure advertisements of smokeless equipment going into new hotels and office buildings proved quite a benefit. The papers have been good to give us many editorials. This will give an idea of the aid given by the press.

The Smoke Abatement Commission is considered indispensable to the unhampered progress of the work. The members have always been willing and ready to make visits with the smoke inspector, talk over the radio in the interest of the work, write personal letters asking for support and cooperation, give addresses before clubs, and in many other ways lend their support.

The radio has assisted us greatly each winter. Station WLAC donated sixteen five-minute periods to the Smoke Department last winter. Prominent citizens from different professions and callings, such as a lawyer, nose and throat specialist, architect, engineer, women's club members, life insurance official, business man, and a manufacturer, were called upon. In the case of business men and manufacturers care was taken to pick men who had followed recommendations offered by installing equipment and who had made appreciable savings. These radio talks, coming from men viewing the smoke nuisance from different angles, aroused much interest, and favorable comment and many letters and telegrams were received.

Instruction cards for domestic use were printed and about 20,000 were distributed. On one side the approved method of firing coal in a residence furnace, together with an explanatory diagram, was printed; on the other side similar information was given concerning coke.

Booklets were printed describing the care and operation of the domestic furnace and the necessary steps to take to decrease the smoke nuisance. About 12,000 of these were distributed among home owners.

Domestic Firing Influenced by Cards

Instruction cards for the firemen of industrial plants were printed and tacked up in all boiler rooms.

Large equipment posters, 24 by 36 in., were printed. These were set up on the premises of new buildings and called the attention of the public to the fact that the owner had cooperated, the Smoke Department was on the job, and a type of smokeless equipment such as an underfeed stoker had been installed. These cards have given excellent publicity to the work. Many persons, out of interest and a desire to learn something new, inspect new construction work, and when they would see this card their curiosity would lead them to the boiler room to see what the equipment was like. This left a definite impression on their mind. Others who would merely pass by would know that something was done.

The theaters were put to work. Actors coming to the city always complained about the smoke. They were given encouragement to work some jokes about Nashville smoke into their act.

A number of articles on smoke and its abatement were written for trade magazines, local and otherwise.

Many addresses were made before clubs and organizations of all kinds. Literature relative to the smoke nuisance would be distributed among them at these times. Requests were made for their cooperation and support, and invitations were offered to any one particularly interested to visit the Smoke Department offices.

Direct Letters Form Part of the Plan

Correspondence has played a big part in promoting smoke abatement. Letters of recommendation constantly are being sent out. The most important item in connection with the work is considered the care taken in obtaining smokeless equipment in new installations. It seems illogical and useless to run one's legs off stopping old leaks and then allow new ones to appear. An individual case is made of every proposed new installation, and letters of recommendation are sent to architect, owner, contractor, and every person interested. Letters of appreciation are sent to all those who have followed our recommendations. This tends to make permanent their support and cooperation and puts them in a happy frame of mind regarding the work.

Articles are printed in the newspapers describing some new installation of smokeless equipment and containing testimonial letters from the owners regarding the large savings and satisfactory service that resulted. A number of these papers will be purchased and the clippings of this article along with an introductory letter sent to about fifty of the livest prospects. These few examples give an idea of what may be accomplished through the mail.

Encouragement on a small scale has been given to civic clubs and various other organizations to appoint committees to aid in the work. As the men appointed on these committees are generally leaders and successful business men who know nothing about smoke abatement but who have decisive views and dominating personalities, there always is danger of a top-heavy organization and of pulling in different directions. Although this method has been used very conservatively and with great success so far, the fact is appreciated that there always is danger lurking around. So long as they can be kept from taking the bit in their mouth and jumping over the traces their customary boundless energy can be used to wonderful advantage.

A number of coke-burning displays have been held in downtown office buildings and at the Tennessee State Fair. Literature on the good qualities of coke and the proper methods of burning it was distributed. These displays were not sponsored by the Smoke Department, but great interest was taken in them.

One of the best ideas on publicity was the promotion of window displays about smoke and its abatement. The displays were placed for two weeks at a time in three locations, a show window of a downtown department store, and in the central public library, and in the show window of a downtown hardware store. Each location was admirable so far as attracting the attention of the maximum number of people. Articles on and photographs of the displays were printed in the newspapers. The center of the display was composed of a large chart of coal and its by-products. About ninety samples of coal by-products were placed in front of the chart, and colored ribbon streamers were run from the name on the chart to the sample. An electric light was placed behind the samples so that the many beautiful colored dyes were shown up to the best advantage. At either side were photographs, literature, and drawings showing the implements to combat smoke. These displays proved an unusual attraction and great crowds collected. It was most interesting to mix in unknown to the crowd and listen to their comments. Their expressions were most informative and showed to an alarming extent the gross ignorance and hazy ideas that the average individual had as to what smoke abatement was all about. This was not confined to the lower classes, but was almost as common among the intelligent and

educated classes. The idea which presented itself most prominently to the author's mind was that what these people needed most was education and not prosecution.

Booth Displays at Local Fairs

Another attractive feature that was promoted by the Smoke Department was a smoke-abatement booth at the Nashville Pure Food and Better Homes Show. The center attraction in this case was a human lung, black with soot, preserved in a jar of alcohol, and resting on a high stand with a light behind it to show it up. On a placard it was called to the public's attention that the natural color of a lung taken from a person in a country district was pink. One man asked why the lung of a farmer was not placed beside it for comparison. A friend with him said the answer to that was easy. The air in the country was so pure and as a result the farmers were so healthy that they never died to furnish a sample. A local aviation official, very bitter against the smoke clouds, offered the suggestion that a handle be attached to the jar and that it be carried through the city by the smoke inspector on his visits as a practical and conclusive argument for smoke abatement. Behind this portion of the display was a table containing baskets of coke samples and jars of fuel oil. On the walls were tacked photographs and literature of smokeless equipment. Near the top of the rear wall a skyline of smoking stacks was painted. Booths containing domestic stokers, oil burners, and coke-burning displays were grouped around the smoke-abatement booth. One unusual display was an open grate, without flue or chimney, burning coke. It was a good lesson on smokeless combustion. During the afternoons and evenings three persons were kept busy at the smoke-abatement booth handing out literature and explaining the details.

An excellent means of educating the public has been through the schools. These classes were carried on under the auspices of the Nashville public schools. A night school for engineers with a professor of mechanical engineering for instructor accomplished excellent results. There is a night school for white firemen and another for colored firemen during each winter. A night school for home owners on firing domestic furnaces smokelessly has been held in school buildings in different parts of the city. It is believed that Nashville is the first public-school system in the country to include a short course in its curriculum on firing and taking care of domestic furnaces. There are only four recitation hours during the term for the eighth grade boys, but it gets them at an impressionable age and has proved of great benefit. Some of these boys have come to the smoke inspector for further information.

From the experience gained in educating the public to the value of smoke abatement the author has come to the conclusion that educational publicity is one of the most important factors in the work. The idea has been formulating for some time that the individual efforts that are being made along this line might be organized, unified, and grouped to a certain extent into a general program of national publicity and education.

A prosperous business in these progressive times is usually one that has such confidence in its product that it pays large amounts in national advertising to get it before the people and to stimulate demand. To a certain extent smoke abatement is a product to be sold to the public, and if it is worth promoting at all and engineers have confidence in its possibilities, why should not the tactics of the successful business man be employed in putting it across.

From the convincing results that quite a number of cities are now getting in smoke abatement by merely local and disunited efforts, it seems possible that the time is almost at hand when such a procedure might solve the smoke problem for good.

Discussion

George W. Bach.[2] The paper is a complete treatment of the subject of smoke abatement, from the smoke inspector's point of view. While it does not deal with deep theories and Ringelmann chart application, it covers the subject completely. The paper brings out the most important phase of smoke abatement clearly, and that is the education of the public to smoke-abatement work. His use of the radio, illustration booths at expositions, and other intelligent propaganda are the most important steps in the studies of any smoke department.

In only one thing does the writer disagree with Mr. Fisher, and that is in the matter of down-draft furnaces. In the early days of that furnace the writer was thoroughly sold on them, but in the last ten or fifteen years he has reached the conclusion that the down-draft furnace is not a practical means of firing a boiler, and prevents smoke only at one particular rating, smoking badly and operating inefficiently at all other ratings. The writer feels that Mr. Fisher will agree with him as time goes on and he gets more down-draft furnaces in use.

By far a better installation is to eliminate hand-firing entirely and use either underfeed or proper overfeed stokers wherever possible. In the smaller boilers now fired by coke or high fixed-carbon coals, the use of natural gas and oil with proper furnaces is an improvement on hand-firing.

H. E. Fishbough.[3] The paper mentions the Scotch marine boiler as one type the least desirable of all. The best-designed boiler or boiler-room equipment will not operate efficiently if the fireman does not have the proper fuel and instructions for its operation.

Fig. 1 shows a correctly designed and installed Scotch marine boiler that incorporates the fundamental principles of the modern

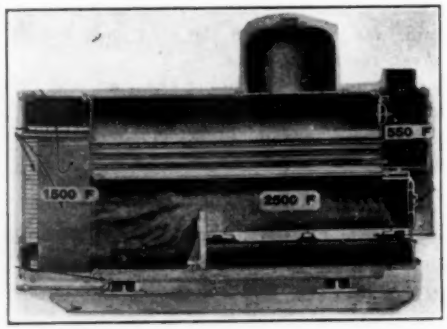


Fig. 1

high-class power unit. The furnace is completely surrounded with water walls. There are no furnace-radiation losses and no furnace refractories to be replaced from time to time. A refractory-lined combustion chamber insures complete combustion of all gases.

From many exhaustive tests the temperature of the gases, taken with electrical and optical pyrometers in passing through the boiler, was 2500 deg. fahr. furnace temperature, 1500 deg. fahr. in the combustion chamber, and 550 deg. fahr. in the flue gas passing to the stack. The flue gases averaged from 10 to 12 per cent CO_2 hand-fired.

Fuel is the next consideration. It is necessary to select the proper size of coal for stoker equipment, and it is just as important to select the proper size of coal for a hand-fired boiler if one expects to eliminate the smoke.

[2] Vice-President and General Manager, Union Iron Works, Erie, Pa. Mem. A.S.M.E.

[3] James Leffel & Co., Springfield, Ohio. Mem. A.S.M.E.

Fig. 2 represents a 3-in., a 2-in., and a 1-in. cube of coal drawn to relative scale and surrounded with a cube of air necessary to produce combustion at 12 per cent CO_2. The 3-in. cube requires approximately 309,500 cu. in. of air; the 2-in. cube requires approximately 91,800 cu. in. of air; and the 1-in. cube requires approximately 11,500 cu. in. of air.

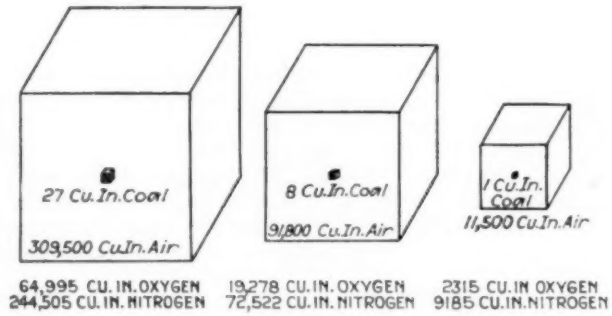


FIG. 2

From this it is apparent that it will be much less difficult to bring to the 1-in. cube of coal the 11,500 cu. in. of air required for its efficient combustion than it will be to supply the 3-in. cube of coal with 309,500 cu. in. of air and secure the intimate contact necessary for efficient and smokeless results. This conclusion has been confirmed by the results obtained on many trial runs made on the Scotch marine boilers.

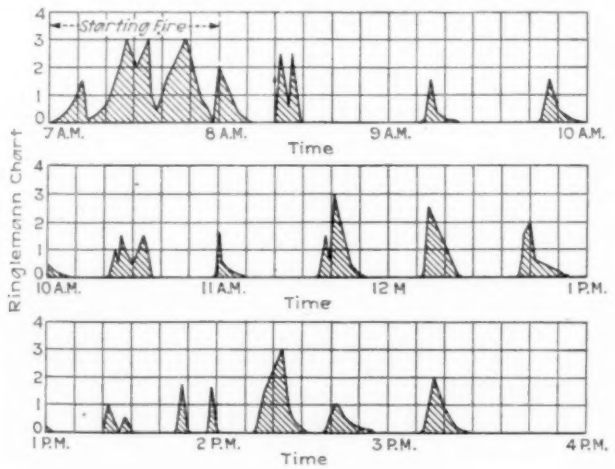


FIG. 3

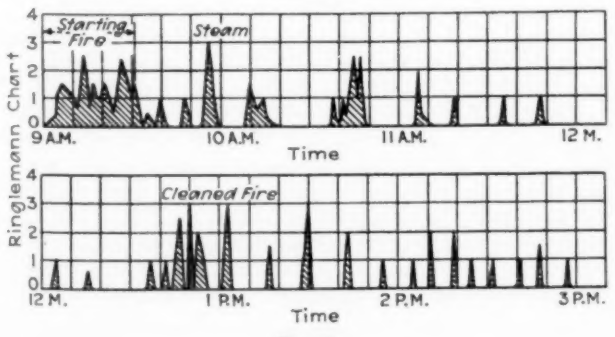


FIG. 4

Ringelmann's charts were used in recording the smoke densities for the charts shown in Figs. 3 and 4.

Fig. 3 shows the observations taken from a 30-hp. Scotch marine boiler with very poor stack conditions. The observations were without the fireman's having any knowledge of being observed. This plant uses egg-size coal consisting of the 2- and 3-in. cubes. The analysis was as follows: Fixed carbon, 47.14 per cent; volatile, 38 per cent; ash, 14.86 per cent. The period from 7 to 8 a.m. was used in starting and raising steam.

Fig. 4 gives the observations from a 40-hp. Scotch marine boiler with good stack conditions. This plant uses nut-size coal of 1-in. cube or less, having an analysis as follows: Fixed carbon, 54.88 per cent; volatile, 36.54 per cent; ash, 7.28 per cent. The period from 9 to 9:30 a.m. was used in starting fire and raising steam.

The writer draws the following conclusions: (1) The equipment must be designed and installed properly; (2) the fuel must be of a size adaptable to the furnace; (3) many stoker-fired plants are found smoking as well as hand-fired plants, and the smoke problem can be reduced only by the careful cooperation of the fireman and the smoke inspector.

AUTHOR'S CLOSURE

It is gratifying to note that the views of Mr. Bach in regard to smoke abatement are in such close accordance with those of the author.

In the matter of down-draft boilers, although no such definite statement was made in his discussion, the author has taken the liberty of believing that Mr. Bach was making special reference to power boilers 100 hp. and over and with relatively high and variable ratings. With such equipment the author agrees with Mr. Bach that the down-draft boilers are a poor substitute for underfeed stokers.

The author will go even farther in saying that he is more than favorable toward the use of underfeed stokers in heating boilers from 10,000 sq. ft. of radiation down to the small residence boiler. It is an accepted fact that average hand firing on any type of furnace is much inferior to stoker firing. Smoke-abatement officials are very fortunate in the fact that the trend of combustion equipment is toward stokers even down to the smallest sizes. Notwithstanding the great increase in stoker sales, their use in the smaller plants is comparatively new, and a recommendation to a prospective purchaser that he not only buy a good boiler but that it would be a paying investment for him to also make an additional outlay of capital for a stoker which will cost more than the boiler itself often serves as a severe shock to his sense of money values. The tendency of the average builder is to decrease the original apportionment for a heating plant as made by the architect and transfer the money thus obtained to the purchase of other things for the building rather than to increase the original cost of the heating plant for better service and economy. It is often difficult to persuade him to buy a serviceable boiler of sufficient capacity. In such cases the logical procedure is to promote the purchase of the most satisfactory hand-fired boiler, which, in the author's mind, is the down-draft.

Regarding the discussion presented by Mr. Fishbough, the author wishes to say that it will be necessary, in view of more recent information, to retract that reference to the Scotch marine boiler which states that "as a rule it is used in sizes too small for the insertion of an underfeed stoker."

The author was recently apprised of the fact that at least two manufacturers of the Scotch marine boiler are considering the adaptation of underfeed stokers to the smaller sizes of this particular boiler. Their efforts in this direction are most commendable, and a satisfactory combination of these two types of equipment will form a basis for agreement among all parties concerned.

FSP-52-11

Relative Economy of Pulverized Coal, Oil, and Gas as Boiler-Plant Fuels

By MARTIN FRISCH,[1] NEW YORK, N. Y.

Where plants are so located that several fuels are available, the management must determine which fuel is most economical. An analysis of the cost of making steam with each of the available fuels is necessary. The various commonly used fuels are compared as to their relative economy. The influence of each fuel on overall plant performance and economy is considered. The paper sums up the relative plant efficiency and the relative plant economy to be expected of the various fuels.

MOST steam plants are restricted in their choice of fuels by their location and by other considerations. But some plants are so located that several fuels of the same type or even of different types become interesting as fuel possibilities. The managers of such plants have to determine which of the available fuels will be most economical to use. The answer to this question is not necessarily the name of the fuel which may be delivered to the plant at the lowest cost per heat unit. This is a well-understood fact, and a careful and complete analysis of the final cost of making steam with each of the available fuels is necessary. But these analyses are too laborious to be made often. For that reason it is of interest to examine the various commonly used fuels and to compare their relative economy for steam making. The purpose of this paper is to present the results of such a comparison.

The ten fuels whose compositions and other properties are given in Table 1 are typical of fuels commonly employed for steam making, and they have been used as the basis of the comparison.

The fuel analysis is only a partial guide to the dollar value of a fuel, and a knowledge of physical characteristics determinable by laboratory tests only incompletely supplements the story told by the analysis. In order to evaluate the relative economy of various fuels it is necessary to consider the influence of each fuel on overall plant performance and economy, preferably on the basis of actual plant experience with each fuel. This necessarily also forces an investigation of the influence of each fuel on plant design and plant cost.

This paper will consider, first, the relative plant efficiency and, second, the relative plant economy to be expected with the various fuels.

TABLE 1 TYPICAL FUELS USED FOR STEAM MAKING

	Texas lignite	Illinois bituminous	Pittsburgh bituminous	Pocahontas coal	River anthracite	Oil		Blast-furnace gas	Natural gas No. 1	Natural gas No. 2	Coke-oven gas
Ultimate Analysis, Per Cent by Weight											
C	39.6	68.0	75.4	80.7	77.6	84.3	CO	27.6	25.6	....	15.2
H_2	3.0	4.3	4.8	4.7	2.3	12.7	H_2	0.2	7.0	....	9.7
O_2	10.0	8.3	6.1	4.5	3.6	1.0	CH_4	0.4	30.3	97.5	41.3
N_2	0.9	1.5	1.4	1.3	0.8	0.2	CO_2	12.1	16.0	....	3.0
S	0.5	1.4	1.4	1.2	0.7	0.8	O_2	0.1	0.8	....	....
Ash	10.0	10.0	8.5	5.0	14.0	....	N_2	59.6	7.3	2.5	30.8
Moisture	36.0	6.5	2.4	2.6	1.0	1.0	C_2H_4	....	5.6	....	....
							C_2H_6	....	7.4	....	....
Proximate Analysis, Per Cent by Weight											
Volatile	32.0	37.0	34.5	18.2	8.2						
Fixed carbon	22.0	46.5	54.6	74.2	76.8						
Ash	10.0	10.0	8.5	5.0	14.0						
Moisture	36.0	6.5	2.4	2.6	1.0						
Other Properties											
Heat value, B.t.u. per lb.	7000	11,700	13,400	14,500	12,450	18,600		1426	15,530	23,400	16,354
Fusing point of ash, deg. fahr.	2200	2000	2100	2500	2400						
Theoretical air required	5.21	9.04	10.14	10.77	9.63	14.12		0.82	10.3	16.8	10.86
Flue-Gas Analysis, Per Cent by Volume (For the plant comparison)											
CO_2	15.0	15.0	15.0	15.0	15.0	13.0		21.6	11.3	10.2	8.4
O_2	4.3	4.2	3.9	4.1	5.0	3.1		0.9	2.0	1.2	1.9
N_2	80.7	80.8	81.1	80.9	80.0	83.9		77.5	86.7	88.6	89.7

Comparative Boiler-Plant Efficiency With Various Fuel

It is a matter of common knowledge among power-plant engineers that a given boiler plant cannot burn with equal efficiency all fuels to which the plant may be readily adaptable. The maximum efficiency attainable with any given fuel in a given plant depends on the character of the fuel. Not only the furnace efficiency, but also the boiler heat-absorbing capacity, the flue-gas temperature, and the draft loss depend on the character of the fuel. The boiler has an uncanny way of guessing changes in its furnace diet, as will be shown.

The influence of each of the ten fuels of Table 1 on the performance of a certain boiler was investigated, and the maximum efficiencies attainable with furnaces and firing equipment most suitable to develop the highest efficiency possible with each fuel were determined.

Since all of the fuels considered may be burned in suspension and some of them cannot be burned any other way, it will be understood readily that for the purpose of this investigation, embracing solid, liquid, and gaseous fuels, pulverized-fuel firing was considered only, in order to simplify the comparison. The conclusions of the investigation, which were checked against the results of comparable actual boiler tests, confirm the value of an index of boiler performance already suggested by others. This is the amount of sensible heat imparted to the products of

[1] Engineer in Charge of Development, Combustion Engineering Corporation.

Presented at the Third National Fuels Meeting, Philadelphia, Pa., October 7 to 10, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

combustion per pound passed through the boiler. The magnitude of this index depends on (1) the heat value of the fuel, (2) the weight of gas and vapor formed in the combustion of the fuel, and (3) amount of heat made unavailable by the incomplete combustion of the fuel and the evaporation of moisture and the formation of water vapor by the burning of hydrogen in the fuel.

The effect of incomplete combustion losses on the index is small because the reduction in available heat due to this cause is compensated for by the reduction in the total amount of gas formed per pound of fuel by a quantity proportional to the amount of unburned fuel.

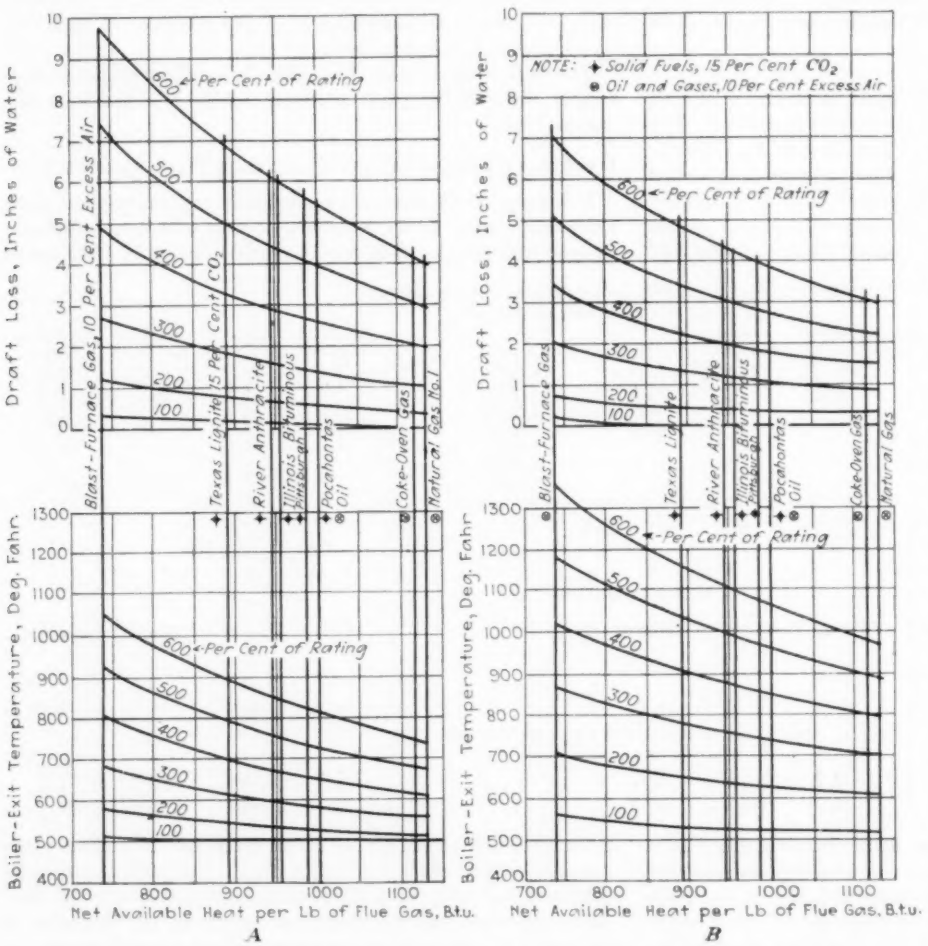


FIG. 1 SENSIBLE HEAT IN FURNACE GASES IS A GOOD INDEX OF BOILER PERFORMANCE
(*A*, typical straight-tube boiler; *B*, typical bent-tube boiler.)

The amount of excess air used influences the magnitude of the index by its effect on the total quantity of gas formed per available heat unit.

The moisture in the fuel and the vapor formed by the burning of the hydrogen have the greatest effect on the magnitude of the index, because the effect is twofold. First, the total quantity of gas formed per available heat unit varies with the amount of vapor formed, and second, the latent heat absorbed in the formation of vapor reduces the amount of sensible heat available. The index evidently depends greatly on the fuel characteristics.

Fig. 1 summarizes the relationships existing between the index (the available sensible heat) and the three factors which measure boiler performance; namely, capacity, draft loss, and exit temperature. A typical sectional header and a typical bent-tube boiler are compared in Fig. 1.

The greater the available sensible heat per pound of gas passed through the boiler the greater the capacity developed with a given draft loss. This can be pictured another way, as in Fig. 2, where the decrease in the temperature of identical weights of the products of combustion of natural gas and blast-furnace gas as more and more heating surface is passed over is shown. Each fuel was burned with 10 per cent excess air. Although the combustion temperature of the natural gas was almost 1000 deg. fahr. higher than that of the blast-furnace gas, the boiler-exit temperatures were within 35 deg. of each other. The heat given up to the boiler by the products of the natural gas and the capacity developed were 55 per cent greater, but the draft loss was very nearly the same in the two cases. An extension of such comparisons to the other fuels of Table 1 makes it possible, as shown by Fig. 1, to determine the boiler performance to be expected with each fuel, in accordance with the amount of excess air used in its combustion, and to determine the exit temperatures to be expected at any rating with the several fuels, as in Fig. 3, and the efficiencies, as in Fig. 4. It is necessary, when deriving the efficiencies peculiar to each fuel, to take into account the fact that at any rating and excess air the exit temperatures will vary with the fuel as in Fig. 3, which is based on 15 per cent CO_2 (approximately 20 per cent excess air) in the products of the solid fuels, 13 per cent CO_2 (15 per cent excess air) in the products of oil, and 10 per cent excess air in the combustion of gaseous fuels.

If the air used for combustion is preheated, the sensible heat available per pound of flue gas increases, because the heat available per pound of fuel is increased without a corresponding increase in the weight of the products of combustion. The effect of this is to increase the capacity attainable with a given weight of gas and with a given exit temperature so that at any given capacity the boiler efficiency will actually be higher than it would be with cold air and the draft loss lower. This gain, which increases with the boiler capacity, is additional to the gain which results from the actual recovery of heat in the air preheater. The effect of the use of preheated air on the boiler-exit temperature and on the efficiency is shown in Fig. 5. At 500 per cent of rating the increase in efficiency is 1 per cent and the decrease in draft loss over $^3/_4$ in.

The differences in boiler-exit temperatures at any given rating only partly account for the differences in efficiency attainable with the various fuels. There are also considerable differences in the completeness with which the various solid fuels may be

burned and in the latent heat losses (which particularly affect the oil and gaseous fuels adversely) inevitable with the various fuels. The gaseous fuels and oil may be burned with negligible incomplete combustion losses. However, this is not true of solid fuels, whether burned in suspension or on grates. Experience indicates that if the solid fuels are burned in suspension

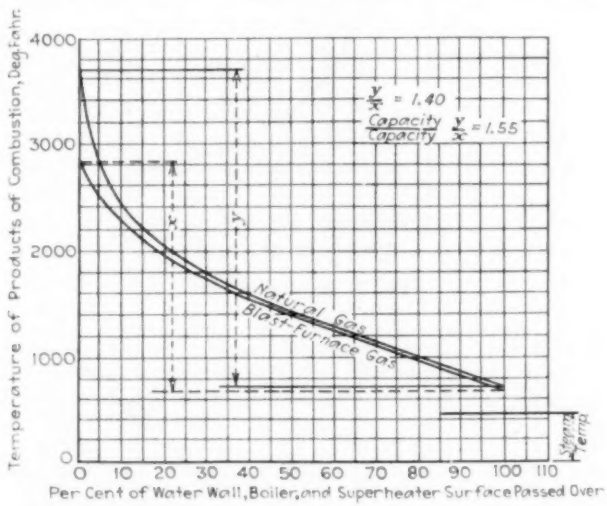


FIG. 2 FLAME TEMPERATURE HAS MORE EFFECT ON BOILER CAPACITY AND THE AMOUNT OF HEAT ABSORBED THAN ON EXIT TEMPERATURES

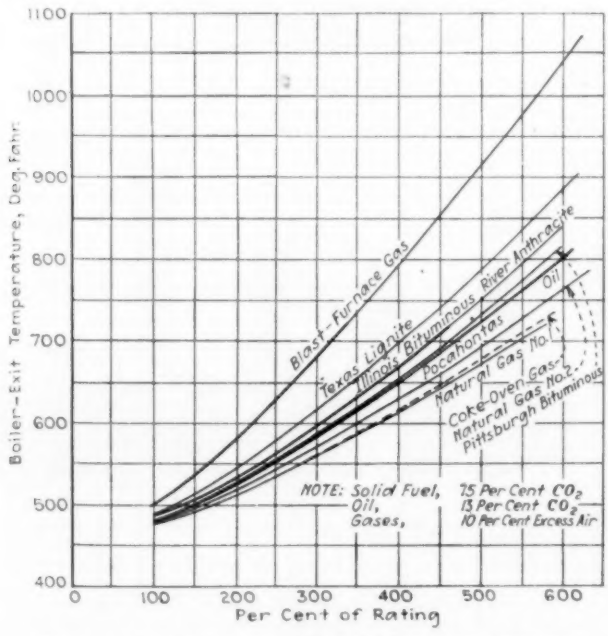


FIG. 3 AT ANY RATING THE EXIT TEMPERATURE DEPENDS ON THE FUEL

after being pulverized to the fineness customary for each fuel the losses due to incomplete combustion will be about as shown in Fig. 6. The high fixed-carbon coals are most difficult to burn completely in suspension and will suffer the greatest wastage due to incomplete combustion. As the fixed carbon decreases, the carbon loss to be expected, all other things remaining equal, also decreases.

This can be substantiated by a study of a large number of tests on pulverized-lignite fuels, Western sub-bituminous coals, Midwestern bituminous, and Eastern coals, including river anthracite. Fig. 6 is based on the conclusions of such a study.

The amount of heat lost by radiation at any capacity will be

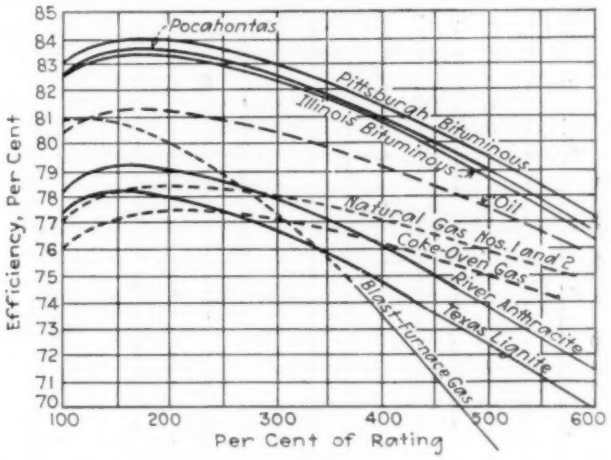


FIG. 4 THE BEST EFFICIENCIES ATTAINABLE WITH DIFFERENT FUELS VARY WIDELY
(Based on Figs. 1, 3, 6, and 7.)

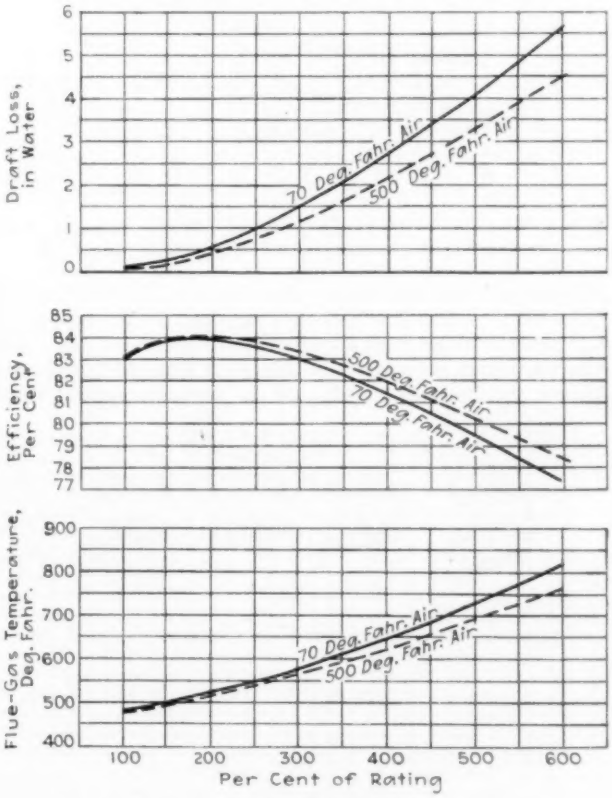


FIG. 5 PREHEATING THE COMBUSTION AIR IMPROVES BOILER PERFORMANCE

very nearly the same for all of the fuels, and no great error in the conclusions of this paper will result from assuming the radiation losses to be as shown in Fig. 7.

If the heat balances of all the fuels be compared, as in Fig. 8, the incomplete combustion will be found to increase in this order:

(1) Lignite, (2) Illinois bituminous, (3) Pittsburgh bituminous, (4) Pocahontas semi-bituminous, and (5) river anthracite. The latent heat losses due to the burning of hydrogen and the evaporation of moisture increase in the inverse order. No incomplete-

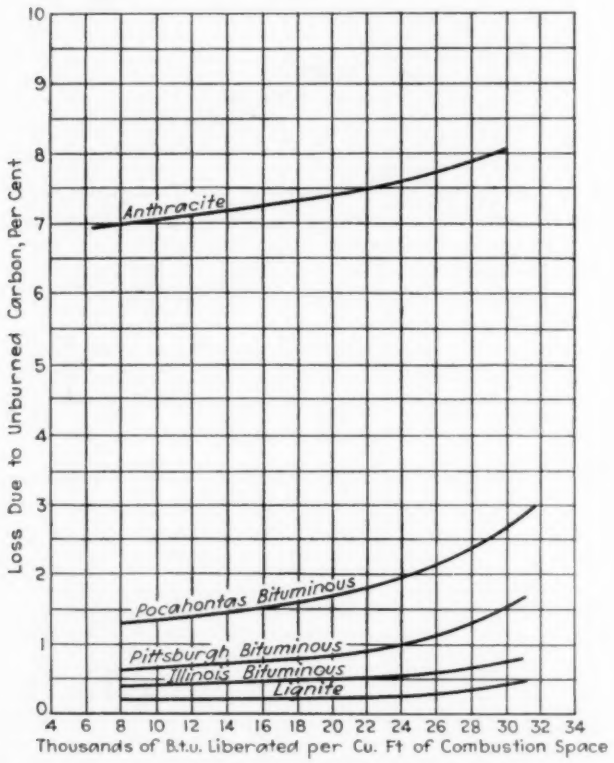


FIG. 6 INCOMPLETE COMBUSTION LOSSES WHEN BURNING PULVERIZED FUELS DEPEND ON THE HISTORY OF THE FUEL

combustion losses are expected when burning oil and the gaseous fuels, but the latent heat losses are very high for these fuels and may exceed 10 per cent for natural and coke-oven gas. The attainable efficiencies are very nearly the same for Pittsburgh, Illinois, and Pocahontas coals. Of the particular coals of Table 1, Pittsburgh coal can be burned with about $^1/_2$ per cent greater efficiency than Pocahontas coal under the assumed conditions,

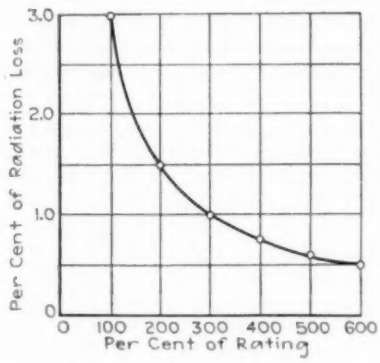


FIG. 7 RADIATION LOSS

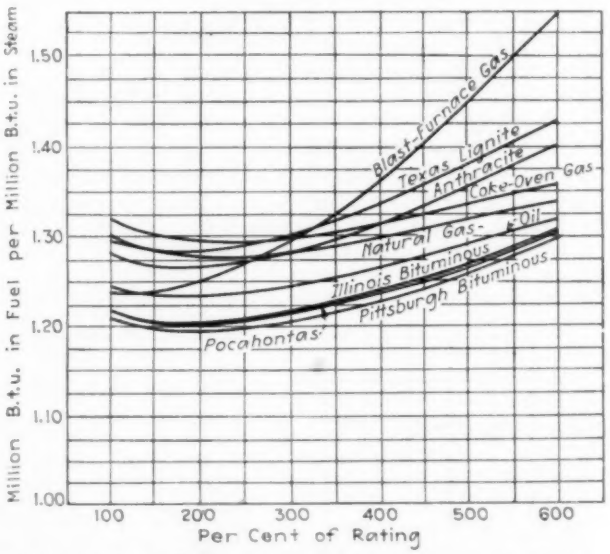


FIG. 9 HEAT UNITS REQUIRED BY FURNACE PER HEAT UNIT IN STEAM

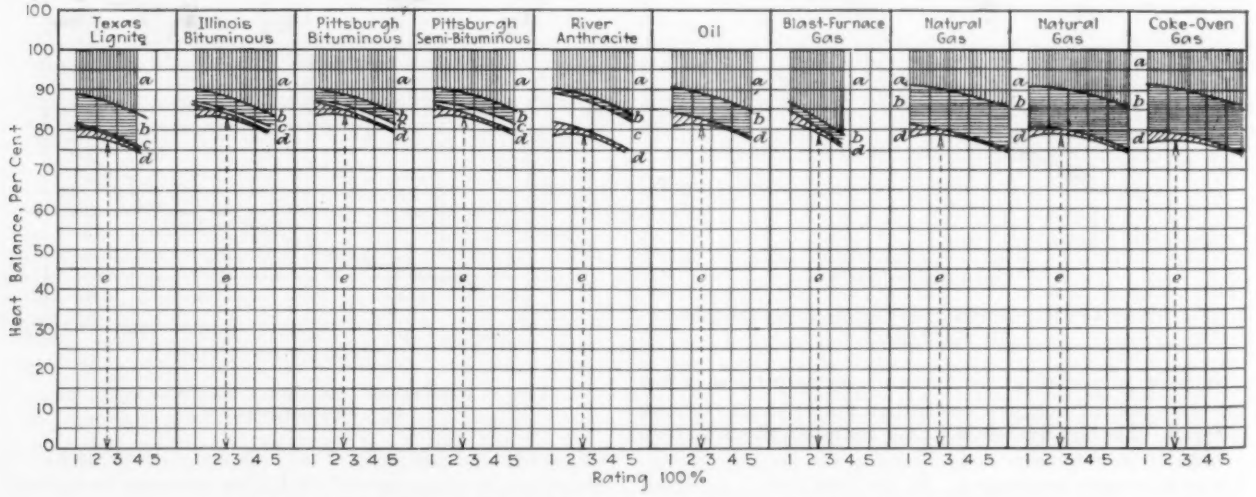


FIG. 8 HEAT BALANCE WITH A GIVEN FIRING METHOD DEPENDS ON THE FUEL

The carbon loss increases as the age of the coal increases. The latent-heat loss depends on the water and hydrogen in the fuel

(*a*, loss due to sensible heat of products of combustion (above 70 deg. fahr.); *b*, loss in latent heat of water vapor in products of combustion; *c*, loss due to incomplete combustion of carbon; *d*, radiation loss; *e*, heat absorbed, efficiency.)

chiefly because of the lower carbon losses to be expected with Pittsburgh coal, and with about ¾ per cent greater efficiency than Illinois coal, because of the somewhat higher exit temperature to be expected with Illinois coal at a given capacity and excess air. Maximum efficiencies attainable with river anthracite and Texas lignite are 5 to 6 per cent lower than with Pittsburgh coal on account of higher exit temperatures and carbon losses with river anthracite and on account of higher exit temperatures and latent heat losses with lignite at identical capacities and excess air ratios. The fact that oil and gaseous fuels can be burned with lower excess air than is the case with the solid fuels was taken into account in comparing the best efficiencies peculiar to these fuels. On this basis mechanically atomized oil can be burned with efficiencies 2 to 2¾ per cent under those to be expected with Pittsburgh coal because of the larger latent heat loss. Steam atomization will have the effect of lowering the boiler-performance index of the oil and will cause higher exit temperature and draft losses at any given capacity. Blast-furnace gas has a very steep efficiency curve on account of large sensible heat losses due to the large amount of flue gas into which the available heat must spread. Efficiencies attainable with blast-furnace gas will be 2 to 20 per cent lower than with Pittsburgh coal. Coke-oven and natural gas can be burned efficiently, but on account of latent heat losses of over 10 per cent or more, the overall efficiencies to be expected with these fuels will run 3 to 5 per cent lower than with Pittsburgh coal.

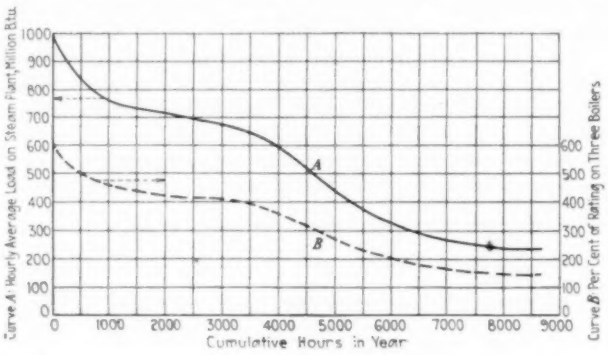


FIG. 10 BOILER-PLANT LOAD-DURATION DIAGRAM FOR COMPARING FUELS UNDER THE SAME LOAD CONDITIONS

The significance of these efficiency differences is to be found in Fig. 9, which shows the number of heat units which must be purchased in different fuels and supplied to the furnace per million B.t.u. of heat absorbed by the boiler.

COMPARATIVE BOILER-PLANT ECONOMY WITH VARIOUS FUELS

The character of the fuel affects not only the number of fuel heat units required per steam heat unit but also the amount of money that must be expended in order to properly burn each fuel. The final cost of steam will depend on the following factors, some common to all fuels and others peculiar to each fuel as shown:

Common to All Fuels.

(*a*) Fixed operating and maintenance costs on boiler plant exclusive of fuel and fan equipment.

Peculiar to Each Fuel.

(*a*) Cost of fuel.

(*b*) Fixed operating and maintenance charges on fuel storage, handling, preparation, and burning equipment.

(*c*) Fixed charges and operating charges on fans and draft equipment.

(*d*) Fixed charges on plant capacities reserved to provide peak requirements of the fuel and fan equipment.

Although a steam plant of a given capacity designed to burn some particular fuel or fuels might cost almost anything, depending on the design and on the labor and economic conditions at the time of construction, it will nevertheless be instructive to determine as accurately as possible to what extent the cost of the boiler plant and the fixed and operating and maintenance

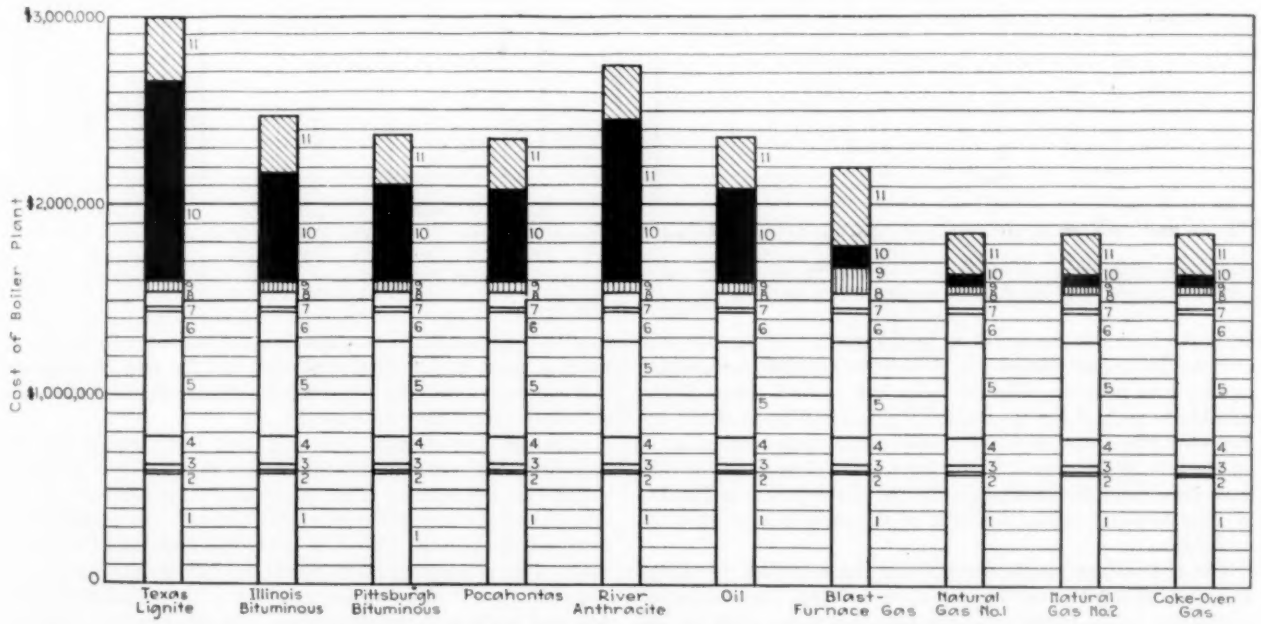


FIG. 11 COMPARATIVE COST OF BOILER PLANTS UTILIZING VARIOUS FUELS

(The figures indicate the total cost, installed, as follows: 1, boilers, superheaters, water walls, settings; 2, meters, instruments; 3, boilerfeed pumps; 4, stacks, breechings, insulation; 5, foundations, structure; 6, wiring; 7, site development; 8, engineering; 9, draft equipment, fans; 10, fuel receiving, storage, handling, preparation, firing equipment; 11, reserve-plant cost for peak-load auxiliary-power requirements of boiler plant.)

costs will depend on the fuel which the plant is intended to utilize. Data on the complete cost of building and operating steam plants were studied, and on the basis of this study the probable fixed, operating, and maintenance costs of steam plants most suitable to any one or more of the fuels of Table 1 were determined. Also, the fuels which will produce steam at the lowest price and the relative economy of all the fuels were ascertained.

The cost of building and operating steam plants best suited to meet the same annual steam-demand curve, say as of Fig. 10, and best suited to each of the ten fuels of Table 1 has been calculated in order to make a concrete comparison of the cost of steam with the various fuels. It is assumed that 1,000,000 lb. of steam per hour is the peak capacity of each plant, and for this purpose four boilers, each capable of evaporating 350,000 lb. of steam per hour at peak load, are assumed, three boilers to carry the load and the fourth to be the spare. Air heaters and economizers are not considered, in order to keep the comparison as simple as possible. The probable cost of such steam plants suitable to each of the ten fuels of Table 1 will be as shown by Fig. 11. The cost of each plant, exclusive of those costs affected by the fuel, is the same. Those costs which depend on the fuel have been carefully ascertained for the development of Fig. 11, in order to arrive at the total cost of the plant most suitable to each fuel.

The cost of equipment required for the receiving, handling, and treating of fuel from the time of its arrival at the plant until its delivery to the furnace ready to burn will vary widely for the several fuels, because of the wide differences in the character and in the quantities of the different fuels to be handled annually, in order to satisfy the load requirements of Fig. 10. Thus the quantities of the several fuels required yearly will be as follows:

Fuel	Tons
Texas lignite	440,000
Illinois bituminous	245,000
Pittsburgh bituminous	212,000
Pocahontas bituminous	196,000
River anthracite	244,000
Oil	157,500
Blast-furnace gas	2,160,000
Natural gas No. 1	195,000
Natural gas No. 2	129,000
Coke-oven gas	185,500

These weights take into account the weighted average operating efficiency over the year, which is 2 per cent lower than the weighted average of the best efficiencies attainable at each load.

The weight of lignite required is more than twice the weight of the other solid fuels. Lignite may arrive at the plant with a moisture content of 30 to 35 per cent, and this has to be reduced to about 27 or 28 per cent in order to make the lignite dry enough to pulverize and burn properly. Drying equipment to accomplish this is required. The cost of the fuel, unloading, storage, handling, preparation, and burning equipment will be about twice as great as for Pittsburgh coal, and the cost of operating this equipment will also be considerably more than for the other solid fuels except river anthracite.

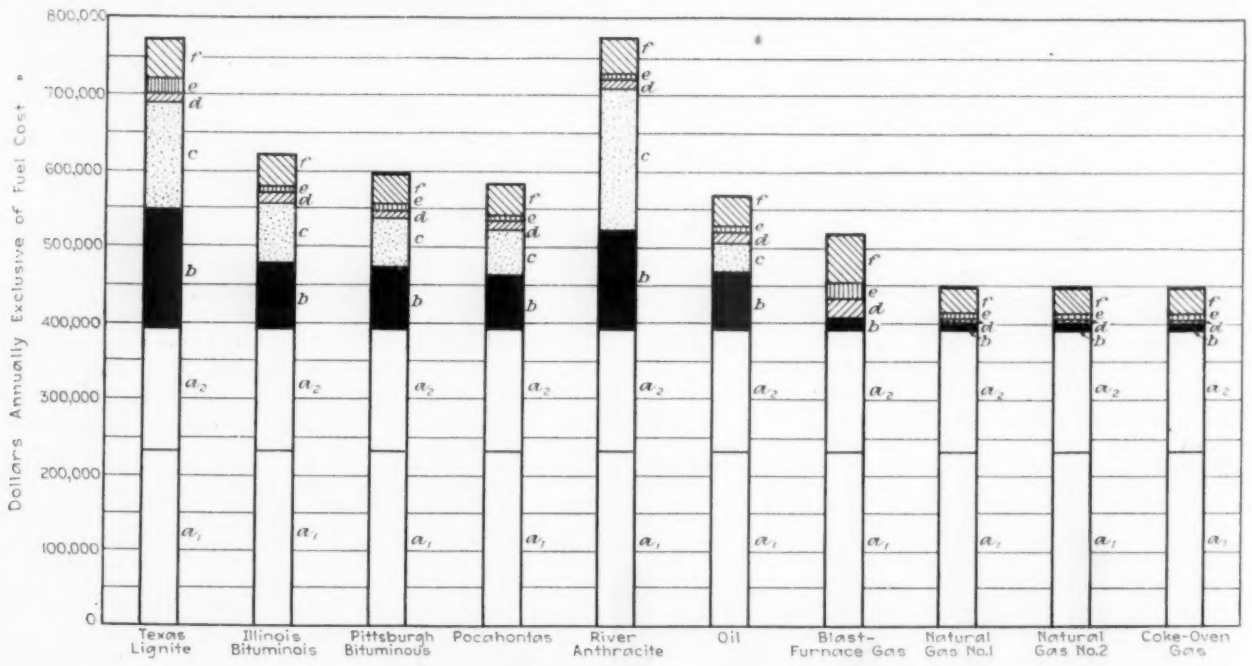


FIG. 12 YEARLY COST OF MAKING STEAM EXCLUSIVE OF FUEL COST WITH VARIOUS FUELS

(a_1, fixed charges at 15% on boiler plant exclusive of fuel equipment $[1 + 2 + 3 + 4 + 5 + 6 + 7 + 8] \times 0.15$; b, fixed charges at 15% on fuel equipment and fans $[9 + 10] \times 0.15$; f, fixed charges at 15% on plant capacity reserved for boiler-plant peak; e, fixed charges on draft equipment; a_2, operating charges on boiler plant, not including draft and fuel equipment; c, operating charges on fuel equipment; d, operating charges on draft equipment.)

Although the storage, handling, and firing equipment for anthracite will cost but little more than that required for Pittsburgh coal, the capacity of the pulverizing plant for river anthracite coal will be at least twice as great because of its greater hardness and also because it must be pulverized finer than Pittsburgh coal. The pulverizing-plant operating and maintenance costs also will exceed those to be expected with bituminous coals, and may easily with certain types of equipment be three times as high. Pocahontas coal, on the other hand, is very easy to pulverize, and the capacity of a given grinding plant may be 30 to 50 per cent greater when handling this fuel than with Pittsburgh coal.

Oil requires heavy outlays for storage facilities, while the gaseous fuels will call for but very modest outlays for fuel handling.

All equipment and plant provisions peculiar to each fuel to be

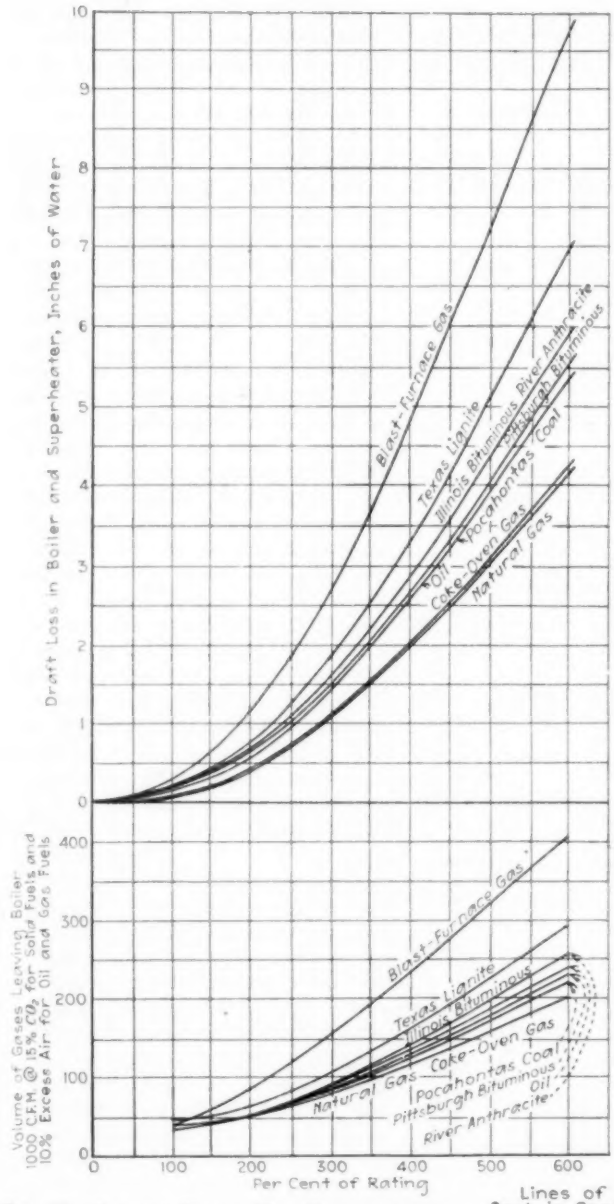


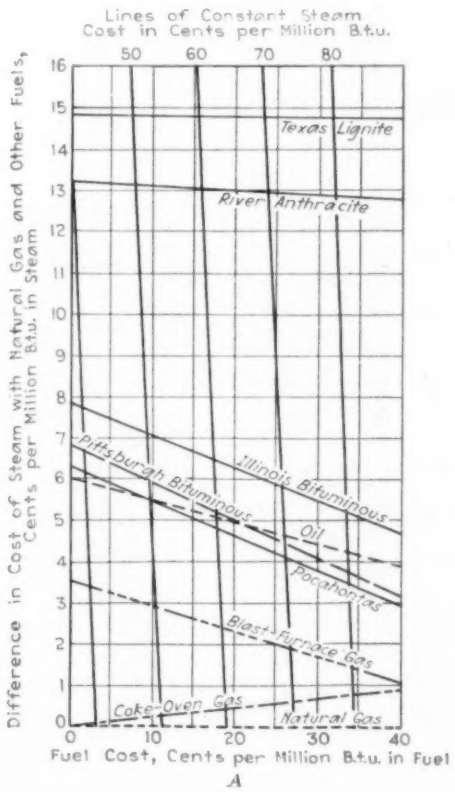
FIG. 13 INDUCED-DRAFT FAN REQUIREMENTS ALSO DEPEND ON THE FUEL

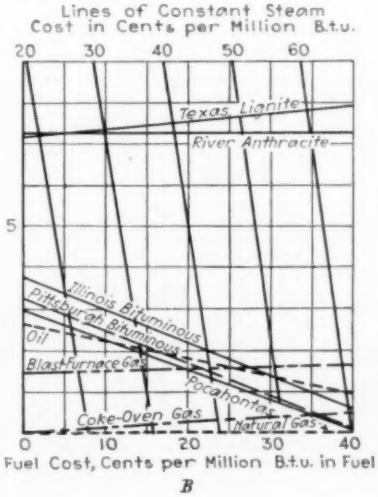
handled have been considered in computing the cost of each steam plant (as finally shown in Fig. 11) for the utilization of each fuel in accordance with the postulated load requirements. The fixed, operating, and maintenance costs peculiar to each fuel were also computed, as shown in Fig. 12.

Differences in the fixed, operating, and maintenance costs peculiar to the several fuels are not entirely accounted for by differences in the fuel equipment, but also by differences in fixed and operating costs, due to differences in the draft equipment required by the several fuels and by differences in the supplementary plant capacity that must be reserved to accommodate the peak-power requirements of the steam plant. The induced-draft fan requirements will be quite different for each fuel, as shown by Fig. 13. Furthermore, the peak-power requirements will be different for the various fuels, and reserve or supplementary plant capacity must be provided in different amounts to satisfy the peak requirements.

Fig. 12 shows the magnitude of the various items entering into the yearly cost of making steam, exclusive of the cost of fuel. The differences in the carrying and other charges which depend on the fuel are striking.

The relative economy of the several fuels on a basis which is





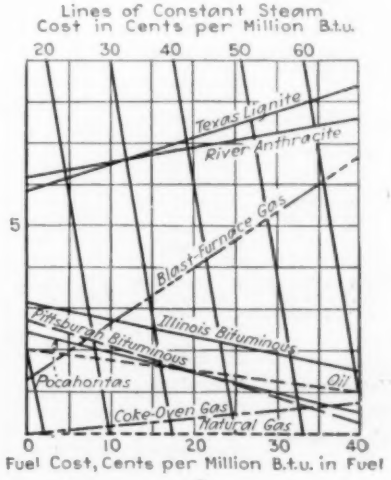


FIG. 14 (RIGHT) COMPARATIVE COST OF STEAM WITH VARIOUS FUELS

(*A*, 170,000 lb. steam per hour plant load, approximately 100% rating on three boilers; *B*, 500,000 lb. steam per hour plant load, approximately 300% rating on three boilers; *C*, 830,000 lb. steam per hour plant load, approximately 500% rating on three boilers.)

TABLE 2 MAXIMUM PRICE TO BE PAID FOR VARIOUS FUELS TO PRODUCE STEAM AT THE SAME COST

(In dollars per ton of solid fuels; in dollars per barrel of oil; in cents per 1000 cu. ft. of gas)

		Total steam cost per 1,000,000 B.t.u. — 30 cents							40 cents							50 cents						
Fuels designed for	Fuel actually used	River anthracite	Lignite	Illinois	Pittsburgh	Pocahontas coal	Oil	Natural gas	River anthracite	Lignite	Illinois	Pittsburgh	Pocahontas coal	Oil	Natural gas	River anthracite	Lignite	Illinois	Pittsburgh	Pocahontas coal	Oil	Natural gas
Lignite only	Lignite	..	1.34	..	..	..	..	..	..	2.38	..	..	..	..	..	..	3.41	..	..	..	..	..
	Illinois	..	..	2.66	..	..	..	..	..	..	4.52	..	..	..	..	..	..	6.38	..	..	..	..
	Pittsburgh	..	..	..	3.15	..	..	..	..	..	..	5.30	..	..	..	..	..	..	7.45	..	..	..
	Pocahontas	..	..	..	..	3.42	..	..	..	..	..	..	5.74	..	..	..	..	..	..	8.06	..	..
Lignite and natural gas	Lignite	..	1.31	..	..	..	..	..	..	2.35	..	..	..	..	..	..	3.38	..	..	..	..	..
	Illinois	..	..	2.59	..	..	..	..	..	..	4.45	..	..	..	..	..	..	6.31	..	..	..	..
	Pittsburgh	..	..	..	3.07	..	..	..	..	..	..	5.22	..	..	..	..	..	..	7.37	..	..	..
	Pocahontas	..	..	..	..	3.33	..	..	..	..	..	..	5.64	..	..	..	..	..	..	7.96	..	..
	Nat. gas	..	..	..	..	..	..	12.1	..	..	..	..	..	..	19.8	..	..	..	..	..	..	27.2
Lignite and oil	Lignite	..	1.18	..	..	..	..	..	..	2.22	..	..	..	..	..	..	3.25	..	..	..	..	..
	Illinois	..	..	2.35	..	..	..	..	..	..	4.21	..	..	..	..	..	..	6.07	..	..	..	..
	Pittsburgh	..	..	..	2.79	..	..	..	..	..	..	4.94	..	..	..	..	..	..	7.09	..	..	..
	Pocahontas	..	..	..	..	3.04	..	..	..	..	..	..	5.36	..	..	..	..	..	..	7.68	..	..
	Oil	..	..	..	..	..	0.62	..	..	..	..	..	..	1.07	..	..	..	..	..	..	1.53	..
Lignite and natural gas and oil	Lignite	..	1.15	..	..	..	..	..	..	2.19	..	..	..	..	..	..	3.22	..	..	..	..	..
	Illinois	..	..	2.30	..	..	..	..	..	..	4.16	..	..	..	..	..	..	6.02	..	..	..	..
	Pittsburgh	..	..	..	2.72	..	..	..	..	..	..	4.87	..	..	..	..	..	..	7.02	..	..	..
	Pocahontas	..	..	..	..	2.97	..	..	..	..	..	..	5.29	..	..	..	..	..	..	7.61	..	..
	Oil	..	..	..	..	..	0.61	..	..	..	..	..	..	1.06	..	..	..	..	..	..	1.52	..
	Nat. gas	..	..	..	..	..	..	10.7	..	..	..	..	..	..	18.3	..	..	..	..	..	..	25.8
Illinois coal only	Illinois	..	..	3.02	..	..	..	..	..	..	4.88	..	..	..	..	..	..	6.74	..	..	..	..
	Pittsburgh	..	..	..	3.55	..	..	..	..	..	..	5.70	..	..	..	..	..	..	7.85	..	..	..
	Pocahontas	..	..	..	..	3.90	..	..	..	..	..	..	6.22	..	..	..	..	..	..	8.54	..	..
Illinois coal and natural gas	Illinois	..	..	3.00	..	..	..	..	..	..	4.86	..	..	..	..	..	..	6.72	..	..	..	..
	Pittsburgh	..	..	..	3.51	..	..	..	..	..	..	5.66	..	..	..	..	..	..	7.81	..	..	..
	Pocahontas	..	..	..	..	3.85	..	..	..	..	..	..	6.17	..	..	..	..	..	..	8.49	..	..
	Nat. gas	..	..	..	..	..	..	13.5	..	..	..	..	..	..	21.0	..	..	..	..	..	..	28.5
Illinois coal and oil	Illinois	..	..	2.72	..	..	..	..	..	..	4.58	..	..	..	..	..	..	6.44	..	..	..	..
	Pittsburgh	..	..	..	3.20	..	..	..	..	..	..	5.35	..	..	..	..	..	..	7.50	..	..	..
	Pocahontas	..	..	..	..	3.51	..	..	..	..	..	..	5.83	..	..	..	..	..	..	8.15	..	..
	Oil	..	..	..	..	..	0.70	..	..	..	..	..	..	1.16	..	..	..	..	..	..	1.62	..
Illinois coal and natural gas and oil	Illinois	..	..	2.64	..	..	..	..	..	..	4.50	..	..	..	..	..	..	6.36	..	..	..	..
	Pittsburgh	..	..	..	3.16	..	..	..	..	..	..	5.31	..	..	..	..	..	..	7.46	..	..	..
	Pocahontas	..	..	..	..	3.47	..	..	..	..	..	..	5.79	..	..	..	..	..	..	8.11	..	..
	Oil	..	..	..	..	..	0.69	..	..	..	..	..	..	1.15	..	..	..	..	..	..	1.61	..
	Nat. gas	..	..	..	..	..	..	12.3	..	..	..	..	..	..	19.8	..	..	..	..	..	..	27.3
Pittsburgh coal only	Pittsburgh	..	..	..	3.61	..	..	..	..	..	..	5.76	..	..	..	..	..	..	7.91	..	..	..
	Pocahontas	..	..	..	..	3.96	..	..	..	..	..	..	6.28	..	..	..	..	..	..	8.60	..	..
Pittsburgh coal and natural gas	Pittsburgh	..	..	..	3.57	..	..	..	..	..	..	5.72	..	..	..	..	..	..	5.87	..	..	..
	Pocahontas	..	..	..	..	3.92	..	..	..	..	..	..	6.24	..	..	..	..	..	..	8.56	..	..
	Nat. gas	..	..	..	..	..	..	13.7	..	..	..	..	..	..	21.2	..	..	..	..	..	..	28.7
Pittsburgh coal and oil	Pittsburgh	..	..	..	3.26	..	..	..	..	..	..	5.41	..	..	..	..	..	..	7.56	..	..	..
	Pocahontas	..	..	..	..	3.58	..	..	..	..	..	..	5.90	..	..	..	..	..	..	8.22	..	..
	Oil	..	..	..	..	..	0.72	..	..	..	..	..	..	1.17	..	..	..	..	..	..	1.63	..
Pittsburgh coal and natural gas and oil	Pittsburgh	..	..	..	3.22	..	..	..	..	..	..	5.37	..	..	..	..	..	..	7.52	..	..	..
	Pocahontas	..	..	..	..	3.54	..	..	..	..	..	..	5.86	..	..	..	..	..	..	8.18	..	..
	Oil	..	..	..	..	..	0.71	..	..	..	..	..	..	1.16	..	..	..	..	..	..	1.62	..
	Nat. gas	..	..	..	..	..	..	12.5	..	..	..	..	..	..	20.0	..	..	..	..	..	..	27.5
Pocahontas coal only	Pocahontas	..	..	..	..	3.99	..	..	..	..	..	..	6.31	..	..	..	..	..	..	8.63	..	..
Pocahontas coal and natural gas	Pocahontas	..	..	..	..	3.94	..	..	..	..	..	..	6.26	..	..	..	..	..	..	8.58	..	..
	Nat. gas	..	..	..	..	..	..	13.8	..	..	..	..	..	..	21.3	..	..	..	..	..	..	28.9
Pocahontas coal and oil	Pocahontas	..	..	..	..	3.60	..	..	..	..	..	..	5.92	..	..	..	..	..	..	8.24	..	..
	Oil	..	..	..	..	..	0.72	..	..	..	..	..	..	1.18	..	..	..	..	..	..	1.63	..
Pocahontas coal and natural gas and oil	Pocahontas	..	..	..	..	3.56	..	..	..	..	..	..	5.88	..	..	..	..	..	..	8.20	..	..
	Oil	..	..	..	..	..	0.71	..	..	..	..	..	..	1.17	..	..	..	..	..	..	1.62	..
	Nat. gas	..	..	..	..	..	..	12.6	..	..	..	..	..	..	20.1	..	..	..	..	..	..	27.6
River anthracite coal only	River anthracite	2.40	..	..	..	..	..	..	4.27	..	..	..	..	..	..	6.14	..	..	..	..	..	..
	Pittsburgh	..	..	..	3.34	..	..	..	..	..	..	5.49	..	..	..	..	..	..	7.64	..	..	..
	Pocahontas	..	..	..	..	3.69	..	..	..	..	..	..	6.01	..	..	..	..	..	..	8.33	..	..
River anthracite coal and natural gas	River anthracite	2.38	..	..	..	..	..	..	4.25	..	..	..	..	..	..	6.12	..	..	..	..	..	..
	Pittsburgh	..	..	..	3.31	..	..	..	..	..	..	5.46	..	..	..	..	..	..	7.61	..	..	..
	Pocahontas	..	..	..	..	3.66	..	..	..	..	..	..	5.98	..	..	..	..	..	..	8.30	..	..
	Natural gas	..	..	..	..	..	..	12.7	..	..	..	..	..	..	20.3	..	..	..	..	..	..	27.8
River anthracite coal and oil	River anthracite	2.10	..	..	..	..	..	..	3.97	..	..	..	..	..	..	5.84	..	..	..	..	..	..
	Pittsburgh	..	..	..	2.99	..	..	..	..	..	..	5.14	..	..	..	..	..	..	7.29	..	..	..
	Pocahontas	..	..	..	..	3.31	..	..	..	..	..	..	5.63	..	..	..	..	..	..	7.95	..	..
	Oil	..	..	..	..	..	0.72	..	..	..	..	..	..	1.18	..	..	..	..	..	..	1.63	..
River anthracite coal and natural gas and oil	River anthracite	2.07	..	..	..	..	..	..	3.94	..	..	..	..	..	..	5.81	..	..	..	..	..	..
	Pittsburgh	..	..	..	2.96	..	..	..	..	..	..	5.11	..	..	..	..	..	..	7.26	..	..	..
	Pocahontas	..	..	..	..	3.27	..	..	..	..	..	..	5.59	..	..	..	..	..	..	7.91	..	..
	Oil	..	..	..	..	..	0.71	..	..	..	..	..	..	1.17	..	..	..	..	..	..	1.62	..
	Natural gas	..	..	..	..	..	..	11.5	..	..	..	..	..	..	19.1	..	..	..	..	..	..	26.6
Natural gas	Natural gas	..	..	..	..	..	..	15.2	..	..	..	..	..	..	22.7	..	..	..	..	..	..	30.2
Oil	Oil	..	..	..	..	..	0.79	..	..	..	..	..	..	1.25	..	..	..	..	..	..	1.72	..
Oil and natural gas	Oil	..	..	..	..	..	0.79	..	..	..	..	..	..	1.24	..	..	..	..	..	..	1.70	..
	Natural gas	..	..	..	..	..	..	13.7	..	..	..	..	..	..	21.2	..	..	..	..	..	..	28.8

fair to each fuel can be determined by establishing the maximum price which each of the ten plants especially designed to develop the best economy with each fuel can afford to pay for fuel in order to produce steam at the same cost. This puts the fuels on a bona fide competitive basis. Or, the cost of making steam with the several fuels at a constant cost per fuel heat unit might be compared. Fig. 14 makes it possible to compare the relative economy of the ten fuels of Table 1 on either basis, at several load factors.

At a given cost per fuel heat unit, natural gas is the most economical fuel, and was therefore used as the standard for comparison. The relative economy of the several fuels is shown by the difference between the cost of making steam with natural gas and other fuels at a given fuel cost. Or, the price to be paid per fuel heat unit in order to produce steam at the same price may be determined by noting the intersections of the lines of constant steam cost with the vertical lines of constant fuel cost.

Of course, the absolute costs as established by Fig. 14 are not promiscuously applicable to any and all plants. But their relative magnitude can serve as an index of the comparative economy of certain fuel types under conditions most favorable to each.

But suppose the various plants should be confronted by the necessity of making a choice between their customary fuel and some other offerings. Some of the offerings could probably be handled without any changes in the plant. Some may require additional capital expenditures. For example, a plant designed for pulverized lignite could conveniently handle any of the other solid fuels, and by the installation of additional equipment it could also use natural gas or oil. Such a plant in Texas would probably only have the choice of lignite, natural gas, or oil.

But a lignite plant in St. Paul or Minneapolis or Duluth although primarily designed for using North Dakota lignite also might have to consider Eastern coals shipped by way of the Great Lakes, or even Midwestern coal. Fig. 15 shows the relative magnitudes of the various items entering into the cost of making steam with various fuels in plants designed primarily to handle lignite and which may or may not also have the necessary equipment to handle either natural gas or oil, or both. On the basis of such an analysis extended to the other plants also, the maximum price of each fuel which will result in the same steam cost is summarized in Table 2.

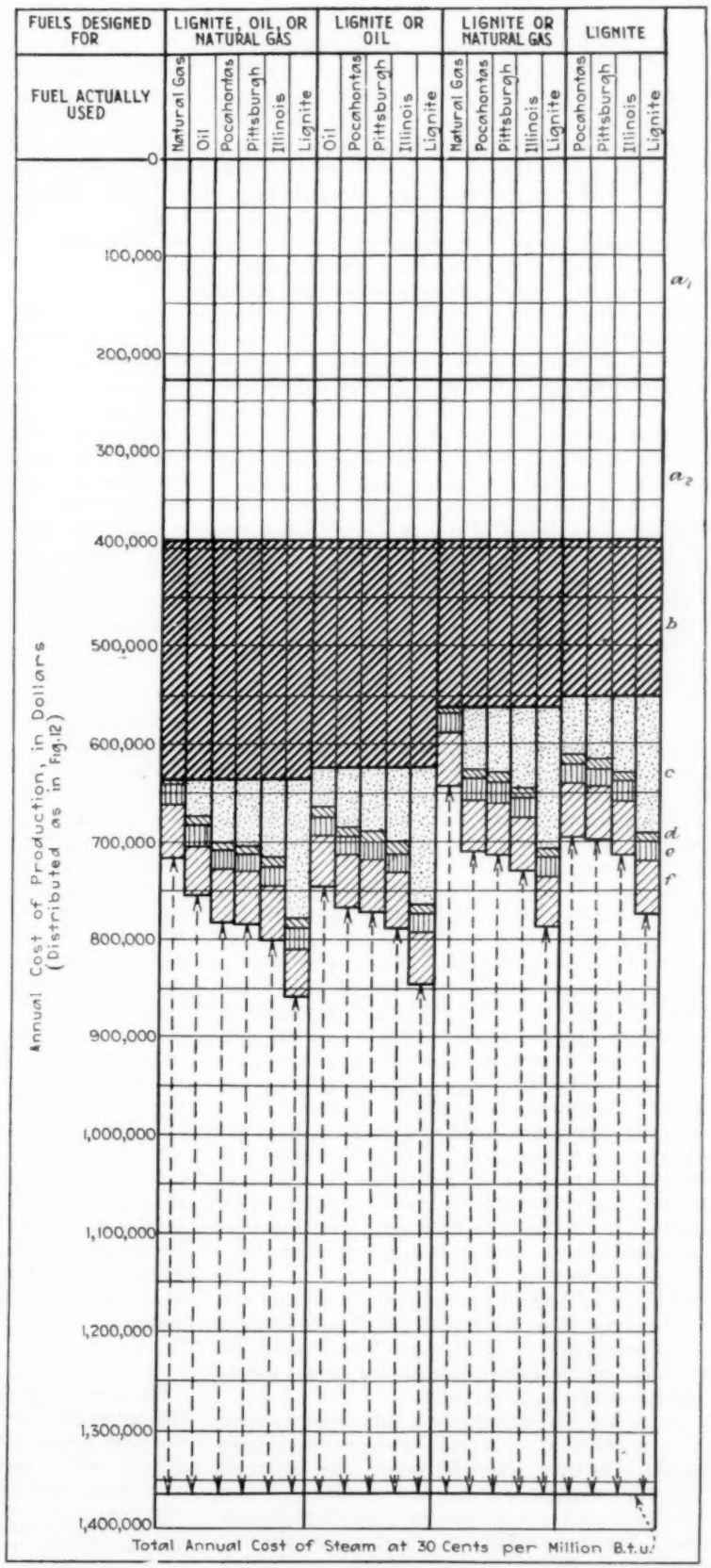


FIG. 15 EFFECT OF FUEL CHANGES ON PERMISSIBLE FUEL COST IN ORDER TO PRODUCE STEAM AT SAME COST

(a_1, fixed charges at 15% on boiler plant exclusive of fuel equipment [1 + 2 + 3½ + 4 + 5 + 6 +7 + 8] × 0.15; b, fixed charges at 15% on fuel equipment and fans [9 + 10] × 0.15; f, fixed charges at 15% on plant capacity reserved for boiler-plant peak; e, fixed charges on draft equipment; a_2, operating charges on boiler plant, not including draft and fuel equipment; c, operating charges on fuel equipment; d, operating charges on draft equipment.)

Table 2 makes it clear that a lignite plant in Texas cannot afford to burn oil. And as long as lignite may be purchased at $1.50 per ton, as is the case with several operating plants, even natural gas becomes attractive only at dump rates. A lignite plant in St. Paul, however, may not find its choice so easy with the cost of lignite from North Dakota $4 or more, laid down at the plant. Eastern coal becomes competitive under these conditions.

At present prices river anthracite would seem to merit consideration in competition with the best coals if a steady supply can be assured to the plant. Anthracite culm at prices comparable to those being obtained for river anthracite would make culm even more attractive than river anthracite. A plant suitable for culm which could be laid down at the plant at $2.50 or less for a considerable length of time could produce steam more cheaply than most plants in the East now do.

In general, it is quite safe to say that the installation of supplementary oil-burning equipment is seldom justified unless under most unusual market conditions. Even at present fuel oil prices it is difficult to do so.

In view of the many projects now under way for the more general distribution of natural gas, as for example the plans now maturing for making such gas available in the Middle West, especially in the St. Louis industrial area, it is of interest to consider the possibility of using natural gas in each of the plants considered in this paper. This is especially true because the additional capital expenditure required to adapt a pulverized coal or oil plant to the use of this fuel is small. At 20 cents per 1000 cu. ft., 1000 B.t.u. gas could compete with the usual $4.50 (delivered) Illinois coal or with $5.50 Pittsburgh coal.

Summary

The best economy attainable with various fuels at a given cost per fuel heat unit depends more on the fixed and operating costs peculiar to each fuel than on the efficiency of the plant with each fuel. In general, the greater the net available heat value of the products of combustion of the fuel as formed, the greater the economy, as shown by Fig. 14.

A plant designed for any solid fuel will generally be most economical when handling that fuel or any higher-grade fuel obtainable at the same price per heat unit which may be used without additional capital expenditures. Thus a lignite plant or a river anthracite plant can utilize any of the other solid fuels more economically than the fuels for which it was designed, if the others may be obtained at the same price per heat unit. In fact, some of them may be more economical at a higher price per heat unit. But as a rule the low-grade fuel is of local interest, and the absence of freight handicaps make it possible for it to compete with the higher-grade fuels. A plant designed for a high-grade solid fuel will generally not be able to develop its full capacity with a very much lower grade of fuel, and therefore cannot take advantage of favorable low-grade fuel prices. Thus a plant designed for Pittsburgh or Pocahontas coals could not develop its capacity without additional capital expenditures when burning river anthracite.

The solid fuels in pulverized form can generally compete with oil or even natural gas except in especially well-favored localities where gas or oil prices might be very low, or where the gas might be a waste product which has to be disposed of anyway, as for example blast-furnace gas in a steel plant.

Low-grade solid fuel, like lignite or river anthracite or anthracite culm, can be utilized in certain localities more economically than high-grade fuels handicapped by higher mine or freight charges. Oil-still residues, pitches, and petroleum cokes, although not considered in this analysis, have proved economical in many cases. The comparative economy of the various fuels considered may be approximated from Figs. 14 and 15 and Table 2.

Discussion

T. W. Harris, Jr.[2] Within the last few years, the number of pulverized-coal plants has materially increased and this undoubtedly has had an effect in increasing the demand for slack coals. This year's shipments up the lakes reflected this change in marketing conditions, and with the increase in demand for slack there is naturally an increase in price tending to bring the price of slack closer to the price of run of mine. This should be borne in mind in the drawing up of the original project covering the amortization of the plant and allowance made accordingly.

O. F. Campbell.[3] When operating cracking stills at high pressures, pulverized coal is not satisfactory due to the inability of the operators to have a clear view of the oil tubes. Hot tubes accompany coke deposits within the tubes and when hot tubes occur the still is shut down for cleaning. The operators must, therefore, be able to see the tubes clearly at all times so as to prevent tube failure and fires. When burning pulverized coal the fly ash prevents clear vision of tubes. Ash also sticks to the tubes, covering them over so as to prevent the determination of a hot tube, were such to occur.

In boiler practice, if a tube fails there is a tendency to put the fire out. In oil-still operation, when a tube fails a bad fire results, accompanied by considerable property damage and possible loss of life. Fuel for cracking stills must, therefore, be confined to liquid or gaseous fuels. Pulverized coal is used successfully on tube stills, where no cracking occurs and with comparatively low oil pressures. The use of coal, oil, or gas on this type of still is primarily one of economy, and is usually used where there is a large price differential between coal and oil or gas.

The burning of any kind of fuel in the furnace of a modern tube still presents a different problem than any discussed at this conference. The design of a modern tube still is such that the fuel must be burned in a furnace of comparatively low temperature. The furnace temperature of modern tube stills is approximately 1200 to 1500 deg. fahr. When burning pulverized coal under these low temperatures, consideration should be given to kind of coal, fineness of pulverization, burner design, use of preheated air, etc. Low volatile fuels are difficult to burn under the best of conditions on modern tube stills and the fixed carbon loss is high.

In an oil refinery it appears to be good economy to equip boilers to burn pulverized coal, pulverized coke, oil, or gas. Petroleum coke, in some cases, is a by-product and must be disposed of. Furnaces designed to burn pulverized petroleum coke can readily burn pulverized coal. Combination burners that will burn either pulverized fuel, oil, or gas appear to be the best installation. In furnaces where combination burners are not used, it is usually difficult to get proper air distribution when burning fuels in combination, or separately. Fuel-oil storage for the boiler house does not present much of a problem in an oil refinery as only small tanks are used at the boiler house into which oil is transferred from regular refinery fuel-oil storage tanks.

Author's Closure

The discussions emphasize the difficulty of making the conclusions of an economic study generally acceptable, and make it clear that specific points of view in evaluating the relative value of fuels lead to specific economic conclusions. Mr. Campbell, who speaks with the experience and point of view of the oil-

[2] Division Purchasing Agent, E. I. du Pont de Nemours and Co., Wilmington, Del.

[3] Sinclair Refining Co., East Chicago, Ind.

refinery engineer, points to factors which compel choices of fuels for certain types of refinery service so as to give but light emphasis to economic considerations.

Each of these "slants" on this wide subject helps to facilitate answers to questions as to why superficially equivalent fuels do not produce equivalent effects in a given plant.

The object of this paper is to show that even though the relative thermal value of various fuels with quite diverse properties may be readily predicted, the relative economy with which these may be utilized requires data on plant construction and operation costs not generally available. Such data were used to evaluate the relative rank of each of the fuels considered and which are representative of fuels commonly used for steam making in this country. Since the thermal rank assigned to the various fuels and the plant construction and operation costs used in this evaluation are representative of present American practice, the relative economy of the various fuels may be approximated from the charts, at least qualitatively, if not quantitatively.

FSP-52-12

Photo-Electric Smoke Recorder

By V. P. GRIFFIN[1] and J. V. BREISKY,[2] PITTSBURGH, PA.

The paper discusses the necessity of measuring smoke density in the stacks of power plants, so as to enable the operator to keep the smoke down to a minimum. Methods of smoke determination that have been available in the past are described. All of these methods depend on the human eye for judging the smoke density, and are influenced by weather conditions, being entirely inoperative at night. The necessity of a continuous and permanent record is brought out, a feature which has not been available in apparatus heretofore in use.

The photo-electric smoke recorder is then described. The scheme of operation is to project a beam of light through the stack or the breeching on a photo-electric cell. Depending on a lighter or heavier density of smoke in the stack, the photo-electric cell, popularly called the "electric eye," receives more or less light, which causes a greater or smaller current to flow through the cell. This current is then amplified so as to be sufficiently large to operate a standard indicating and a recording meter. These meters can be located at any desired point.

This device gives a continuous indication as well as a permanent record of smoke density, and also sounds an alarm or lights a lamp whenever the smoke density approaches a predetermined value. It works equally well day or night and is independent of weather conditions.

Operating results, including chart records of several installations, also are given, showing that smoke can be reduced to a minimum in a well-designed and properly operated plant.

MUCH has been said and written about smoke abatement, the evils of excessive smoke production, its deleterious effect on health and property, and the wastefulness of badly adjusted combustion.[3,4,5,6] Cities are passing ordinances and every one is interested in the problem. During recent years various efforts have been made to combat the causes of smoke. Combustion engineers have made noteworthy progress in doing away with improper combustion of fuel. Furnace designs have been improved so that greater effectiveness is secured from the fuel consumed. Experience has shown that even with the best designed boilers it is almost impossible to avoid the production of smoke under all conditions. It is, therefore, necessary to supervise the stacks in such a way as to get an immediate indication of the smoke being produced. Means for doing this have been rather crude and ineffective in the past, and those commonly used are unsatisfactory because they depend upon the fallible human eye. Such means of judging are inaccurate during the day as weather conditions vary, and altogether ineffective at night.

An eye more accurate than the human eye has been perfected by research. The photo-electric cell is an automatic device which is on duty 24 hours a day without suffering fatigue or variability such as affect the human eye. This cell has made possible the design of a reliable and accurate device which will both indicate and record smoke density.

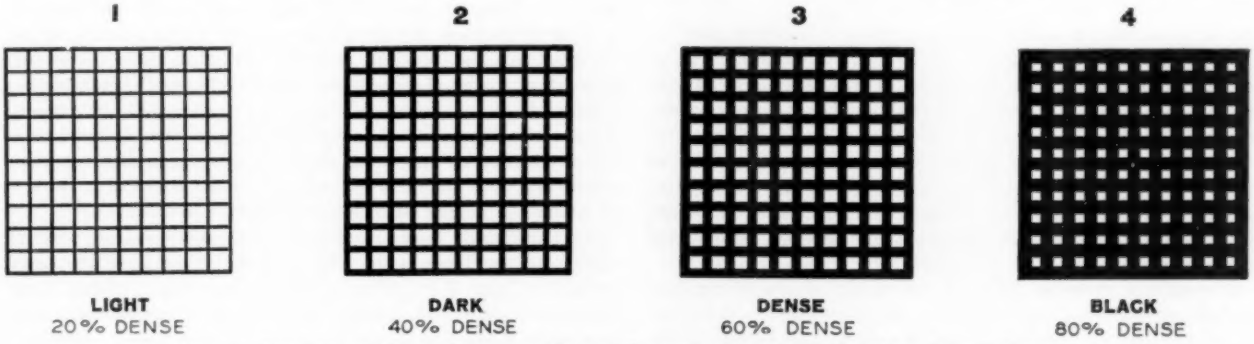


Fig. 1 Ringelmann Chart Reduced—In the Original the Charts Are Two Inches Square
(To use this a full-sized chart is set up directly in line between the observer and the stack at a distance of about fifty feet from the observer, so as to cause the lines to merge.)

Reasons for Measuring Smoke Density

Smoke is the term applied to the volatilized products of combustion of organic compounds when they are charged with finely divided particles of soot or carbon. The discoloration of the combustion gases as influenced by soot will vary with the amount present. Accordingly the color density will be an indication of the extent to which the combustion of the hydrocarbon constituent of the fuel has been completed. The slogan of former days, "where there is fire, there must be smoke," has been discarded. It is now an accepted fact that smoke is the result of poor combustion, and that its production can be attributed either to poor operation or to inherent defects in the design of the fuel-burning equipment or both.

No matter how excellent in design the equipment may be, good results can be obtained only if the equipment is correctly operated. This cannot be accomplished unless the operating personnel are supplied with the necessary instruments. In the case of boiler installations the steam-pressure and water-level gages are accepted as absolute necessities, but in addition, for economical reasons, many other instruments are required to guide the operating personnel properly. These consist of water-input or steam-output meters, temperature indicators and re-

[1] Mechanical Operating Engineer, Duquesne Light Co.

[2] Supply Engineering Department, Westinghouse Electric & Mfg. Co.

[3] "Problems and Methods in Smoke Abatement Work," by H. H. Kugel, Trans. A.S.M.E., FSP-50-75.

[4] "Damage Due to Smoke," by H. H. Meller, Trans. A.S.M.E., FSP-50-75.

[5] "Organizing a Smoke-Abatement Campaign," by Erle Ormsby, *Mechanical Engineering*, Nov., 1927.

[6] "Smoke Abatement Methods Used in Cleveland," by Elliott H. Whitlock, *Mechanical Engineering*, Oct., 1927.

Contributed by the Fuels Division and presented at the Third National Fuels Meeting, Philadephia, Pa., Oct. 7 to 10, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

corders, instruments to indicate the pressure throughout the air ducts, boiler setting, and flues, and some means of determining the composition of flue gases. The CO_2 meter or the air-flow steam-flow ratio meters are generally supplied to indicate the composition of the products of combustion.

There is, however, a considerable loss in furnace operation which is not directly indicated by any of the instruments mentioned, even though indirectly the air-flow steam-flow ratio or the CO_2 meters have been applied. This is the loss incident to the production of smoke. A predetermined standard CO_2 in the flue gases can be obtained with both a deficiency or an excess in air supplied. To prevent smoke there must be an excess of air, which excess, however, should be kept to a minimum so as to keep the percentage of CO_2 as high as possible. Smoke also may be produced in localized sections of the furnace, even though the CO_2 meter or the total steam-flow and air-flow ratio meter indications are correct.

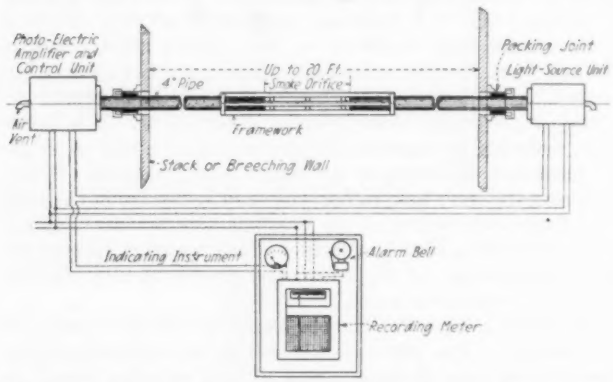


FIG. 2 SCHEMATIC VIEW OF PHOTO-ELECTRIC SMOKE RECORDER INSTALLATION

A condition entering into the operation of the steam-flow air-flow ratio meter which is not always recognized is that produced by irregular feeding of the water to the boiler, or by variations in feed-water temperature, commonly found where bleed heating is used. Instruments are now available which will proportion the rate of feeding to the steam output, and also compensate for differences in feed-water temperature. These are very beneficial, but they do not fulfil all conditions.

As a further check on furnace conditions, engineers have long recognized the desirability of determining the presence of flocculent, finely divided carbon particles in the products of combustion. Unless some reliable and convenient means of detecting the production of smoke is made available, the operator is left more or less at his own resources. Of course, ocular inspection of the furnace or stack by an experienced operator will show when smoke is produced. But such inspection, although effective, is not always practicable. There is, therefore, a demand for equipment which will give a reliable indication and record of the presence of amorphous carbon in the flue gases.

AVAILABLE METHODS OF SMOKE-DENSITY DETERMINATION

It may be well to review methods of smoke-density determination that have been available in the past.[7]

1 *The Ringelmann chart* (Fig. 1) is the method most commonly used. This system of classifying smoke densities into five grades, called No. 1, 2, 3, 4, and 5, has become quite generally adopted. Smoke ordinances in most cities are based on this method, most of them prohibiting the emission of No. 3 smoke for a period or periods aggregating more than six minutes in any one hour.

This method, of course, presupposes the ability of the human eye to judge colors correctly, and due to the fact that in practice the observations are nearly always made without the actual use of the charts, the possibilities of error become proportionately greater. Readings will vary depending on (1) whether the sky is clear or clouded, (2) the position of the sun in relation to the observer, (3) the direction in which the wind blows the smoke in relation to the observer, any satisfactory result being almost impossible of attainment if the smoke is being blown directly toward the observer, (4) on the distance of the observer from the stack, smoke appearing more dense from a distance than when close up. Inability to make any observations after dark is of course the biggest handicap in using this method.

2 *The Roberts smoke chart and the Donkin's smoke chart* are similar to the Ringelmann chart, in that they have the same gradation from 0 to 100 per cent dense smoke. Both these methods show smoke density by a definite greyness and therefore make it possible to use the charts close at hand. Observations are thus somewhat simpler than with the Ringelmann chart. In all other respects these two methods are comparable with the Ringelmann chart.

3 *The Umbrascope* consists of a cylinder containing four

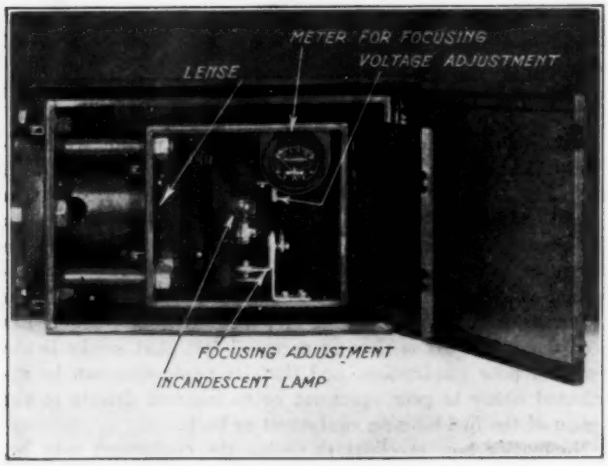


FIG. 3 LIGHT-SOURCE UNIT

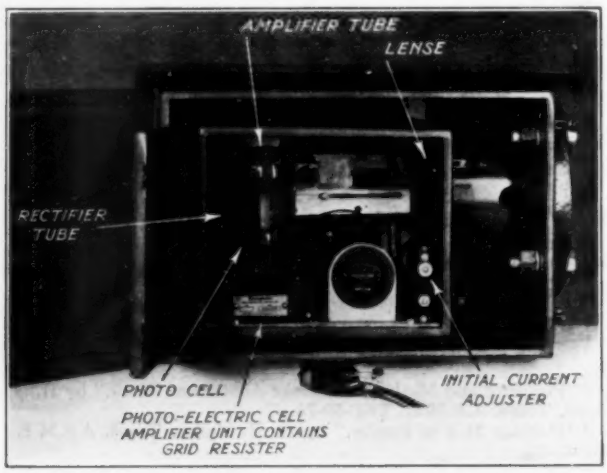


FIG. 4 PHOTO-ELECTRIC AMPLIFIER AND CONTROL UNIT

[7] Power Test Codes, A.S.M.E., Series 1923, Instruments and Apparatus, Part 20, "Smoke Determinations."

like thicknesses of glass hinged to swing in and out of the field of vision, each of them cutting off a certain percentage of light. The glasses are arranged to extend over only one-half of the field of vision, the other half of the cylinder being left open, through which the smoke may be seen. A direct comparison can therefore be made between the smoke and a standard. Thus it is easier to obtain comparative readings than by means of a chart.

To be able to use any of the above-mentioned methods, the man in the boiler room must have a clear vision of the stack from where he is working. This is usually not possible. Sometimes, however, use is made of a mirror mounted outside a window in such a position as to show the top of the stack to a person in the boiler room, so as to give at least an approximate indication of the amount of smoke emitted. This method is, of course, more subject to error than the direct use of the Ringelmann chart.

4 *Photographic Observation.* None of the methods described so far leaves a permanent record of the readings taken, except as these may be noted down by the observer. Thus it is impossible for the engineer in charge to check conditions regularly. Some plants have, therefore, adopted the scheme of automatically taking photographs of the stacks at predetermined intervals of time, so as to have a check on the fireman. This method is a step in the right direction toward automatic recording of smoke density. However, this scheme is also ineffective during the night, and the accuracy of the record obtained during the day depends greatly upon weather conditions. Besides, this record is of value only so far as a check on the operator is concerned, not being available instantaneously so as to be of use in correcting smoke conditions.

Necessity for Continuous and Permanent Record

An instrument is needed which will give an accurate continuous indication as well as a permanent record of smoke density. Such an instrument should operate during the day as well as during the night, and should be independent of weather conditions. Preferably it should also sound an alarm in the boiler room whenever the smoke density approaches a predetermined value. The indicating instrument should be located in the boiler room, while the recording instrument could be in the boiler room or in the chief engineer's office.

It is the purpose of this paper to describe the construction and operation of an instrument which was designed to fulfil the conditions outlined.

Construction and Operation of the Photo-Electric Smoke Recorder

General. The heart of this instrument is the photo-electric cell, a device which translates light impulses into electrical impulses, the amount of current flowing through the photo cell being directly proportional to the amount of light falling on it.

Talking movies, television, transmission of pictures by wire, and radio have the same photo-electric cell as their nucleus. Naturally a great deal of research work has been done in recent years on the development of this tube, and as now available it is constant in its characteristics, and has a long life.

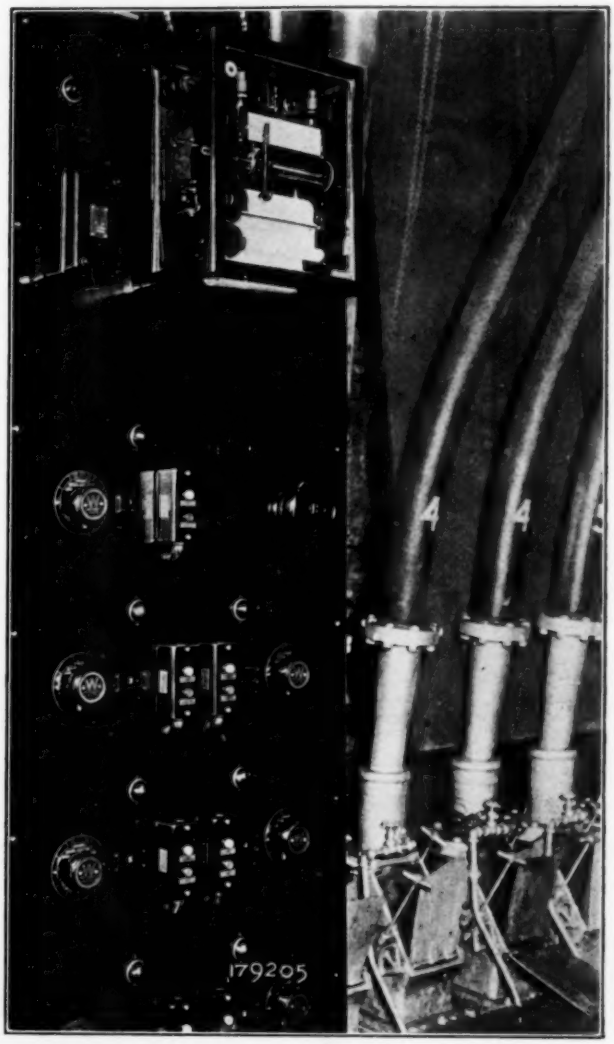


Fig. 7 Instrument Panel in Boiler Room

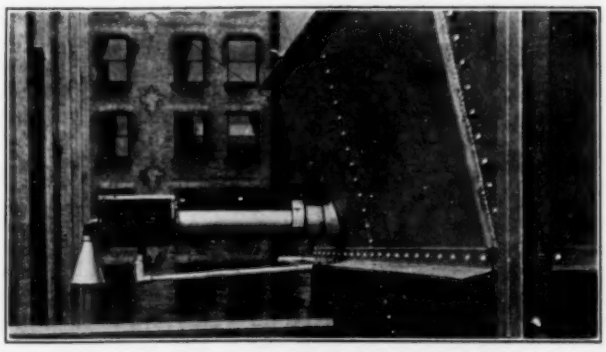


Fig. 5 Installation of Light-Source Unit

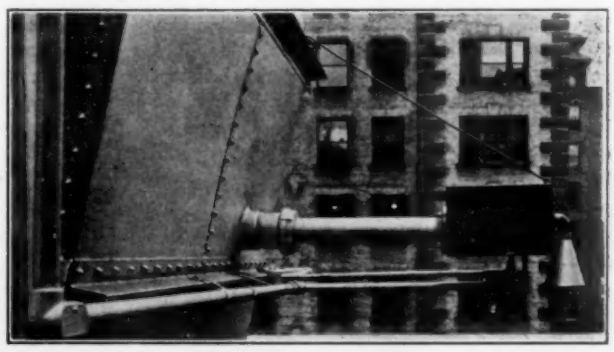


Fig. 6 Photo-Electric Amplifier and Control Unit Installed

Engineering developments have kept pace with the research work on this device, and industrial applications have been worked out.[8] The photo-electric smoke recorder is only one of the developments which make use of the photo-electric cell.

Early experiments made by Dr. Zworykin[9] showed that modern photo-electric cells were sufficiently sensitive to detect small amounts of smoke passing through a beam of light. Further laboratory tests made later on, as well as field tests, proved the practicability of designing a smoke indicator based on this principle. These experiments also caused efforts on behalf of other engineers toward the development of a smoke recorder.[10]

Description of Device. The photo-electric smoke indicator and recorder as developed by the Westinghouse Electric and Manufacturing Company[11] is shown schematically in Fig. 2.

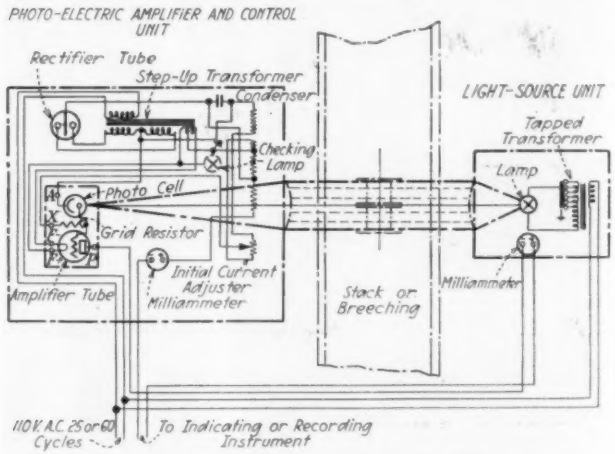


FIG. 8 WIRING DIAGRAM OF SMOKE RECORDER

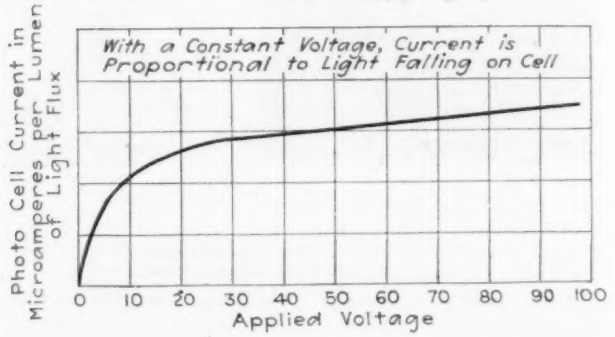


FIG. 9 CHARACTERISTIC CURVE OF PHOTO-ELECTRIC CELL

Briefly the scheme of operation is to project a beam of light through the stack or the breeching on a photo-electric cell. Depending on a smaller or greater density of smoke in the stack, the photo-electric cell receives more or less light, which causes a greater or smaller current to flow through the cell. This current is then amplified so as to be sufficiently large to operate standard instruments. The instruments are calibrated in degrees of smoke density as read on a Ringelmann chart.

[8] "Some Photo-Electric and Glow Discharge Devices and Their Application to Industry," by J. V. Breisky and E. O. Erickson; Abstract in *A.I.E.E. Journal* for April, 1929. For full article see A.I.E.E. reprint No. 28–146.

[9] *Electric Journal*, March, 1926, "The Photo-Electric Cell as a Smoke Detector," by S. H. Reynolds.

[10] "A Smoke Density Meter," by Frank Sawford, *Mechanical Engineering*, September, 1927.

[11] "A Photo-Electric Smoke Recorder," by E. H. Vedder, *Electric Journal*, May, 1929.

The device consists of two main parts: first, the light-source unit and the photo-electric amplifier and control unit, mounted on opposite sides of the stack; second, the indicating and recording instruments, as well as the bell alarm or indicating lamp, located in the boiler room or any other convenient location. Details of the stack units are shown in Figs. 3 and 4. Figs. 5, 6, and 7 show a complete smoke-recorder installation. The light-source unit consists essentially of a low-wattage incandescent lamp, a condensing lens for collecting the light rays into a parallel beam, which is projected through the stack, as well as adjustments for properly focusing this beam of light. The photo-electric amplifier and control unit contains the photo-electric cell on which, through a condensing lens, the beam of light sent out from the light-source unit is focused. It also contains the amplifier unit which requires direct current for its operation, and the necessary equipment for rectifying the alternating-current supply.

The double-box construction is used for the purpose of making the units weatherproof, so that they can be mounted either

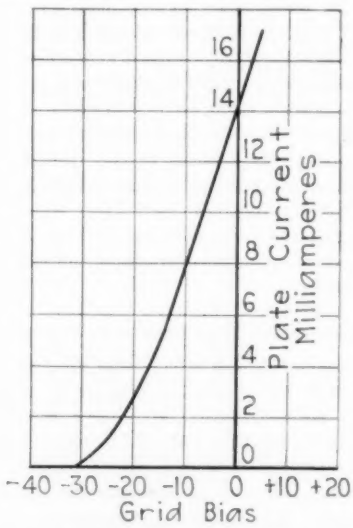


FIG. 10 CHARACTERISTIC CURVE OF THE AMPLIFIER TUBE

indoors or outdoors. Air is allowed to be drawn in at the vent at the rear of the unit and around the internal box, which is kept covered, and into the holes in the pipe connecting the internal and external boxes. The air drawn in flows through the pipe on which the units are mounted and into the stack. Thus the smoke has no chance of getting to the lens which is mounted on the front of the internal box. The flow of air through the units also prevents excessive heating of the apparatus due to the absorption of heat from the sun on particularly hot and bright days.

Installation. The construction of the mounting and the smoke orifice is shown in Fig. 2. The pipes on which the units are mounted are joined inside the stack by a framework (the smoke orifice), which forms a rigid connection between the pipes, thus assuring maintenance of alignment, but at the same time permitting the flow of smoke through the light beam which traverses the pipes and framework. The pipes pass through the walls of the smokestack in semi-flexible packing joints. This is necessary because it has been found that at high ratings, particularly in large stacks, the walls of the stack or breeching warp considerably due to the high flue-gas temperature. By using an orifice of a given length which allows a definite portion of the smoke to cross the beam of light, the total diameter of the stack has no influence on the calibration of the device.

All stack-mounting details (framework, packing joints, pipe flanges, etc.), with the exception of the 4-in. pipe, are supplied with the units. The user simply cuts two openings through the stack or breeching on opposite sides and supplies two pieces of standard 4-in. pipe cut to proper lengths depending on the width

A full-wave rectifier tube supplies direct current to the load resistor across which is a condenser to smooth out the ripple. The remainder of the apparatus is supplied with direct current from the rectifier. The characteristic of the photo-electric cell is shown in Fig. 9, and the vacuum-tube characteristic is shown

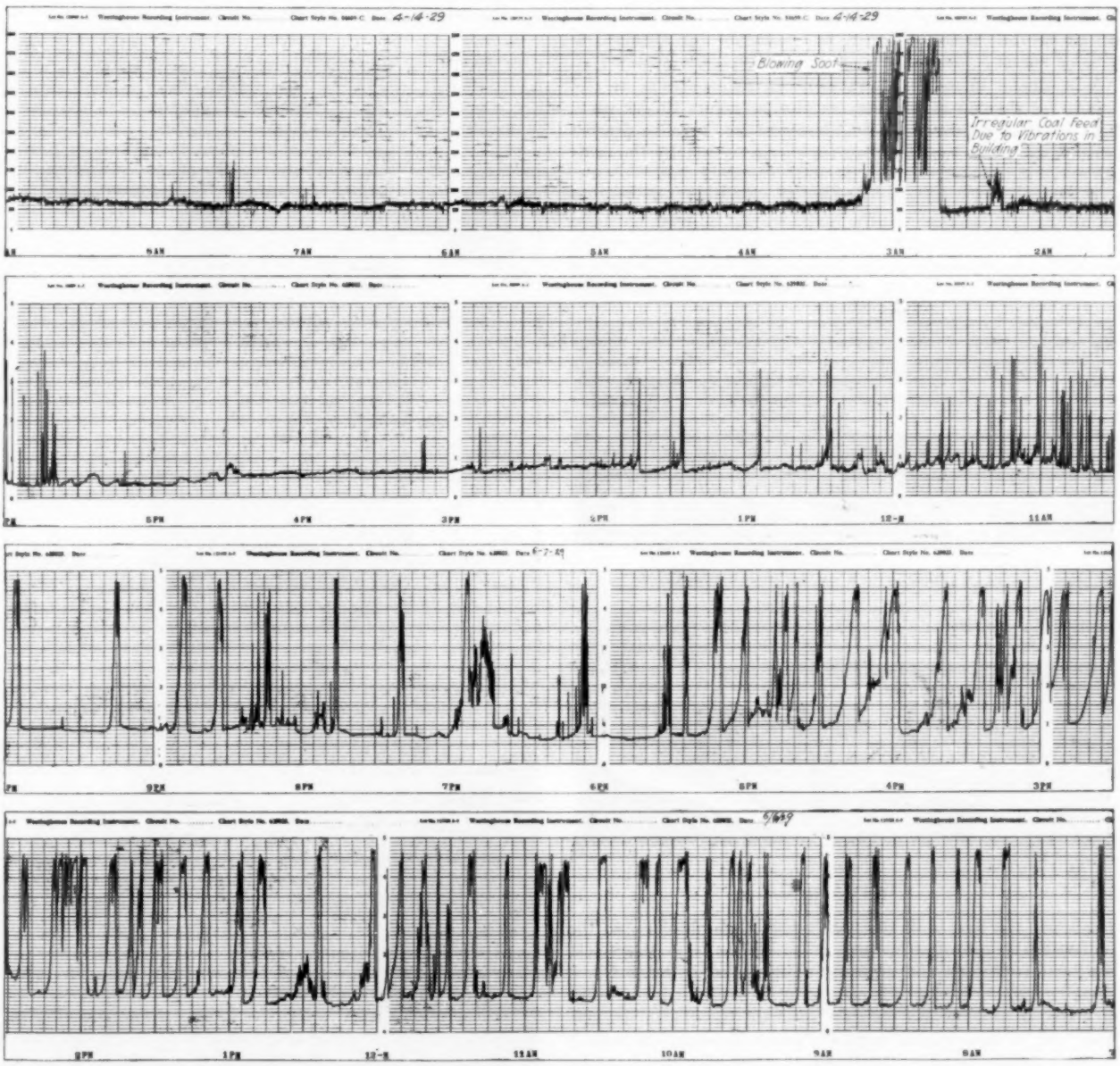


FIG. 11 SECTIONS OF SMOKE-RECORDER CHARTS OF SEVERAL INSTALLATIONS

of the stack, and then makes the assembly as shown in Fig. 2.

Electric Circuits. The electric circuit of the device is shown in Fig. 8. The units must be supplied with 110 volts, 25 to 60 cycles, no other source of electric energy being required. The circuit of the light-source unit is obvious. The electric characteristics of the photo-electric amplifier and control unit are described hereafter, in a rather sketchy manner. For a detailed description reference should be had to another article.[12]

[12] "Some Photo-Electric and Glow Discharge Devices and Their Application to Industry," by J. V. Breisky and E. O. Erickson, A.I.E.E. reprint No. 28-146.

in Fig. 10. It may be seen from the circuits in Fig. 8 that, when the photo-electric cell is dark and passes no current, the amplifier will have a negative bias on the grid, of a value determined by the setting of the initial current adjuster. If light is permitted to fall on the photo-electric cell, a current will flow from the cell to the load resistor, out of the load resistor at the initial-current adjuster, through the grid resistor, and back to the cell. This flow of current results in a potential over the grid resistor of a value determined by the quantity of light (in lumens) entering the cell, and also by the ohmic resistance of the grid resistor. If the unit is designed properly,

the amplifier tube will operate over the straight portion of its grid-voltage plate-current characteristic, and a straight line relation between smoke density as indicated and that read from a Ringelmann chart will result.

Compensation for Voltage Variation. The apparatus compensates automatically for effect of changes in line voltage. This has been accomplished in a very simple manner in the design of the circuit. Correct compensation for changes of plus or minus 5 per cent change in the voltage is obtained at No. 3 smoke density. Almost perfect compensation is obtained for changes of as much as plus or minus 10 per cent. The circuit is calibrated to compensate at No. 3 smoke density because this is the legal limit set for smoke density by the ordinances of most cities.

Calibration. The smoke recorder is calibrated by shutting off all the light, and moving the initial-current adjuster until the recording instrument reads No. 5 smoke density, corresponding to No. 5 smoke density on the Ringelmann chart. The light is then allowed to fall on the cell, and the lamp is focused with

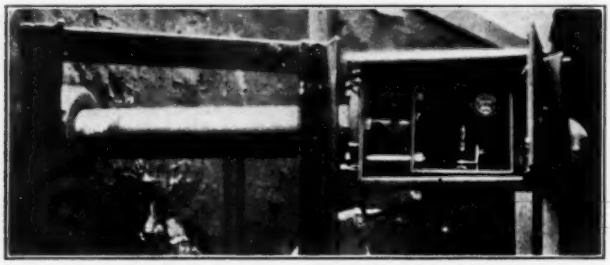


FIG. 12 INSTALLATIONS OF LIGHT-SOURCE UNIT

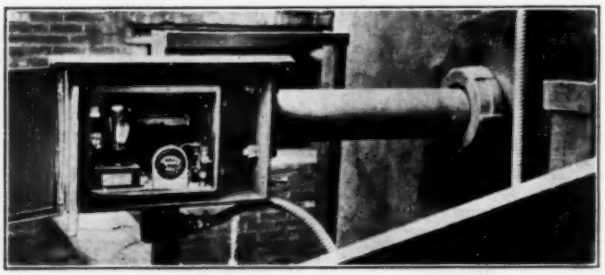


FIG. 13 PHOTO-ELECTRIC AMPLIFIER AND CONTROL UNIT INSTALLATION

respect to the lens until the proper reading is obtained on the indicating or recording meter. If no smoke is in the stack, the light is focused and adjusted until the meter reads No. 0 smoke density. If smoke is present in the stack, it is necessary to have an experienced observer estimate the smoke density. The light is then focused until the meter indicates the correct value of smoke density. The apparatus will then be in adjustment for all values of smoke density.

It should be mentioned here that all adjustments can be made at the light-source unit and the photo-electric amplifier and control unit without reading the recording meter, since each unit contains a small indicating instrument which is connected in series with the main instrument in the boiler room.

The graphic instrument (see Figs. 2 and 7) is a standard recording meter of the relay type, calibrated directly in degrees of smoke density. It is equipped with contacts which can be set too close at any desired value of smoke density. This makes it possible to ring a bell alarm or to light an indicating lamp the instant the smoke density approaches a prohibitive value. If the recording instrument is located in the boiler room, no indicating instrument is required. If, however, it is located in the engineer's office, an indicating meter must be mounted in the boiler room for the fireman.

Maintenance. After proper installation and calibration no difficulty is experienced in keeping the instrument calibrated. Changes in calibration, if any, will be caused by a change in the characteristics of the amplifier tube, photo-electric cell, lamp, dirt on lenses, etc. These parts, with the exception of the lamp, will have a life of a year or more. The lamp should be replaced once each six months. During this time its characteristics will be constant to such an extent that circuit adjustment

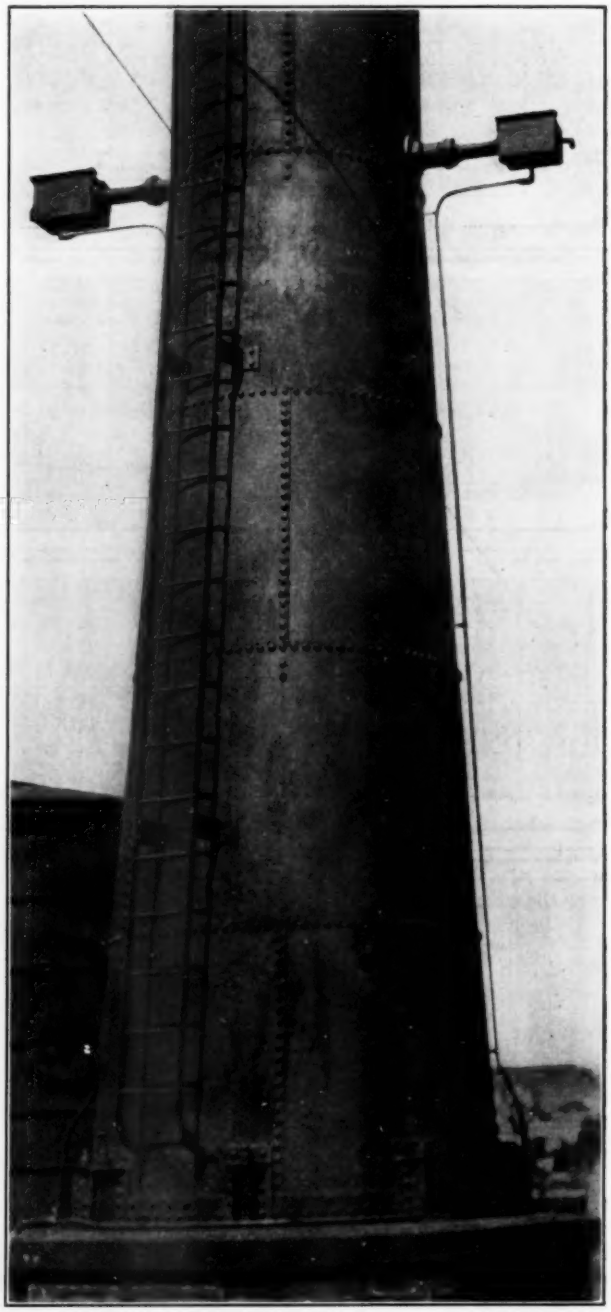


FIG. 14 SMOKE-RECORDER INSTALLATION IN SMALL PLANT

TABLE 1 CAUSES OF SMOKE AS SHOWN ON SECOND SMOKE CHART, FIG. 11

Time	Cause		*Time*	Cause
9:38	Hooked fire boiler No. 5.	Not shown on chart	11:30	Banked coal falling on fire.
9:42	Hooked fire boiler No. 5.		11:37	Banked coal falling on fire.
10:03	Hooked fire boiler No. 6.		11:40	Banked coal falling on fire.
10:22	Hooked fire boiler No. 5.		11:46	Banked coal falling on fire.
10:25	Cleaned fire No. 5.		11:50	Hooked fire No. 4.
10:28	Cleaned fire No. 6.		11:55	Hooked fire No. 5.
10:30	Green coal entered.		12:02	Banked coal falling on fire.
10:31	Speeding up stoker No. 6.		12:08	Banked coal falling on fire.
10:32	Speeding up stoker No. 6.		12:13	Forced draft fan put on.
10:33	Speeding up stoker No. 5.		12:22	Banked coal falling on fire.
10:35	Cleaned fire No. 4.		12:25	Speeded up No. 3 stoker.
10:40	Cleaned fire No. 3.		12:26	Speeded up No. 4 stoker.
10:42	Cleaned fire No. 3.		12:37	Banked coal falling.
10:43	Cleaned fire No. 2.		12:40	Banked coal falling.
10:45	Cleaned fire No. 2.		12:54	Hooked fire No. 2.
10:48	Green coal entered No. 3 and No. 4.		1:25	Hooked fire No. 2 and No. 6.
10:49	Green coal entered No. 2.		1:28	Hooked fire No. 3.
10:50, 10:52, 10:54	Green coal entering recently cleaned fires.		1:43	Hooked fire No. 2.
			1:45	Hooked fire No. 3.
			1:50	Hooked fire No. 6.
10:58	Hooked fire No. 4.		1:53	Stoker No. 6 speeded up.
11:00	Hooked fire No. 3.		2:15	Speed up No. 2 stoker.
11:01	Hooked fire No. 2.		2:20	Hooked fire No. 3.
11:08	Banked coal falling on fire.		2:25	Hooked fire No. 2.
11:11	Hooked fire No. 2 and No. 3.		2:35	Speeded up stoker No. 4.
11:12	Hooked fire No. 2 and No. 3.		2:47	Hooked fire No. 2.
11:16	Banked coal falling on fire.		3:10	Hooked fire No. 6. Load reduced at 2:30 p.m. accounting for no peaks from 3:10 to 4:25.
11:19	Banked coal falling on fire.			
11:21	Hooked fire No. 3.			
11:25	Hooked fire No. 5.			
11:28	Banked coal falling on fire.		4.27	Speeded up stoker No. 5

will have to be made only about once a month. Long life is obtained from the light source, amplifier tube, rectifier tube, and photo-electric cell by extremely conservative design of the circuit and apparatus, as well as by the use of specially designed tubes. All of these are operated considerably under their rating in order that rapid deterioration will not result. The lenses will need to be cleaned only about once each week or ten days, depending on the weather.

Operating Results

The first installation of this smoke recorder was made on a pulverized-fuel boiler. Previously to this, visional inspection of the stack by means of a mirror located outside the building had been resorted to at this plant. Shortly after the installation of the recorder, a marked improvement was evident as revealed by the chart records, in decreasing the density of the smoke produced, particularly at night, when ocular inspection of the stack was impossible.

Photographs of this installation are shown in Figs. 3 and 4. The light-source unit, photo-electric amplifier, and control unit are mounted to the breeching near the point where the latter joins the stack. The breeching is 18 ft. wide at this point. In the boiler room are located the graphic instrument, the indicating light, and the alarm relay, as shown in Fig. 7 (see top portion of panel).

A smoke density chart of the above installation is reproduced in the first chart of Fig. 11. The boiler control is very effective, and therefore there is very little increase in smoke density due to load changes. Most of the smoke shown on the chart is caused by blowing soot, and comes at relatively infrequent intervals. This chart shows how the emission of smoke can be kept to a minimum in a well-designed and properly operated plant.

The smoke-recorder installation shown in Figs. 12 and 13 was made in another plant, and is on a stack to which are connected six-600 hp. boilers equipped with underfeed stokers. The second section of the chart shown in Fig. 11 is for an average day's operation except that soot was not blown from the tubes during this period. However, the first chart in Fig. 11 shows this effect very clearly for another installation, and it would be the same here. Table 1 explains in detail what caused the various periods of high smoke density in the second chart of Fig. 11, and the cause for this smoke. Such a study should help in reducing the amount of smoke produced.

There is considerable difference between various boiler installations in the quantity and duration of smoke emitted. The first chart in Fig. 11 is taken from a plant burning pulverized fuel, and consists of a very jagged line, at low smoke density most of the time. The very jagged line is caused by slight puffs of smoke emitted by the flames in the various burners and is quite regular. The chart from the stoker-fired plant in the second section of Fig. 11 shows a more regular low limit, but shows greater tendency to smoke. The peaks as shown in the tabulation are nearly all caused by firing methods incident to stoker firing such as hooking and green coal falling down in chunks, cleaning fires, etc. Blowing soot is of course incidental to both methods of firing.

Fig. 14 shows a smoke-recorder installation in a smaller industrial power plant, using underfeed stokers. The stack is four feet in diameter. The third and fourth charts in Fig. 11 show that there are frequent peaks of long duration of dense smoke, which should call for improvements in operating methods or improvements in the boiler installation itself.

Conclusions

As mentioned previously, there is a direct loss of fuel concomitant with the coloring of the stack gases with finely divided carbon particles, the extent of the loss varying with the color density of the smoke. No specific value can be assigned to the losses encountered that will cover all conditions, but the production of smoke and loss are synonymous.

The elimination of smoke is not only conducive to economy, but also contributes to the intangible good-will of the surrounding territory. It can, and should be avoided by the use of properly designed equipment and by the adoption of satisfactory operating methods.

Discussion

P. Nicholls.[13] Measurements of the products emitted from a chimney which are based on appearance or on absorption of light are at best crude, and their numerical values are not dependable as a relative measure for different conditions of the same chimney, and still less as a true comparison of the undesirable emissions from different chimneys. To be really just, different chimneys should be compared on the basis of their area, and the length of smoke viewed, that is, the length of the slot in the pipe of the instrument, should be some fixed numerical value multiplied by the product of the area of the flue or chimney and the velocity of the gases.

An instrument such as described is, however, sufficiently satisfactory for present purposes, and the more important requirements are that it should be reliable without constant attention, and should be of reasonable price. A price that would put it within the reach of the smaller plants would make it a great boon. If there should come a day when inspectors could visit a plant periodically and inspect smoke charts in the same way as a fire-insurance inspector calls for the watchman's clock cards, then their life would be easier. It is to be hoped that this new instrument will be manufactured soon on a production basis, and simplified to reduce its cost rather than complicated for greater normal accuracy.

Everett H. Bickley.[14] While the writer believes the photo-electric smoke recorder to be the best method of recording smoke

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[14] Engineer, Bickley Manufacturing Co., Cynwyd, Pa.

at the present time, there are certain conditions under which the readings would not check with the Ringelmann density standards. White or yellow smoke, as generated by oil burners, ore smelters, or zinc roasters, would be recorded as black (as it should be), but the color would not check by visual observation.

A. S. Langsdorf.[15] This paper is of great interest since it makes use of an apparatus somewhat similar to a device that has been planned to yield measurements in a related field of smoke-abatement work. The particular instrument described in the paper is intended to produce a record of smoke issuing from an individual stack; however, in smoke-abatement work there is great need of a device which will give a more reliable indication than can now be obtained of the amount of solid material in suspension in the atmosphere over considerable areas. In measuring such suspended material, present practice has to be content with the weighing of deposits collected in containers of various types, or else with dust counts made with some such device as the Owens' dust counter. In both cases the results are subject to the errors common to all sampling, these errors being aggravated in this instance because of the fact that the samples are necessarily very minute in comparison with the total quantity to be determined. It was hoped that a new device, for the measurement of suspended material in a large area, might be ready for presentation at this meeting, but circumstances prevented the completion of the work. It may be in order at this time to report that the device makes use of a photo-electric cell upon which is projected a beam of light from a projector at a considerable distance, thereby making it possible to measure the amount of absorption of the light due to scattering caused by suspended material between the two stations. Many difficulties have been encountered, chief among which has been the elimination of the effect of daylight upon the photo-electric cell, but a way has been found to get around this difficulty. It is apparent that an observation which bears a quantitative relation to the condition of the atmosphere over a considerable distance, say a quarter of a mile, will be a measure of average conditions far more representative than that of a very small sample. It is expected that difficulties will be encountered in calibrating the device, since these readings will be dependent upon all kinds of floating particles, such as limestone dust, and not upon smoke and soot alone. In any case, the work thus briefly described will be presented for publication at a later date.

J. H. Walker.[16] This device will be welcomed by every enterprising power-plant operator. It fills a long-felt want.

[15] Dean, Washington University, St. Louis, Mo. Mem. A.S.M.E.

[16] Superintendent of Central Heating, The Detroit Edison Company, Detroit, Mich.

In Detroit at various times several devices have been experimented with for observing smoke. Periscopes and mirrors have been tried and an observer has been stationed to watch continuously the appearance of the stacks during certain periods when trouble was being experienced from excessive smoke. Now automatic cameras are trained on the stacks of central heating plants which take pictures at ten-minute intervals and record the date and time. All of these devices are more or less makeshifts. What is really needed is a continuous and instantaneous record right in front of the fireman.

Two important objections appear to the apparatus which has just been demonstrated. One is that the cost is very high. An instrument of this kind should be on each boiler if it is used at all. The purpose is not entirely accomplished if only one is installed for the entire plant, for it is necessary to know not only that there is smoke, but which boiler is the offender. If installed in this manner the total cost is a rather large figure.

The second objection that might be raised is that the instrument as now designed is hardly what one would call appropriate for a boiler room where there is exposure to coal dust. No doubt both of these objections will be remedied as the instrument becomes perfected and more of them are manufactured.

Author's Closure

The Ringelmann chart is based on the assumption that an area 60 per cent white for instance can be compared accurately with a sample of smoke in such a way that smoke which will transmit 60 per cent of the light will look the same to the eye as the mottled chart when held at a distance. The smoke recorder operates absolutely on quantity of light transmitted. The Ringelmann chart is in error, however, as soon as smoke is emitted which will reflect light, such as is the case with white smoke from various unusual fuels. The light reflected makes the smoke appear to be transmitting light even when such is not the case.

It is believed that considerable difficulty will be encountered in the development of apparatus to measure density of smoke, etc. over long distances due particularly to such things as fog, rain, etc. which it would not be desirable to measure.

The cost of the smoke recorder will decrease as the device is produced in quantities with tools, and in more or less continuous production.

The apparatus outside of the recording meter is very serviceable in any location, and is not affected by coal dust and water such as is obtained in ordinary weather outdoors. The recording meter is a standard instrument which is satisfactory in most cases for boiler-room installation. If exceptionally dirty conditions are encountered, a special case can be designed with the same meter parts which would be exceedingly rugged for severe conditions.

FSP-52-13

Stokers for Heating and Small Industrial Power-Plant Boilers

By JOSEPH HARRINGTON,[1] HARVEY, ILL.

Satisfactory small stokers for domestic or apartment-building heating are needed. The author describes the underfeed and overfeed principles of stoking and the design of the overfeed forced-blast progressive-combustion type.

IN APARTMENT buildings the boiler usually is devoted strictly to the purposes of heating. Occasionally there is a high-pressure boiler where part of the steam generated is used for cooking, laundry, or, in hospitals, for sterilization purposes. Schools, colleges, hospitals, and small hotels come into this category, the major portion being purely a heating proposition. To this large class may be added a certain limited number of small industrial plants of which the laundry is the principal exponent. Boilers used in this service are 250 hp. down to little fellows that do not exceed 10 or 15 hp. The cast-iron sectional boiler is one of the largest subdivisions of this class.

Up to the present time these have been practically all fired by hand, and the use of low-volatile or so-called smokeless coal, generally known as Pocahontas, has been the outstanding fuel, especially in the cities where smoke ordinances are enforced.

Residence heating can be divided into two broad divisions—first, that of heating the single residence building, in which the owner or occupant does the firing, and second, a class where a janitor is employed. Furnaces for the first class are usually small, one or two thousand feet of radiation being attached to them, and they are usually served by very low chimneys with very indifferent draft. These boilers are frequently round in cross-section, and some of them are hot-air furnaces. In any event they are handled by people who are in no way conversant with the principles of efficient combustion. The path must be smoothed so that the householder shall not be called upon for any technical or expert knowledge. The oil burner is filling this field with considerable satisfaction at the present time. It is probably the most difficult field for the automatic mechanical stoker.

The second class is the larger one wherein there is a janitor whose sole business is to take care of the fires. This janitor has a certain amount of expert knowledge; at least, it is his particular business to run the heating plant, even though he does it at long intervals and with indifferent care.

This class of buildings includes the multiple-resident apartments such as the hotels, hospitals, schools, and apartment buildings. Some of these jobs run into considerable size, and a mechanical stoker of large type is suitable, but there are tens of thousands of small apartment buildings in which there is need of a janitor only occasionally and wherein the firing attention is at considerable intervals.

There is very little more skilled attention in this case than in the case of the home, but at least there is one person specifically delegated to do the work. If he is supplied with reasonably efficient mechanical means which are pretty nearly fool-proof, he will get by with fairly satisfactory results. However, beyond doubt the equipment must be designed so as to withstand neglect and be capable of being operated in every possible improper manner without injury to the equipment.

The successful stoker for this service must be of such a type that it can scarcely be burned up or broken down by neglect. Coal of the wrong size will be used, ashpits will be neglected and allowed to fill up, drafts will be cut off, and all sorts of things will happen or fail to happen that ordinarily would put a stoker out of business. The nature of this service is such that the

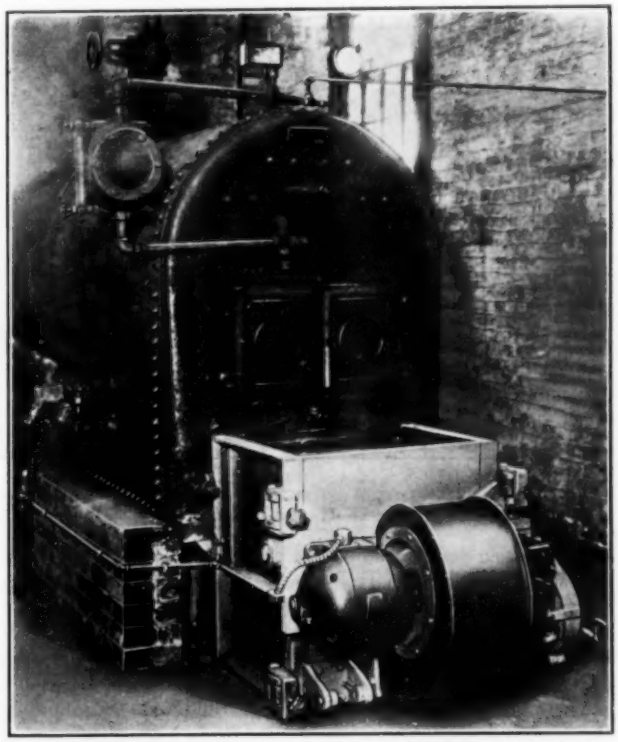


FIG. 1 INSTALLATION OF A WHITING STOKER UNDER A KEWANEE BOILER OF STANDARD HEIGHT

successful stoker in this field must be able to withstand all of these adverse circumstances. There must be some continuous removal of the ash in order to come within these specifications, inasmuch as the ash is the one element that is most likely to cause damage. It is comparatively simple to get the coal into the furnace, and comparatively simple to introduce the air, but the handling of the refuse involves the most difficult and complex part of the problem. If this is done automatically and continuously, the most prolific source of trouble is removed and there will result the necessary uniformity of conditions which are required for even control of temperature.

The average person comprehends but vaguely the tremendous magnitude of the residence-heating field. Of the half billion tons of coal consumed annually in the United States approximately one-third is consumed in the heating of buildings. Every building north of the Mason and Dixon line, with practically

[1] President, Joseph Harrington Company (subsidiary of Whiting Corporation).

Presented at the Third National Fuels Meeting of the A.S.M.E. at Philadelphia, Pa., October 7 to 10, 1929.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

no exception, must be heated during the greater portion of the year. The number of individual units serving this territory is almost beyond computation, and there is no branch of combustion engineering that so intimately affects the health and comfort of the people. It is a problem upon which an immense amount of thought has been expended, but up to recent months mostly in the design of the boiler itself, and upon the design of the radiation in the rooms to be heated. The purely combustion end of the business has been more or less neglected.

A firebox as commodious as could be obtained in connection with the boiler under consideration has been provided, and into this the best coal obtainable is shoveled at infrequent intervals. Provision for a little overfire air is made by placing a small grid in the firedoor, but beyond this practically nothing has been done to insure smokelessness or high efficiency in the heating furnace.

Some splendid work has been done by members of this association in improving the design of the hot-air furnace, but very

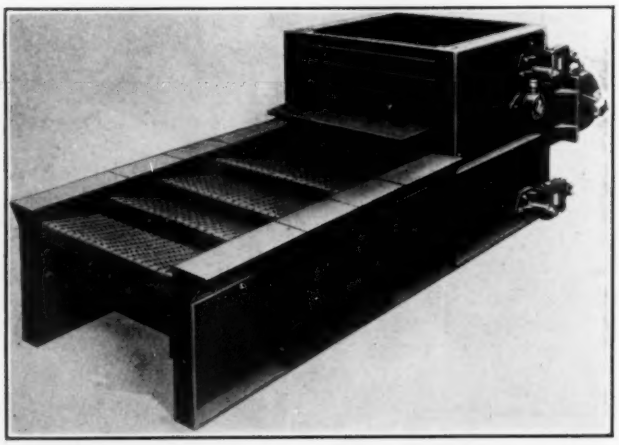


FIG. 2 REAR VIEW OF WHITING STOKER SHOWING UPWARDLY INCLINED GRATE SECTIONS

little has been done in the way of providing automatic mechanical stoking until within a comparatively short period, which may be counted in months rather than in years. In other words, up to three or four years ago, stokers for the smaller heating jobs were practically unknown, but today they are coming into use with remarkable rapidity.

There is little doubt that the conditions surrounding the usual shovel-fired domestic heating furnace are uneconomical in the extreme. Nothing much worse could be designed than the method of burning coal which is comparatively general in this service. Efficiencies down to 20 or 30 per cent are common enough in boilers of this type, whereas there is no fundamental reason why the heating boiler should not be operated at efficiencies comparable to the better class of power-plant boilers. Just because the unit is small, is no real reason why it should not be efficient. There is undoubtedly room for improvement in the boilers themselves, but there are many boilers on the market today that in themselves will take up heat in a very efficient manner, so that if this heat could be economically provided the unit would be quite an efficient one. However, the system of shovel firing does not provide the essentials of smokeless or economical combustion, and it remains for the mechanical stoker to provide these.

The oil burner is a fuel feeder that might readily be classed as a "stoker." It introduces the fuel mechanically and uniformly, and thereby provides for efficient combustion to that extent. When accompanied by proper air mixture and temperature, excellent combustion conditions are secured and fine efficiencies are obtained.

During the war the author worked on a special oil-fired boiler. Uptake temperatures, so low that the burned hydrogen condensed and ran out as water, proved that it was almost too good as a heat absorber, and it was necessary to allow it to soot up some, so as to keep the stack gases above 212 deg. fahr. A careful laboratory test gave an overall efficiency of something like 86 per cent.

This boiler is not on the market, although the author has one in the basement of his home, where it has given good service for about 12 years. It simply goes to prove that the residence heater can be just as efficient as the big power-plant boiler. It is also just as good evidence that some form of mechanical fuel feeding is at the root of such efficiency.

One-half of the fuel used in many installations can be saved by the use of efficient stokers. What this would mean to the American household can readily be calculated. In addition to the actual coal saving there must be the accompanying conditions of labor saving, cleanliness, convenience, and fully automatic operation. The day of the coal stoker embodying all of these is at hand.

The oil-burner people were the first to enter this field, beginning with the strictly single-family residence buildings, and they have, by the expenditure of large sums of money, frequent advertising, and persistency, developed the field to a point where now it is unnecessary to sell the prospective customer the idea of mechanical stoking, either for liquid or solid fuel. The large stoker in the power-plant field has reached a stage where it is a thoroughly accepted piece of equipment, and it is no longer necessary to argue with the customer the basic proposition of whether he should have stokers. It is only a question of what kind of stokers he should have.

In the new field of the small heating plant it is necessary frequently to convince the prospect that there is something better than Pocahontas coal and the shovel. There is also a human factor which one does not always meet with in the larger plants, i.e., the janitor. It is the custom for each man to have a string of buildings within walking distance, sometimes as many as 10 or 12, which he visits with a certain regularity, firing on each return trip.

It is necessary under these conditions to have a firing period long enough for the janitor to make his rounds. This frequently is of two or three hours' duration, so that when he arrives at a plant in his circuit, he shakes the fire, cleans out a few clinkers, and then puts on anywhere from 15 to 25 scoops of coal. This is supposed to last until his return trip, on which occasion it is most probable that the fire has burned pretty well down and sadly needs replenishing.

This intermittent extreme method of stoking a furnace is just about as inefficient as could possibly be devised, and without doubt the mechanical stoker has a wonderful field in which to work if only from the standpoint of uniform, continuous firing, smokeless combustion, and uniform heat evolution. Also here is a double waste, the very inefficient firing method and the expensive fuel.

The oil-burner people have certainly done a good job in preparing the public mind for some form of automatic equipment in the basement. There still remains, however, the argument that oil fuel is expensive and at times dangerous, and the field is still open for a successful automatic mechanical stoker handling raw coal or coke, in which form heat can be purchased at the lowest possible price. If this low-priced unlimited fuel source can be combined with a cleanliness that is equal to oil, and all automatically operated so as to respond readily to thermostatic control, and the absence of this disagreeable ash handling that

accompanies shovel firing, a prospect that is extremely interesting is opened up.

The very first attempt at mechanical stoking of fuel was made quite a number of years ago. It consisted of a hand-operated plunger forcing coal up through a circular duct to a vertical riser in the center of the grate. This forced the coal up into a mound-like formation, and it then burned on the grates as usual. This was first applied to hot-air furnaces, and it preferably called for anthracite buckwheat or similar non-coking coal. Today there are may be a dozen small stokers built that employ the underfeed principle.

The restricted areas available in the domestic heater are not sufficient for much of anything in the way of automatic ash removal with this type of stoker. Side dump plates such as are employed in the type E stoker are not practical, so that these stokers consist of merely a bedplate covering the entire area of the furnace with the exception of a central retort into which the fuel is screwed or pushed, and through which it flows upward into the characteristic mound of the underfeed stoker. As it passes the tuyere line it reaches the zone of ignition, ignites, and the fixed carbon burns in a layer on top of the green fuel. As the fixed-carbon combustion terminates, the ash is left at the surface of the heap, where it generally rolls down the sides on the bedplate opposite the clean-out doors. If the temperature is below the melting point of the ash, it will be loose and soft, but if the temperature exceeds the melting point, the ash melts and runs into rather generous clinkers which must be removed with the firing tools through the door. This operation is sometimes difficult and usually results in considerable disturbance to the fuel bed, production of dust in the boiler room, and possible smoke.

From the foregoing it appears highly desirable to have a coal with a high fusing point or else operated at the rate of combustion that insures a low fuel-bed temperature. Selection of the coal to be used on this stoker therefore becomes of considerable importance, and manufacturers are now classifying the coals of their district with a view to selecting those which have the least clinker-forming tendencies. This is usually a direct function of the fusing temperatures of the ash. Not all coals present the desirable characteristic of high melting temperature, and it therefore becomes a source of considerable concern to the manufacturer and also to the users of this equipment to find a coal which will operate satisfactorily.

These stokers are governed by automatic control which is always of the "stop-and-go" type. In other words, a sensitive spring or a mercury tube is connected to the boiler in such a manner as to be affected by the steam pressure. As the steam pressure goes up this closes a circuit, which, acting through a relay, shuts down the motor on the stoker and conversely starts it up again after the pressure has dropped a few ounces or a few pounds. Reasonably satisfactory pressure control can be obtained in this way. These fans are usually substantial, and it is immaterial mechanically whether they start and stop often or not. If some care is taken in adjusting the damper in the stoker and the amount of fuel feed, it can be made to develop a long "on" period, but if the load is such that the "off" period lasts for an hour or two, it is inevitable that the fire should burn down to a very small amount, so that when the stoker again starts up the fan will simply blow cold air into the furnace, and still further reduce the pressure.

Under these conditions the recovery of steam pressure is very slow as things must wait until the stoker has again filled the furnace with enough coal to cover the tuyeres. When this occurs the fire will build up and temperature or pressure can be recovered. However, this is sometimes a matter of a considerable period, possibly half an hour or so, before a good fire is again built up after a long period of idleness.

This suggests a method of control which is known as the floating control in which the controller floats with the steam, moving slightly up or down as the steam pressure varies, and correspondingly affecting the fuel feed and air supply. For high-pressure units these floating-control devices are readily enough obtainable, but for low-pressure or vapor systems they do not seem to exist. The "stop-and-go" method is therefore apparently the only one which is applicable to such installations.

An interesting contrast to the underfeed principle is afforded by a fuel feeder which feeds coal through a pipe or chute into the furnace above the fuel bed, allowing it to drop freely on the top of the pile. At the same time air is blown in directly on the surface of the fuel bed. No attempt is made to remove ash or shake the grate. It is limited in size, and is intended to serve the countless single family residence, which it is reported as doing successfully.

Outside of the boiler, there is a hopper holding a considerable quantity of coal. A small motor operates both the fan and the

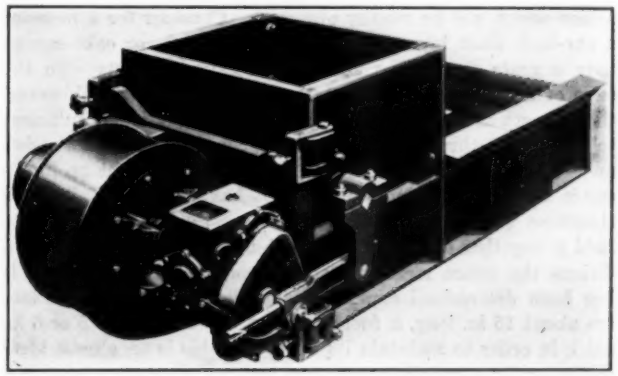


FIG. 3 FRONT VIEW OF WHITING STOKER SHOWING DRIVE AND FAN ASSEMBLY ON FRONT DOOR

means for getting the coal into the furnace. It is applicable to any of the small, round hot-water or air furnaces.

In contrast to these principles of stoking is the overfeed progressive-combustion type. The one that the author is interested in at present consists of a series of four-grate steps or sections, the first and the third section reciprocating adjustably from nothing to 4 in., and the other two steps are stationary. Each step is made up of a plurality of ventilated grate bars arranged side by side so that the air space can be adjusted from minimum to any required amount by simply wider spacing of the bars. A hopper is placed over the front of the stoker ahead of the front wall of the boiler. In front of this there is a panel or door carrying on it the blowing equipment, motor, and speed-reducing device.

There are certain novel principles of design incorporated in this stoker. It is a front-overfeed stepped grate without gravity action and without inverting the fuel bed. The fuel enters at the front under a vertically adjustable gate just the same as a chain grate. The reciprocation of the first grate section which forms the bottom of the coal hopper advances the green fuel at a thickness determined by the height of the gate. On the inward stroke of the feed plate the feed is carried in a matter of 3 or 4 in., whereas on the return or back stroke, the fuel is held in place by the vertical portion of the hopper apron and the weight of the coal in the hopper, and the plate slides under the mass of fuel, thereby advancing the fuel relative to the feed plate a certain amount. This alternate action continues until the fuel is advanced to the end of the first row of grate bars which are attached to the feed plate. At this point it falls off

over the end of the grate bars on the second or stationary step. Then immediately following the forward stroke of the plate, this coal which has fallen over on the second step is pushed along in compression, and the return stroke then causes more coal to fall over the nose of the grate into the space provided by the previous forward action. This coal is in turn then pushed along and eventually the entire grate surface is covered.

There are two very distinct actions in this motion; first, the entire fuel bed breaks down over the end of the grate section, thus breaking up any caking tendency there may be due to the cracking or breaking down of the fuel bed. Each stroke then embodies a compressive thrust causing a compacting or compression of the fuel bed to a point where all of the granulated fragments of coke are pushed together into a compact mass. This mass, of course, is not permitted to consolidate into continuous coke, because shortly afterward it is again caused to fall over the end of the succeeding step, which breaks it up and again subjects it to alternate compression and release.

There is a curious, interesting, and valuable result of this action which will be readily observed. Consider for a moment a one-inch thick layer of coal, or sand, or ash, or coke spread over a grate step. The advancing section contacts with the edge of this thin layer and starts to push it along. However, the strength in compression of the granular layer is not sufficient to transmit a thrust sufficient to overcome the frictional resistance on the grate. It therefore piles up or thickens up in advance of the advancing step until it has gradually built up a thickness wherein the weight of the layer will be sufficient to hold it together and resist further buckling. Under these conditions the entire mass will slide along the grate section. It has been determined experimentally that when these sections are about 15 in. long, a fuel bed must of necessity be 5 or 6 in. thick in order to maintain its form, and this is an almost ideal thickness for a traveling-grate fuel bed. Uniformity of fuel bed constitutes one of the essentials required of good combustion, and in this stoker the uniformity of the fuel bed is particularly marked.

Imagine a thin spot occurring in the fuel bed due to either physical disturbance or burning out of a porous section more rapidly than the surrounding area. Just as soon as this spot thins down to a point where it becomes structurally weak it will no longer sustain the thrust of the moving fuel bed, and it therefore buckles up and becomes as thick as the surrounding area. Under these conditions it can transmit the thrust and move along just the same as its neighbor particles. It is obvious from this that there cannot be a thin spot or hole in the fire, and it is due to a combination of this characteristic plus the granular porous character of the fuel bed that is responsible for the rather unusually high CO_2 that is habitually obtained.

The uniform rate of combustion indicates also a uniform termination. In other words, the coal finishes its combustion, and the ash is finally deposited in a very straight line at the extreme rear end of the stoker. At this point another one of the author's favorite principles of stoker design is brought into play, viz., the continuous and automatic discharge of the ash.

Just as the chain grate is characterized by a perfectly uniform condition of fuel bed, so is the Whiting stoker. The constant and uniform discharge of the ash as formed means, of course, that the fuel bed is always in just the same condition, and efficiency once obtained can be maintained continuously.

The mechanical features of the stoker can best be seen from the accompanying illustrations. Only the entire operating and blowing mechanism is carried in one assembly on the front door of the stoker. This is the only machine-shop part. This assemblage is carefully put together and is shipped out as a unit; it being necessary only to put the hinge pins in at the two hinges on either side of the door, and it is ready to close up and start to work.

Heating boilers which are most commonly encountered are cast-iron sectional boilers or all-steel types. In both cases they are of the water-leg type in which a water-cooled member approaches the floor to within 18 or 20 in. The grate surface of the Whiting stoker is cut down to a maximum height of 16 in. so that the entire structure can be slid under the front water leg of one of these boilers without any disturbance of the foundation or the boiler setting. In order to meet variations in these boilers, the stoker is made sectional both as to width and length in variations of a few inches. It is possible, therefore, to fit a stoker to any boiler furnace, filling it out and supplying practically as much grate area as it originally had with shaking grates. The stoker being essentially a forced-draft stoker, it is possible to secure a considerably higher rate of combustion on a slightly reduced grate surface and thereby develop much higher ratings from the boilers.

Even with the restricted furnace volume that is encountered in some of these boilers, the stoker is completely smokeless by virtue of the fact that the fuel bed is unusually uniform and the resultant penetration of oxygen very evenly divided. The flame is short, sharp, and completely smokeless. This stoker is adapted to thermostatic pressure control, and will carry steam pressures or water temperatures within a close range. The fan is perfectly silent and a hopper of several hours' capacity can be supplied.

The problem of heating residences involves a number of conditions which the steam engineer never encounters in the boiler room of the power plant. The peculiar janitor situation already referred to calls for perfectly automatic operation, and it is essential that control equipment be provided that will stop and start the stoker in accordance with steam requirements. This question of stopping and starting is all right provided the intervals are spaced so that the fire is not ruined during the off periods. The Whiting stoker is equipped with standard stop-and-go controls just like all of the other small stokers. It seems to be the most practical form of controlling equipment for low-pressure systems, and there is no reason why it should not be satisfactory provided there is within the stoker necessary controls manually adjusted so that the rate of burning or rate of steam generation can be controlled carefully. If the hand controls of the stoker are set so that the recovery is slow and the "off" periods short, then there is practically continuous operation. This is quite possible with most of the small stokers in which there is control of both the amount of air supplied and the amount of coal fed. This is in sharp contrast with certain types of oil-burning equipment used for the same service. In these burners the air is not adjusted below a certain point because of practical difficulties. The oil is cut down mechanically to a point corresponding with the steam requirements, but it is obvious that when the oil is cut down beyond the corresponding air admission there is an excess of air, and the CO_2 will drop rapidly. Under these conditions efficiency is lost, and while the steam generation is controlled, it is controlled in an expensive manner.

There is a difference, however, between the stoker firing coal and the burner using oil as a fuel. Whenever the oil burner goes off entirely, the consumption of oil absolutely stops, and the oil must be re-ignited upon resuming operation. With the coal stoker, however, there is a certain weight of incandescent fuel left on the grate surface, and this continues to produce heat and steam for quite a length of time. It is very noticeable, however, that with the forced-draft stoker the steam generation drops very rapidly the moment the blower is cut out, so that there is a very satisfactory control because of this effect. There

is not, however, the strain on the boiler and brickwork due to absolutely shutting off the heat, and upon resuming operation there is a warm furnace already there for the early stages of combustion. With the oil burner the furnace may get perfectly cold between fires, and it is quite a handicap to oil, to fire it into a cold chamber and expect it to burn without deposit of soot or the production of smoke.

With the Whiting stoker there is a double form of the so-called "stop and go." First there is the complete stoppage of everything caused by the mercoid cutting out the motor and allowing it to come to a full stop. Then there is the duplex system wherein the mercoid merely cuts in or out a certain amount of resistance, and slows the motor down a predetermined amount. This speed will produce steam at a rate somewhat lower than that required, whereas when it is going on full it produces steam at a greater rate. The operation of the stoker is therefore a short period of overproduction and a short period of underproduction, the average being suited to the requirements of the job. This, of course, means that at no time is the stoker standing still and the fuel bed without air. It is always in good condition

FSP-52-14

Coke as a Domestic Heating Fuel[1]

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This paper deals with one of several types of fuel used in heating houses. It sets forth the burning characteristics of coke when used for this purpose as shown by work done by the U. S. Bureau of Mines and others. Of the solid fuels, anthracite and coke are truly smokeless and are held in high regard for this quality. Ways are being found for making commercial coke from a wider range of coals than heretofore, and the use of coke for domestic heating is increasing.

UNTIL recently, the term "coke" has had a definite and limited meaning and has referred to "high-temperature" coke, more specifically defined as "beehive," "gas-house," and "by-product" coke; all three types having the common property of a low-volatile-matter content of about 1 to 2 per cent. Of recent years, "low-temperature" cokes have been coming on the market and have been widely discussed. These are produced at lower coking temperatures, and a smaller part of the volatile matter is driven off. Another kind of coke comes as a residue from the refining of oil, termed "petroleum coke" which is available to a limited extent in some districts, its outstanding property being its low ash content. The term "coke" unqualified as used in this paper refers to high-temperature coke, any other type being specifically defined.

Table 1 shows typical compositions of three types of coke. The per cent of ash in a low-temperature coke will be less than that in a high-temperature coke made from the same coal because less of the combustible is removed.

TABLE 1 TYPICAL COMPOSITIONS OF MOISTURE-FREE COKES

	High-temperature	Low-temperature	Petroleum
Fixed carbon, per cent	88.5	81.0	83.5
Volatile matter, per cent	1.5	10.0	9.6
Ash, per cent	10.0	9.0	1.9
Calorific value, B.t.u.	12,900	13,500	15,600

The use of coke as a domestic fuel has increased rapidly in recent years; in 1928 it was 2¼ times what it was in 1924. Table 2 shows the figures as reported by the Bureau of Mines[4] (3, 4) for the years 1923 to 1928.

TABLE 2 COKE SOLD FOR DOMESTIC USE IN THE UNITED STATES, MILLIONS OF TONS

	1924	1925	1926	1927	1928
Beehive coke	0.14	0.39	0.37	0.11	0.07
By-product coke	2.8	4.1	5.0	4.7	6.25
Gas-house coke	about 1.3 million tons per year				
Petroleum coke	0.235 million tons estimated for 1928				
Pitch coke	No data available.				

These quantities compare with 60 to 70 million tons of domestic sizes of anthracite produced per year, and 56 to 77 million tons of bituminous coal which the bureau estimates as used for domestic purposes; cokes, therefore, at present form about 6 per cent of the total of solid fuels used for domestic purposes.

Producers and distributors of fuel are offering greater service. This includes (1) advertising and sales effort; (2) inspecting the furnace of a prospective customer to see that it is suitable for burning the particular fuel and is in good repair; (3) instructing a purchaser as to firing and damper adjustments; and (4) following up sales to ensure that the users obtain satisfactory results. Improvements in quality of coke have been mainly in grading the coke by size, supplying the size well suited to the particular furnace, and in avoiding excessive dust and moisture. The chemical and burning characteristics, as affected by the coals used and the process of coking, have also been studied in the effort to produce a fuel that will be best suited to average conditions.

Quality of Coke

The reputation of coke as a domestic fuel was formerly not good. The viewpoint of householders was expressed by saying that they could not be bothered with it; that it required too much attention; that it either all burned away or the fire went out, although there was plenty of unburned coke in the furnace; and that in cold weather such bad clinkers were formed that the fire had to be rekindled after removing them. It is true that the burning characteristics of coke as compared with other fuels may have permitted such conditions, but the causes are preventable. In many districts coke was a by-product and little effort was made by the producers to supply it in the right size or to give the service needed. This was particularly the condition when it was distributed through dealers who had not the proper facilities for storing and handling it, nor sufficient interest in promoting its sale.

The qualities of a domestic fuel cannot be described in detail without associating them with the size of the pieces. This applies to coke (6, 7, 8) and to all free-burning fuels, but to a less extent to those coals which cake and coke because the pieces fuse together and tend to form a single solid mass. Coke has, however, some general characteristics which are well recognized. These are listed briefly in the following paragraphs. However, it should be recognized that there are differences even between high-temperature cokes, and that their qualities have a range of variation even though it be small.

Favorable Characteristics of Coke

1 Coke is a relatively clean fuel, although if too dry it may be dusty, and such dust is disagreeable because of its gritty nature.

2 Coke is a truly smokeless fuel and is referred to as the ideal solid fuel from that standpoint; as a consequence, the user has no trouble from soot accumulations nor from reduction of the available draft.

3 Coke gives a uniform-burning fuel bed which requires no attention between times of firing.

4 Coke requires little draft and the fire responds quickly when the dampers are opened.

5 Coke gives a high thermal efficiency. The overall efficiency based on the calorific value, obtainable with coke burned in a given boiler, is at least equal to that obtained with anthracite, as fuel-bed burning characteristics are very similar. Anthracite having 3 to 5 per cent of hydrogen against 1 to 2 for coke may have a larger loss in the latent heat of the steam carried in the flue gases, while the loss of combustible in the ashpit may be larger with the fuel having the larger percentage of ash, since small pieces of unburned fuel get smothered in the ashes. The heat obtainable per pound of each fuel will depend on the quantity of ash-free combustible, and will be proportional to the calorific

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[2] Fuels Engineer, Pittsburgh Experiment Station.

[3] Assistant Chemical Engineer, Pittsburgh Experiment Station.

[4] Numbers refer to bibliography at end of this paper.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

value, and it may be taken as approximately proportional to 100 minus the sum of the ash and moisture contents.

Numerous tests of coke burned in domestic furnaces have been reported; a few references are given in the bibliography (10, 11, 12). The efficiencies obtained have depended on the furnace; values of 60 to 65 per cent for warm-air furnaces and 68 to 72 for boilers have been reported. The average efficiency that a householder would obtain during a heating season would be lower than these test values; tests with coke under less careful attendance such as reported (13) by the Bureau of Mines for coals would be useful.

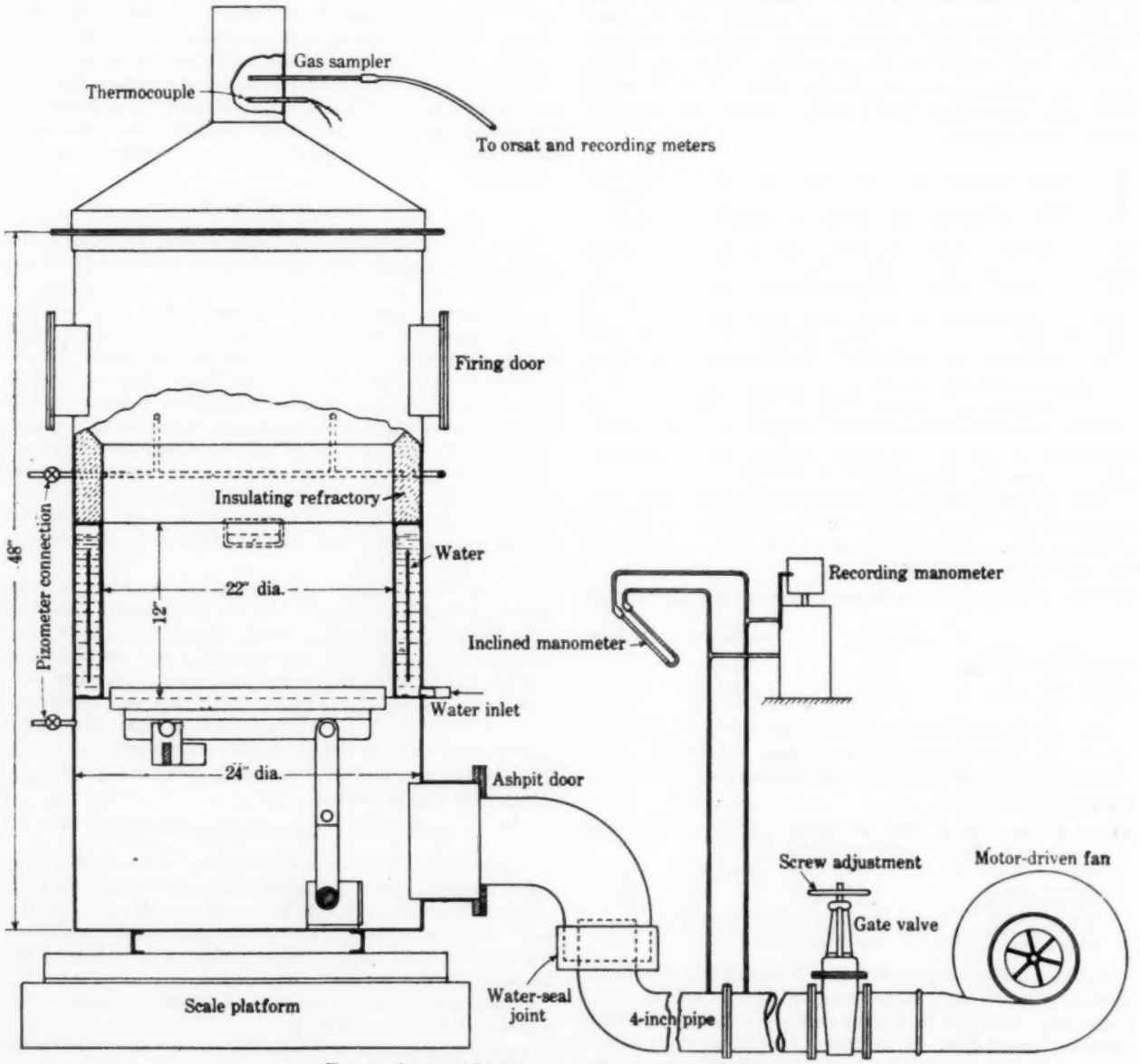


FIG. 1 SPECIAL FURNACE AND SET-UP USED IN STUDIES

COMPLAINTS REGARDING COKE

Besides the favorable qualities mentioned there may be placed complaints which vary according to the skill of the fuel user. These complaints are listed with brief comments as follows:

1 "Coke requires more frequent firings than do anthracite and bituminous coal." This, of course, is true when the comparison is based on the weights of equal volumes of the fuels. The weights per cubic foot compared on the basis of sized fuels are: Coke 20 to 28 lb., with an average of all coke as sold of probably 26 lb.; bituminous coals, 45 to 50 lb.; anthracite 50 to 56 lb. A mixture of sizes of any one of the fuels will give larger weights per cubic foot. On the basis of firing equal volumes of the fuels, the time a firing would last with equal rates of heat production would be proportional to pounds per cubic foot, times calorific value per pound, times average thermal efficiency. The relative time that equal volumes of the fuels would last can be taken in round numbers for the sake of illustration as, coke 1, bituminous coal $1^1/_4$, anthracite 2. Such a basis of comparison is not the whole story; the possibility and advisability of firing a given volume of fuel are fixed by the size of the fuel, the combustion space it leaves above the fuel in the furnace, and the draft available. Bituminous coal requires more combustion space over the fuel bed, and heavy firings in ordinary furnaces will produce excessive smoke and loss of heat value from unburned volatile

matter. There should therefore be more frequent firings for this reason. The combustion spaces required for anthracite and coke for equally good results do not differ much with equally suitable sizes of fuel, but would slightly favor the coke. The comparison between these two fuels depends more on the average requirements of householders and whether a furnace will hold sufficient coke to last for the period desired, which is usually such that the housewife will not have to add fuel during the daytime. Coke advocates have claimed that the average practice has shown that coke is fired $1^1/_2$ times as often as anthracite, but this does not disprove the 2 times possibility that anthracite has.

2 "It is difficult to start a coke fire." This again is a matter of comparison. Fuels vary considerably in their "ease of ignition" or "ignitability." A pile of wood can be ignited by starting one small portion of it, and a bituminous fire can be nursed to complete ignition if a relatively small portion of it gets a start. On the other hand, coke and anthracite must be ignited over a larger portion of the bed if they are to continue burning; the lack of flame from volatile matter must be replaced by higher surrounding temperatures. The absence of volatile matter is not, however, the only factor, as the carbon in the various types of cokes has varying ability to combine with oxygen at a given temperature. The ease of ignition depends also on the size of the coke pieces, and the quantity of wood required to start the fire increases with increase in size of coke.

3 "Coke is difficult to bank so that it will either not go out during the night or not be all burned up before morning." The ease with which a fire is maintained overnight depends on the weight of fuel that can be put in the fire pot and on the probability of guessing the correct adjustments of the dampers. Coke suffers by comparison because there is less weight in a given fire pot, and because the range of movement of the dampers required to produce rapid burning on the one hand and to extinguish the fire on the other, is often smaller than that of the fuel with which it is compared, thus increasing the probability of not setting them correctly. It is necessary that the ashpit be tight and that all the entering air be under the control of dampers. Successful banking is then a question of experience with each furnace. A fire is more likely to go out when using large coke than with small coke.

4 "In mild weather it is difficult to adjust the dampers so that the desired heating of the house is obtained, and unless the furnace is given frequent attention either the house gets too hot or the fire goes out." This complaint is more commonly heard from those who have used bituminous fuels and have been accustomed to operate the furnace in mild weather by maintaining a small bed of live coals in the center of the grate with ashes around the sides. This method cannot be used with coke, and the recommendation is always made to fill the fire pot and regulate the rate of burning by the dampers. It is not necessarily true that there is economy in having a small bed of fire as the burning is then very inefficient. However, with any fuel there is difficulty in keeping a fire all day when only a small quantity of heat is required in morning and evening.

5 "Coke makes worse clinkers than do coals, and one is likely to have to use more kindling after removing the clinkers." The latter difficulty is overcome by removing the clinker before the fuel bed gets too low. The process of coking a coal does not alter the clinkering characteristics of the ash. Tests made by the Bureau of Mines showed that with the same rate of burning the same quantities of clinker were formed by the coal and by the coke made from it. However, a user is more liable to form clinker when burning coke because he can make it burn faster than he can burn the coal, and thus get a hotter fuel bed. If this forcing of the fuel bed overheats the house, he may shut down all the dampers and stop the flow of air through the fuel bed; this allows its heat to travel down to the ash and thus increase the clinker formation. Table 3 gives the results of tests made by the bureau and shows the increase in clinkering caused by including a one-hour banking period in burning tests, each of which lasted about five hours.

TABLE 3 EFFECT OF BANKING FIRE ON CLINKER FORMATION

Fuel	Ash-softening temp., deg. fahr.	Rate of burning lb. per sq. ft. per hr.	Per cent clinker over 2 in. Not banked	Banked
Coke	2610	10	6.8	12.7
Coke	2610	20	9.3	19.0

It is to be expected that the increase in clinkering would have been greater if the ash of the coke had been more fusible.

It is desirable that coke for domestic use should be made from coals having a softening temperature of the ash of not less than 2400 deg. fahr. Coke with a lower fusion temperature can be used without trouble in furnaces which are sufficiently large to heat the house easily, but if they are small and have to be forced to the limit, troubles from clinker may result.

6 There is sometimes a belief that burning and destruction of the furnace grates accompany the use of coke. Such trouble may occur with any fuel if, to obtain a rapid rate of burning, the ashes are all shaken out so as to leave the hot fuel in contact with the grates. Especially is this true if the fire is banked when the fuel bed is at a high temperature. The damage will be more severe if molten slag drops on the bare grate when hot, because it will combine with the iron. Coke does not differ from other fuels in this respect, except that one can obtain more readily a higher rate of combustion and a hotter fire than with other fuels.

7 Complaints have been made that when coke is used in a warm-air furnace the fumes permeate the house, whereas this trouble was not experienced when burning bituminous coal. As far as this occurs it is caused by incorrect operation of the furnace, and the closing of the stack damper so far that a positive pressure is produced in the fire pot, gases being forced through the firedoor and into the warm-air ducts if the joints in the furnace are not tight. When bituminous coal is burned, the person adjusting the dampers easily recognizes when he has closed them too far because smoke will issue from the firedoor, whereas with coke, the issuing gases are invisible.

Instructions for Burning Coke

There are a few simple instructions which show how the causes for the complaints can be minimized and the good qualities emphasized. Such instructions have been given wide publicity by the Bureau or Mines (5) and others, and by the coke producers and distributors through advertisements and pamphlets. All of the instructions have been similar and are briefly as follows:

1 Before starting to use coke, the furnace should be inspected to see that it is in good repair and that the dampers will close, that the ashpit is tight and has no leaks which will admit air when its damper is closed.

2 The flues and smoke-pipe should be cleaned.

3 The size of coke recommended for the heater should be used. There is no standard scale for coke sizes and producers differ in their methods of grading, both on size and on the number of sizes carried; consequently, recommendations have varied. The effects of size are discussed later.

4 A deep fuel bed should be carried and as large a charge fired as the furnace will hold. A shallow bed should not be carried, but the fire pot filled to its capacity, even in mild weather.

5 The fire should be leveled but not poked and the ashes removed by shaking the grates, leaving a layer of ash to protect the grates and to increase the resistance of the fuel bed.

6 Dampers should be opened the minimum distance necessary to obtain the desired heat.

Such instructions can only be very general, and they leave a

lot for the user to learn from experience if he is to get the most satisfactory results. His problem with coke is, however, no worse than with other fuels, except that the adjustment of dampers may have to be finer. Success with coke does, however, call for more self-restraint than with other fuels; it is easier to obtain a fierce fire, and trouble is less likely to result if one exercises patience and gives the house time to warm up without making use of the full possibilities of the extra hot fire obtainable with coke.

Investigation on the Burning of Coke in Domestic Furnaces

The foregoing part of this paper has consisted of statements of generalities concerning the characteristics of coke as a domestic fuel. A great amount of investigative work (1, 2) has been carried out during the past eight years on the manufacture, properties, and use of cokes, both of the high- and low-temperature types. Most of this work and the causes for its being done are foreign to the purpose of this paper; it is pertinent, however, to show how far the general characteristics previously set forth

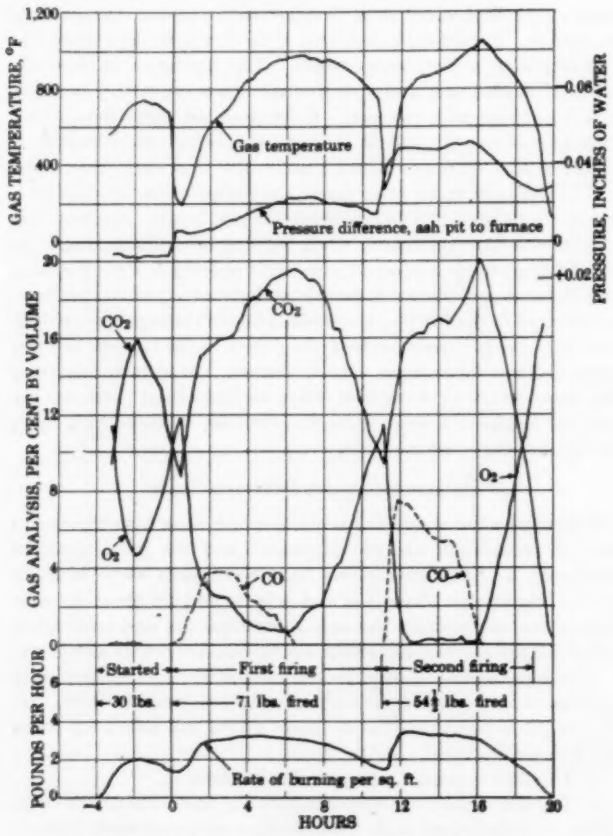


Fig. 2 Graphs of Data From a Test of Coke

can be confirmed by experimental data. The data to be given are extracted in the main from work done at the Pittsburgh Experiment Station of the U. S. Bureau of Mines; unless references are quoted the results have not previously been published. Much work has also been done in England, but it is not applicable to usual American conditions because in England domestic coke is burned mainly in open grates.

The actions involved in the operation of a heating furnace can be considered in four stages: (1) The burning of the fuel with the primary air from the ashpit, and the transfer of some heat from the fuel bed through its direct contact with the sides of the fire pot and by radiation from the top of the bed; (2) the burning of the combustible gas with the secondary air supplied through the door slots; (3) the transference of the sensible heat of the hot gases to the water or air; and (4) the exhausting of the gaseous products by the chimney action. Although the rate at which the fuel burns is dependent on the chimney action, yet the characteristics, that is, the actions in the fuel bed and the results that follow, at each rate of burning in a given furnace are dependent only on the fuel, and a study of what occurs in the fire pot should give quantities which can be compared.

Studies of the fuel-bed actions have been made by the Bureau of Mines; the tests have been conducted under strictly controllable conditions which simulated those of household use. It is not necessary for the purposes of this paper to describe the apparatus and methods of test in detail or to give the results in full. Fig. 1 shows the set-up of a special furnace having a 20-in. diameter fire pot; a similar set-up was used with a furnace having one of 28 in. Both fire pots were water-cooled so that the conditions surrounding the fuel bed corresponded to those of a steam or hot-water boiler. The furnace was mounted on scales and arranged so that accurate measures could be obtained of the weight of fuel gasified during any short period. The primary air supplied through the ashpit was measured and could be controlled closely, the air pressure in the ashpit also could be maintained at any desired amount. It was thus possible to subject each fuel tested to similar conditions, and to eliminate the variations of draft, control, and manipulation associated with tests of fuels in heating boilers which hamper close comparisons between results unless a large number of tests are made. The methods of conducting the tests were rigidly alike. The quantity of fuel fired affects results. The basis adopted for cokes was to fill the fire pot to the same depth, $16^1/_2$ in., at every firing, which corresponds to what householders are instructed to do.

Fig. 2 is a typical plotting of some of the data obtained during a complete test. It shows a log of the rate of burning, the composition and temperature of the gases above the fuel bed, and the draft above the fuel bed. It will be seen that there were three stages to the test, starting the fire with a small amount of fuel followed by two full firings.

It is rather difficult to summarize briefly the results of such tests. Each period between times of firing is a cycle in which the fuel bed is changing from moment to moment. The results will therefore be shown as average of a complete period between firing because it is the average the householder is interested in. The results will also be shown as curves without indicating the test points because this paper is only concerned with tendencies.

Effect of Size of Coke Pieces

Statements have previously been made that the size of the coke pieces materially affects the burning characteristics of coke and that for each furnace a certain range of sizes will, on the average, be found to give the best satisfaction. It is of importance to have at least a general idea why that is so, how the results obtained may depend on the size, and how the size may affect the way the furnace should be operated to obtain the best results.

The investigations on the effect of size were made for several types of high-temperature coke, but because such cokes do not differ materially the results for one only are shown. They were also made for at least three rates of burning, but because the householder would need the same rate of heat supply regardless of the size he was using, the comparisons will first be made on the basis of the same rate of burning. A rate of $3^1/_2$ lb. per hour per square foot of grate area has been chosen because it is about the average rate during cold weather. The term "rate" as used in this paper means the pounds of combustible burned, thus eliminating the effect of variations in the quantity of ash in the coke.

Four sizes were tested, 1/2 to 1 in., 1 to 1 1/2 in., 1 1/2 to 2 in., and 2 to 3 in. All were sized by square-mesh screens.

Size of Coke and Draft Required

By the draft required for a furnace is usually implied that at the smoke outlet, which includes the draft required to move the gases through the passes of the heater. However, by measuring the draft over the fuel bed the effect of the construction of the heater is eliminated and the comparisons are confined to the fuel bed. Fig. 3 shows this draft plotted against the mean screen size; both the average for the period between firings and the maximum values are shown. The curves show that the draft required decreases rapidly with increase in size of coke pieces.

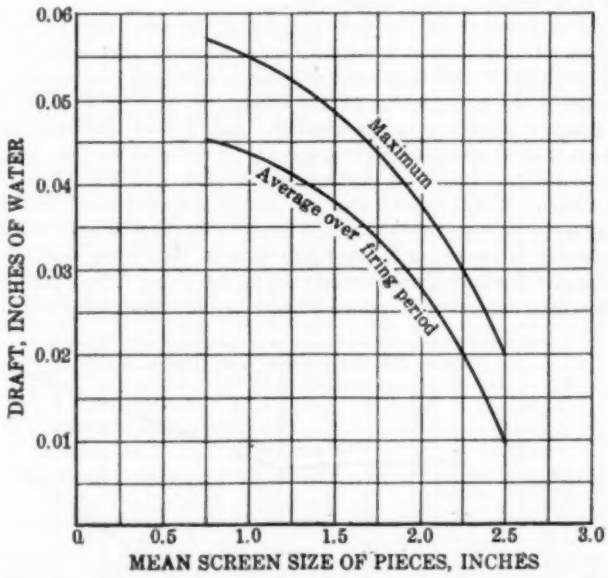


Fig. 3 Draft Required Over Fuel Bed as Related to Size of Coke for High-Temperature Coke Burning at an Average Rate of 3 1/2 Lb. per Square Foot per Hour

It is interesting to note that the draft required decreases with increase in size of coke in spite of the fact, as will be seen from Fig. 6, that the quantity of air passing through the fuel bed to obtain the same rate of combustion is greater for the larger than the smaller size. Thus the draft required for a 2 1/2-in. coke is only about one-fifth of that for a 3/4-in. coke, whereas the quantity of air through the fuel bed for the 2 1/2-in. size is more than twice that required for the 3/4-in. coke.

Fig. 4 shows the relation of the maximum draft required above the fuel bed to the rate of burning for 1- to 1 1/2-in. coke. The draft increases with the rate, but not as rapidly as one at first might expect. The primary air required to gasify a pound of fuel decreases as the rate increases, while the temperature head increases; both of these causes reduce the draft above the fuel bed.

Size of Coke and Secondary Air Required

The composition of the gases arising from the fuel bed of a domestic furnace, even when burning a uniformly sized fuel, is not the same over its entire area because of the cooling effect of the sides of the furnace which retards combustion. As a result, at the top of the bed free oxygen may be found around the sides, whereas the gases in the central portion may contain no oxygen and considerable carbon monoxide (CO); if there is good mixing of the gases after they leave the fuel bed the free oxygen may combine with the CO. The amount of air which must be supplied above the fuel bed through the firedoor slots (secondary air) will be that required in addition to that passing up the sides. It will evidently be more for a warm-air furnace in which the sides of the fire pot are much hotter than the water-cooled sides of a steam or hot-water boiler.

This occasion is taken to point out the misapplication or misuse which has often been made of the data and statements contained in Bureau of Mines Technical Papers No. 137 (1916) and 139 (1918) on the combustion in the fuel bed of hand-fired furnaces. These publications deal with furnaces in which the fuel bed is large, and its thickness kept the same by relatively short intervals between firings. These publications clearly state that some of the secondary air which the experimental tests showed to be necessary would be supplied through holes in the fuel bed, around the sides of the bed, through cracks in the wall and around the doors, and not all would have to be supplied through the door slots. Also, it was stated that all the experimental data were obtained with one size of fuel, 1 to 1 1/2 in. In spite of this, isolated statements have often been quoted, without the qualifying contexts, and applied to domestic furnaces to support some device being sold to improve their combustion efficiency.

It will be seen by an inspection of Fig. 2 that the amount of CO above the fuel bed varies during the period between firings, starting from zero, reaching a maximum, and then decreasing to zero again. Comparisons of the composition of the gases above the fuel bed as it is affected by the size of the coke can be made by taking the average of the composition over the whole period. Fig. 5 shows such averages for the carbon dioxide (CO_2) and the CO. These values are only of interest because the composition of the gases determines the secondary air required. It will be seen that the quantity of CO is high for small coke and low for large coke.

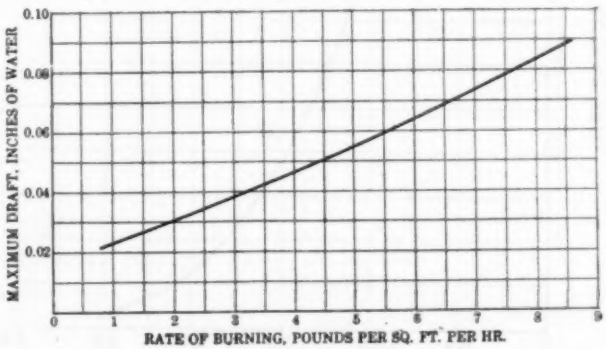


Fig. 4 Draft Above Fuel Bed as Related to Rate of Burning 1- to 1 1/2-In. High-Temperature Coke

Because the CO above the fuel bed varies in amount during a period, the quantity of secondary air supplied should also be varied if it is to be restricted to just that amount necessary to burn the CO. This, of course, is not practical and the best the householder can do is to open the slots a certain amount and leave them so; moreover, the only way he can set them, if he thinks of it at all, will be by guess, and the results obtained will be a matter of chance. However, it is logical to make comparisons on the basis that each size of coke has an equal chance. The basis adopted was that for each size the slots are opened such an amount that the average of the CO_2 over the whole firing period would be 12 per cent. The quantity of primary air necessary to burn or gasify the same weight of each size of coke in the same time was determined by the tests; the composition of the gases and the draft in the furnace were also known from the tests; the draft at each instant fixes the rate at which secondary air will be drawn into the furnace; a size of slot opening could therefore be found which would give the specified average CO_2.

Fig. 6 shows for each size of coke the weights of primary and secondary air required per square foot of grate area per hour when burning at the average rate of $3^1/_2$ lb. an hour. The $^3/_4$-in. mean screen-size coke requires about equal quantities of each, and the quantity of primary air increases and the secondary decreases as the size of coke increases.

If there were complete control over the admittance of the secondary air and no leakage around the slides of the slots and

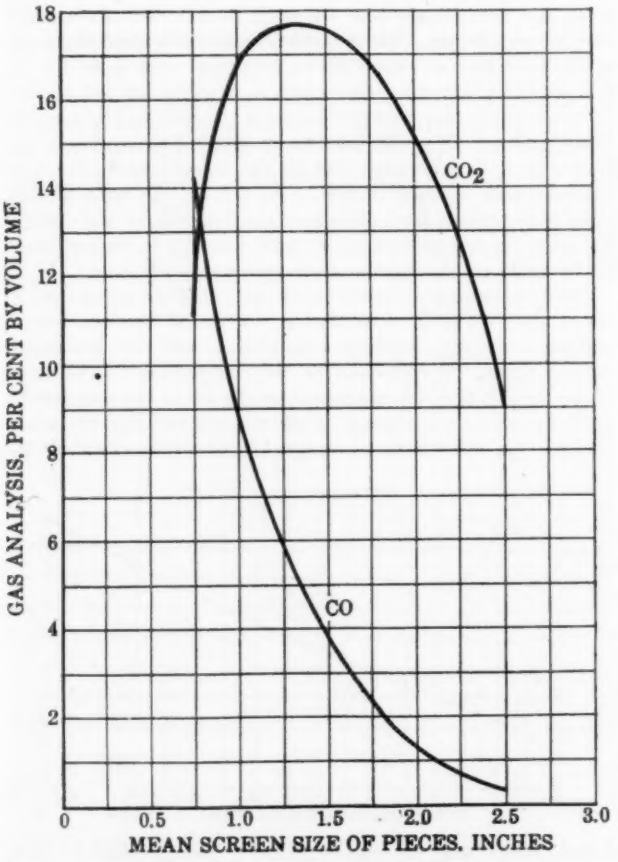


FIG. 5 CO_2 AND CO IN GASES ARISING FROM FUEL BED, AVERAGED OVER PERIOD BETWEEN FIRINGS, AS RELATED TO SIZE OF COKE, FOR HIGH-TEMPERATURE COKE BURNING AT THE AVERAGE RATE OF $3^1/_2$ LB. PER SQUARE FOOT PER HOUR

around the door, then that admitted could be gradually reduced until a $2^1/_2$-in. mean size of coke was reached when no secondary air would be required, and the necessary air to give a 12 per cent CO_2 would come through the fuel bed. Actually, there is considerable leakage and, based on tests of a commercial furnace, Fig. 6 shows that such leakage would supply all the secondary air required by a $1^1/_2$-in. mean size coke. Therefore, the assumed condition of 12 per cent CO_2 could not be obtained with cokes larger than $1^1/_2$ in., and the total air used would rise, as shown by its curve.

The same forms of curves would have resulted if 10 per cent average of CO_2 had been adopted for the basis of comparison. The quantities of primary air would of course have remained the same; the quantities of secondary air would have been increased by approximately the same amount for all sizes.

The slots in a firing door are in effect orifices, and knowing the draft in the furnace, the correct size of opening to admit the required amount of secondary air can be predicted; however, from this required opening an area equivalent to the cracks around the door must be deducted to give the slot area. The simple thin-plate orifice law is:

$$\text{Weight of air} = \text{constant} \times \text{area} \times \sqrt{\text{draft in furnace}}.$$

Tests made by the Bureau of Mines of a door of a domestic furnace showed, as would be expected, that there was appreciable friction in addition to the orifice effect, and the index for the draft is 0.57 instead of the square root.

A domestic furnace is not operated at the same rate all the time. It is of interest to see whether it would be advisable for a householder to adjust the slot opening to a definite position and leave it the same during all the season, or whether he should change it according to the weather. Fig. 7 shows the relative area of slot opening for a given furnace by different sizes of coke, as dependent on rate of burning, and to give equally good combustion. It will be seen that the size of slot opening increases with the rate, and that it would be more efficient to have more opening in December to February than in the Spring and Autumn. However, things would even up somewhat if the opening best for a mean rate of burning of about three pounds were chosen.

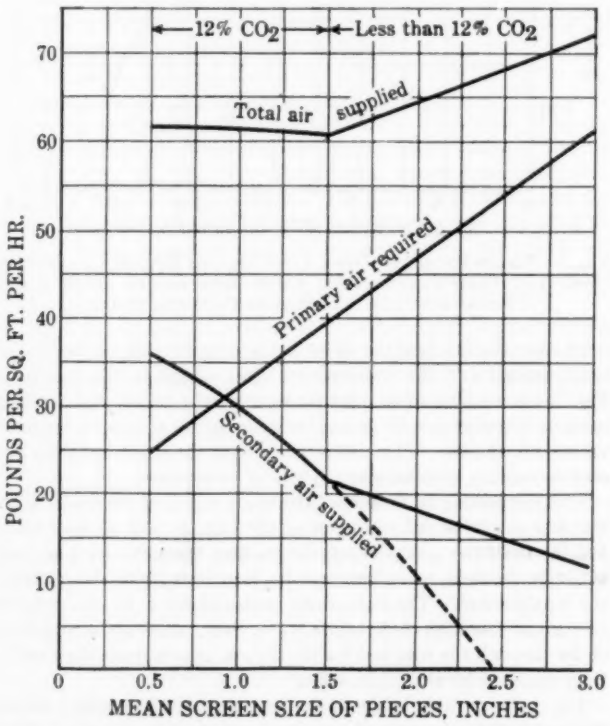


FIG. 6 WEIGHTS OF AIR REQUIRED AS PRIMARY AIR SUPPLIED FROM THE ASHPIT AND SECONDARY AIR SUPPLIED OVER THE FUEL BED TO OBTAIN THE SAME EFFICIENCY OF COMBUSTION WHEN USING DIFFERENT SIZES OF COKE AT THE SAME RATE OF BURNING, FOR HIGH-TEMPERATURE COKE BURNING AT THE AVERAGE RATE OF $3^1/_2$ LB. PER SQUARE FOOT PER HOUR AND WITH 12 PER CENT AVERAGE CO_2

Fig. 7 shows at what rate of burning for each size of coke the full opening of the slots would not be large enough to supply the secondary air required, which, for instance, would be at 7 lb. an hour for the 1- to $1^1/_2$-in. coke. However, more air could be drawn in by partly closing the ash-pit damper as Fig. 7 is based on zero ash-pit pressure. This figure also shows, for each size of coke, the rate of burning which must be reached before it is necessary to begin opening the slots. This for a $1^1/_2$- to 2-in. coke would be 4 lb. per sq. ft. per hour.

There is therefore an area of slot opening which for a given

coke burned in a given furnace at a given rate will, on the average, give the best results, and the data shown indicate what it should be for the coke tested. These data are based on a zero ashpit pressure, that is, with the ashpit damper open. The closing of this damper increases the draft above the fuel bed by an amount equal to the draft produced in the ashpit, and is equivalent to increasing the resistance of the fuel bed by carrying a greater thickness of ash. Such increase of draft above the fuel bed will therefore decrease the area of slot opening required. The effect upon Fig. 7 of closing the ashpit door will be to move the curves to the right and also to decrease their slope.

It is of interest to consider the function of the ashpit damper. It has been seen that closing it increases the draft over the fuel bed, and therefore the quantity of secondary air that will be drawn in for a given opening of the slots. Also, because it increases the equivalent fuel-bed resistance, it makes it easier to adjust the rate of burning in that it requires a larger movement of the stack or choke damper to obtain the same change in the rate of burning. When burning large coke, the householder will find that he can adjust the rate more easily if he keeps the ashpit damper closed, but by doing this he may draw in more secondary air than is needed, even though the slots be closed. With small coke the resistance of the fuel bed itself is higher, and, if his chimney draft is small, to obtain the rate he may find it necessary to open the ashpit damper; at the same time he should increase the slot opening. This will be satisfactory as long as the slots are large enough to supply all the air required while still maintaining zero pressure in the ashpit, but will limit the best control if the slots are not large enough.

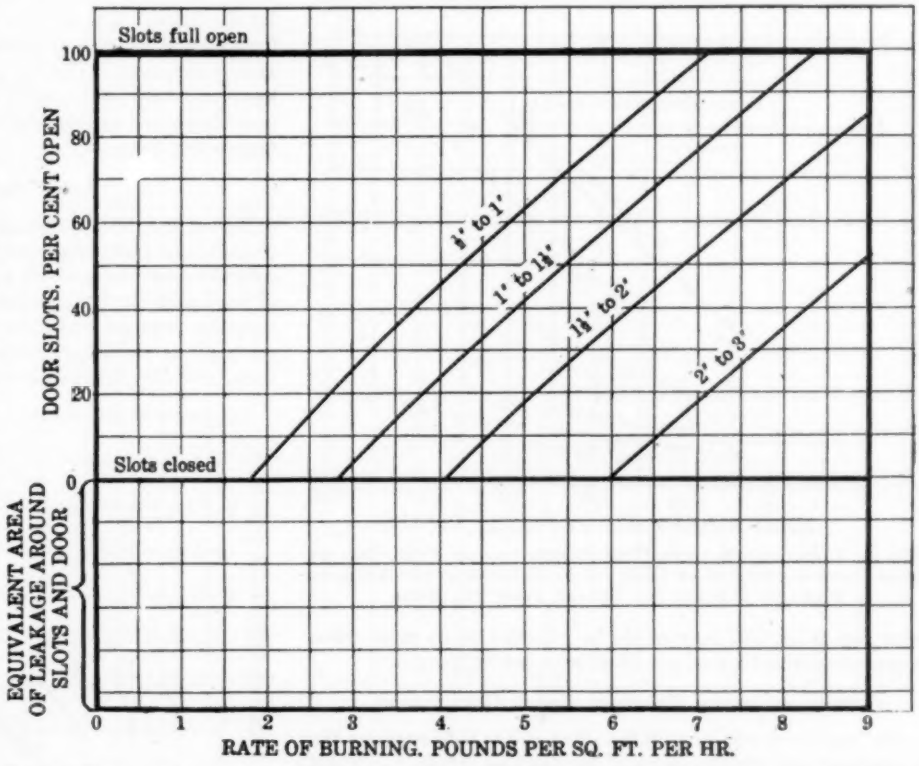


FIG. 7 RELATIVE AMOUNT OF FIREDOOR SLOT OPENING REQUIRED IN A GIVEN FURNACE TO GIVE EQUALLY GOOD COMBUSTION FOR HIGH-TEMPERATURE COKE OF VARIOUS SIZES WHEN BURNED AT VARIOUS RATES

The question can be asked whether a householder using coke and guessing at the amount of slot opening to allow will benefit more by tending to keep it large or small. It is important not to have unburned CO passing into the chimney because the resultant loss in efficiency is high. An average of 1 per cent CO means a loss in efficiency of $3^1/_2$ to $4^1/_2$ per cent, whereas a variation in the excess air as expressed by a variation in the CO_2 from 10 to 16 per cent is not great (17). It would therefore seem preferable to allow more than enough secondary air. On the other hand, the mixing of the secondary air with the gases is not perfect, and some CO gets to the stack even with high secondary air. Tests have shown (12) that the difference between the CO content with 12 and with 16 per cent CO_2 is small; that is, it is possible to operate a domestic furnace with a low excess of air when burning coke.

SIZE OF COKE, UNBURNED FUEL, AND EASE OF STARTING

Fig. 2 shows that the method of test was to use 30 lb. of coke to start the fire, then run two full firing periods, and then allow the fire to burn itself out. The residue was taken out and the weight of unconsumed combustible in it determined. Fig. 8, curve *A*, shows the pounds of combustible per square foot of grate area for the 20-in. furnace. The curve shows that the unburned fuel increases rapidly with the size of coke.

A desirable quality of a fuel, from a householder's viewpoint, is that it shall keep alight as long as possible, so that he will not have to relight the fire if he delays to refire it. The quantity of unburned fuel is a measure of its ability to keep alight, and the curve shows that it will be necessary to refire earlier with large coke than it will be when a smaller size of coke is used.

A fire that goes out more easily is for the same reason hard to start. The ease of starting was also measured directly by recording the time it took to burn the same weight (gas was used to light the coke), and the values come in the same order but are not as reliable data. The fact that it is more difficult to start a fire with large sizes of fuel is common experience, although there is a limit to the smallness because of the coke falling through the kindling and smothering the flame.

EFFECT OF SIZE OF FIRE POT

The data shown previously were for a fire pot 20 in. in diameter. The influence of size is limited to the relative cooling effect of the sides on the fuel bed; therefore, as the area of the fuel bed is increased it would be expected that:

1 The resistance of the fuel bed to the flow of air would be somewhat less, because the area near the sides has less resistance than the center portion. On the other hand, the temperature head of the hot gases will be somewhat greater. The changes in these two effects will be small, but both will tend to reduce the draft required.

2 The combustion of the fuel will be better because the cooling effect is smaller, and therefore less primary and more secondary air will be required with the larger fire pot.

3 There will be less combustible per square foot of grate area left in the fire pot when the fire dies out, because the greater portion of such combustible is found around the sides.

The tests when using the 28-in. furnace supported these expectations. The composition of the gases was slightly higher in carbon content and the combustible in the residue less as shown by curve *B* of Fig. 8. As might be expected, the effect of the size of the furnace increases as the coke pieces are larger. The differences between results with the 20- and 28-in. fire pots

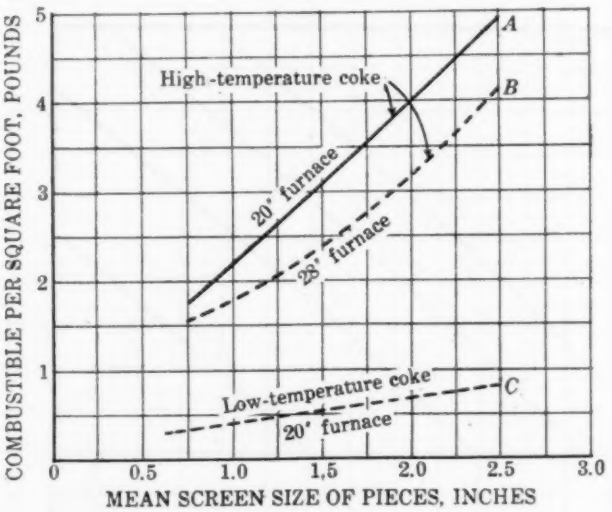


FIG. 8 COMBUSTIBLE LEFT UNCONSUMED IN THE FUEL BED AS DEPENDENT ON THE SIZE OF COKE—FUEL BURNING AT THE AVERAGE RATE OF 3½ LB. PER SQUARE FOOT PER HOUR

were not great, but they would be expected to be more when comparing a smaller size, say 15-in. with one of 20-in.

SUMMARY OF THE EFFECT OF SIZE OF COKE PIECES

The easiest way to sum up the effect of size is to compare the advantages and disadvantages of the smallest (½ to 1 in.) and the largest (2 to 3 in.) size of cokes tested.

Advantages. With the small coke the favorable points are: (1) It is easier to start the fire; (2) it is less likely to go out; (3) there is more range of damper adjustment, and thus the furnace will be more easily controlled; and (4) firings will not have to be quite as frequent.

Disadvantages. The poor qualities with small coke are: (1) A much greater draft is required; (2) a hotter fire can be obtained and thus worse clinkers may be formed; and (3) the efficiency will be lower, both because the CO will be higher due to poor mixing of the gases and air and because the exit temperature of the gases will be higher due to there being more combustion above the fuel bed and a higher initial temperature of the gases entering the passes.

Householders will differ in their preference for the large- or small-size coke, but usually would be more critical of qualities that caused them personal trouble or annoyance. It can be believed, however, that the best average satisfaction would be obtained with a medium size, and a 1½-in. mean screen size would bring fewer complaints. This size presents the further advantage of requiring little or no slot opening, especially if the ashpit damper is kept closed.

DIFFERENCES BETWEEN HIGH-TEMPERATURE COKES

Differences in high-temperature cokes because of the nature and amount of ash they contain need not be considered, and they are fixed by the coal from which the coke is made.

Extensive studies have been made by the Bureau of Mines (15, 16, 17) and the Massachusetts Institute of Technology (14) of the properties of coke made by different processes from the same coal. The chemical compositions of the cokes were closely the same, and the differences found in their use in domestic heaters can be associated with their physical characteristics. Those which were lighter in weight were also softer, and tended to abrade and create fines and dust more easily. A complete set of tests was run on a heavy- and a light-weight coke, the data previously shown being for the latter. The average composition of the gases over the fuel bed and the ratio of primary to secondary air were approximately the same. The light coke ignited more easily and left less combustible in the residue. Fig. 9 shows this weight of combustible plotted against size for a heavy (by-product) and light (gas-house) coke.

LOW-TEMPERATURE COKE

It may be expected that there will be greater variations in the chemical and physical properties of low- than of high-temperature cokes because the methods proposed, in the experimental stage, of producing the low-temperature coke vary materially in temperature, pressure, and time duration of treatment. The chemical analyses which have been published have shown wide variations, and the physical properties will evidently vary still more

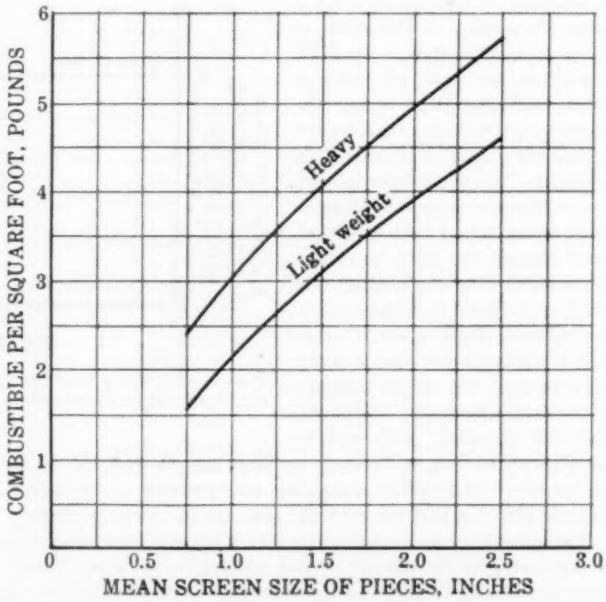


FIG. 9 COMPARISON OF AMOUNT OF UNBURNED FUEL IN RESIDUE FOR A HEAVY- AND LIGHT-WEIGHT HIGH-TEMPERATURE COKE

because the processes vary considerably, some including a briquetting under heavy pressure of the coked product. It follows that the characteristics of low-temperature cokes when burned in domestic furnaces will differ and that generalization should not be made from tests of a few.

It is, however, of interest to consider the tendencies that some of the characteristics of low-temperature coke can be expected to show relative to those of high-temperature coke. A complete set of tests identical to those described for the high-temperature cokes have also been made on one low-temperature coke. These tests were complete and determined not only the CO, but also the hydrogen and hydrocarbons arising from the fuel. Only a few of the data will be presented.

The proximate chemical analysis of the low-temperature coke was:

Constituent	Per cent
Moisture	2.6
Volatile matter	13.2
Fixed carbon	69.3
Ash	14.9

Fig. 10 shows the average percentage CO and CO_2 in the gases arising from the fuel bed, plotted against size, when burning at the rate of $3^1/_2$ lb. per sq. ft. per hr. a low-temperature coke, a high-temperature coke, and a Pennsylvania anthracite. The difference between the values for the high-temperature coke and anthracite is not great, but the low-temperature coke differs materially from both. It indicates that the low-temperature coke unites with the oxygen of the air and forms CO much more readily than do the other two. This is expressed technically by saying it has a higher combustibility and reactivity, and popularly by saying it burns more readily.

It will be noted from Fig. 10 that the relative difference between the percentages of CO and CO_2 for the low- and high-temperature cokes is greater for the large sizes than it is for the small; that is, the properties more nearly approach each other as the size gets smaller. It will also be seen that the CO and CO_2 of the $2^1/_2$-in. size low-temperature coke are about the same as those for the 1-in.

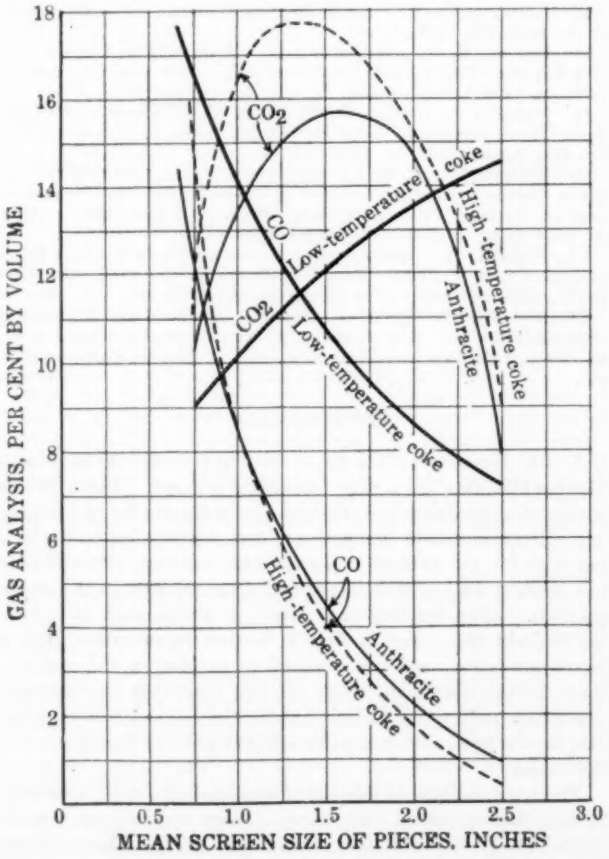


FIG. 10 COMPARISON OF AVERAGE COMPOSITION OF GASES ARISING FROM THE FUEL BED WITH DIFFERENT SIZES OF A HIGH- AND A LOW-TEMPERATURE COKE AND AN ANTHRACITE—RATE OF BURNING $3^1/_2$ LB. PER SQUARE FOOT PER HOUR

high-temperature coke; that is, to obtain equivalent results in this respect a larger size of low-temperature coke must be used.

Fig. 11 shows the primary and secondary air required for the low-temperature coke; this corresponds to Fig. 6 for the high-temperature coke. It shows that the ratio of secondary to primary air is greater, and that less draft and more opening of the firedoor slots will be required for the low-temperature coke.

The amount of combustible in the ashpit residue was included in Fig. 8 as curve C. It is much less than for the high-temperature cokes, showing that the refiring can be delayed longer. One can also deduce that the low-temperature coke was easier to ignite, and this was found to be so for all sizes.

It has been shown generally that all low-temperature cokes have the properties described above, although it is to be expected that their departure from similar values for high-temperature

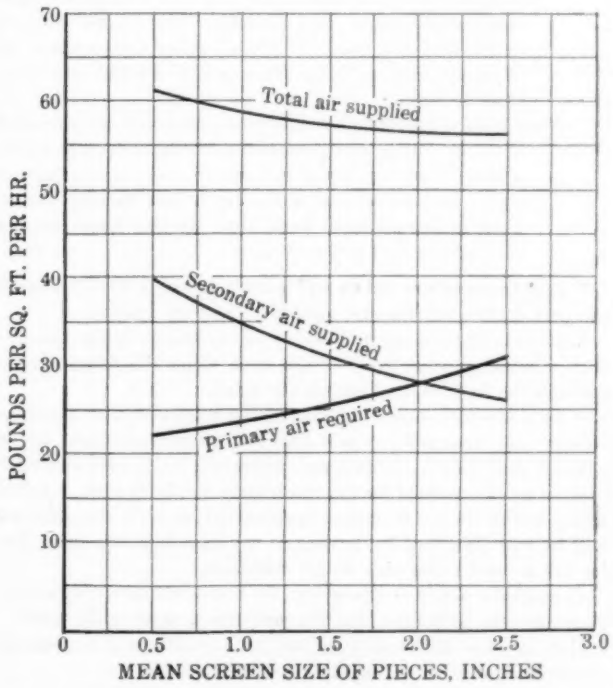


FIG. 11 WEIGHTS OF PRIMARY AND SECONDARY AIR REQUIRED BY A CERTAIN LOW-TEMPERATURE COKE BURNING AT THE AVERAGE RATE OF $3^1/_2$ LB. PER SQUARE FOOT PER HOUR AND WITH 12 PER CENT CO_2

coke will not be the same. Some tests which have been made on two other low-temperature cokes show that one tended nearer to the high-temperature coke and one further away than that discussed in the foregoing.

Certain safe deductions can be made on the relative performance and characteristics of low- compared with high-temperature cokes when burned in domestic furnaces. The numerical values corresponding to these deductions will vary with the furnace and with the attendance; the comparisons are on the basis of equal sizes of the two types of cokes:

1 Low-temperature coke will require less kindling and less time to start the fire.

2 Low-temperature coke will burn more readily and will require less air through the fuel bed and less draft; consequently, the dampers must be in good condition or it will not be possible to control the burning so that the house will not be overheated.

3 Low-temperature coke will require more air above the fuel bed.

4 Low-temperature coke when banked is never likely to cause trouble through the fire going out, leaving the fire pot full of unburned fuel, but rather it will be more likely to burn away before morning.

5 Low-temperature coke could be successfully refired with a smaller amount of live fuel in the fire pot.

6 The efficiency with low-temperature coke—

a will tend to be higher because there will be less combustible loss in the ashpit, and the amount, with reasonably careful handling, will be small.

b will tend to be lower because the gases from the fuel bed will contain a larger percentage of combustible, and there will be a greater probability of a larger amount passing to the chimney unburned because of poor mixing with the air supplied through the firing door.

c will tend to be lower because the larger amount of combustible gases above the fuel bed will require a larger combustion space, and therefore the flames will be longer and will tend to give a higher temperature of the gases passing to the flues and thus a higher loss in sensible heat.

d will tend to be lower because, when fired in large amounts with the consequent complete covering of the live coals, a larger loss will occur because of the volatile matter which is distilled out of the fresh fuel passing to the flues unburned until such time as the flame breaks through the bed and ignites it.

7 Low-temperature cokes will give a haze of smoke during the period after refiring and until the gases are ignited.

8 There will be more probability of explosions in the furnace with low-temperature coke at the time when the flame breaks through the fuel bed and ignites the gases.

A first consideration of the foregoing might lead to the conclusion that householders will find low-temperature coke to be a less desirable fuel. This is not necessarily so, as less discord is created in a household by too much than too little heat. A fuel which burns readily is always appreciated, and the householder will be less disturbed by having to try to reduce the heat that he will by not being able to get sufficient.

It would be a mistake, however, for manufacturers of low-temperature coke to believe that the production of an easily burning fuel is the sole desirability or that this quality will necessarily counterbalance the accompanying disadvantages.

Petroleum Coke

The sale of petroleum coke for domestic furnaces has been small and confined to the Middle West. E. B. Swanson, of the Bureau of Mines Petroleum Economics Division, in an unpublished report states that its largest sales occur in Kansas City, Chicago, Minneapolis, St. Paul, and Wichita, Kansas.

No analytical tests have been made of petroleum coke, but its burning qualities resemble those of low-temperature cokes, though the absence of ash necessitates a different method of handling. Most of the small amount of ash it contains is carried up the stack. None remains on the grates; consequently, the burning fuel is in contact with the grates and destroys them in a short time. They must therefore be protected by a layer of broken firebrick or of ash or clinker from some other fuel. The former method is preferable as the firebrick will take longer to fuse together.

Acknowledgments

The experimental work reported in this paper was conducted at the Pittsburgh Experiment Station of the U. S. Bureau of Mines under the authority of O. P. Hood, chief mechanical engineer; M. L. Orr, assistant fuels engineer; and W. M. Brewer, junior fuels engineer, assisted by Charles Schramm, cooperated in making the tests.

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Discussion

A. M. Beebee.[5] This paper contains most valuable and worth-while data on a most important subject. There is one comment which impresses the writer as being worthy of bringing up. Experience both practical and technical has led to the feeling that for the average domestic-coke customer there is only one position for the slide in the fuel door, namely, in the closed position. After reading the paper it seems that the data, particularly the curve in Fig. 7, further substantiate such a recommendation, and yet a complete evasion of this point is made in the discussion, when through ignorance and previous experience with other fuels it has become almost general practice for the coke customer to be advised to leave the slide in the door open.

The tests outlined in this paper were carried out in theoretical rather than practical equipment. Work carried out for the 1926 A.G.A. Carbonization Committee by the Massachusetts Institute of Technology and the Koppers Company, report of which follows, was carried out in practical standard equipment and further confirms the logic of this recommendation. Reference to these data is particularly interesting. Therefore, it seems that in view of the already rather general practice it is highly desirable that rather definite recommendations be made with regard to the leaving of the slide in the fuel door in a closed position when burning all but the very finest coke fuel.

[5] General Superintendent, Gas Department, Rochester Gas and Electric Corporation, Rochester, N. Y.

Proper Combustion of Coke—Slide in the Door Closed—1926 A.G.A. Carbonization Committee Report

Apparently the practice of leaving the slide in the door open is a result of the use of anthracite coal, which the following data show to be necessary in the case of coal. The 1926 Carbonization Committee brought out some points on the subject in the tests which were carried out as part of its program. H. W. Rose, Assistant Chief Chemist of the Koppers Company, in his report on a similar set of tests reports:

Two new Ideal Arco boilers were recently installed in a domestic-coke testing station. Using three different by-product cokes, it was found that both of these boilers, when operated at the normal 35 per cent rate with the secondary air damper closed, leaked enough air to burn up all of the CO, and gave, in addition, very high excess oxygen in the stack gas. In the majority of the cases no CO whatever was found, and the highest amount was 0.3 per cent, which is almost within the error of gas analysis. The average percentage of excess oxygen was 9.5 per cent (equivalent to 45 per cent air). When the secondary air damper was opened, the CO content of the stack gases was 0.0 per cent or 0.1 per cent, and there was 14.4 per cent excess oxygen (or 69% air). This means that there were more than two parts of air mixed with each part of true stack gas.

When the boilers were operated at the excessively rapid rate of 120 per cent, with the secondary air damper closed, the average CO percentage found in the stack gases was only 1.8 per cent. At the same time the average CO_2 content was 17.7 per cent and the oxygen percentage 1.9 per cent so that the efficiency was very good indeed. Even at this excessive rate of firing, every trace of CO disappeared as soon as the secondary air damper was opened, the following average analysis being obtained: 0.0 per cent CO, 16.2 per cent CO_2, and 4.7 per cent O_2.

In the paper by P. Nicholls of the U. S. Bureau of Mines before the A.G.A. in 1926 as a result of his tests appears the following statement:

The curves show that the secondary air required over a fuel bed decreases with increase in size of coke and that absolutely none will be required for the high-temperature cokes having a mean screen size of about 1¾; that is, the door slots should be kept closed. For larger sizes the air needs are negative, which is equivalent to saying that they could not be used with equal efficiency.

In Professor Haslam's paper on the combustion of cokes (in the 1926 Carbonization Committee report) in a domestic furnace the curves shown in Fig. 12 resulted. Note the large amount and continued duration of CO with anthracite and the small amount and duration in the case of various cokes. Professor Haslam concludes:

However, the larger amount of CO generated both in amount and time of duration with anthracite fuel shows the need for admitting secondary air over the fuel bed when using this fuel. Due to the smaller amount of CO generated with coke fuel, admission of secondary air is of less importance in securing maximum efficiency. The change in the ratio, however, indicates the necessity for more frequent adjustment in the case of coke than in the case of anthracite coal. Finally, in so far as these tests are representative of ordinary operating conditions, there is much less need of secondary air with coke fuel than with anthracite and less loss in proper air adjustments are not made by the householder.

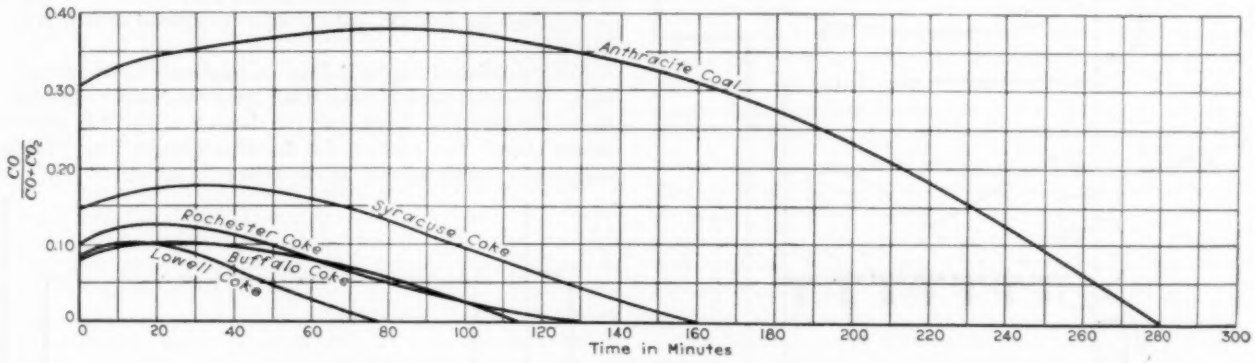


Fig. 12 Curves From Professor Haslam's Report on Combustion of Cokes in Domestic Furnaces—American Gas Association—1926 Carbonization Committee Report

Conclusion

From the above it appears that customers should be advised to keep the slides in doors closed, not only to improve combustion efficiency but also to reduce the danger of products of combustion being given off in the cellar of a customer, especially immediately after firing.

Henry Kreisinger.[6] High-temperature coke has been used by the writer for house heating during the last five winters with very satisfactory results. The size used is about 1 to 1½ in., and is the smallest size that can be obtained from local coal dealers. The indications are that better results could be obtained even with a smaller size than this. One winter the coal dealers did not have the regular sized coke on hand, so a load of a gas-house coke of much larger size was purchased. Some of the pieces were 3 to 4 in. in diameter and were entirely too large. It was difficult to keep the fire going with this large coke. Unless the fire was kept very hot it would go out, and therefore satisfactory uniform heating could not be obtained. I think that much of the difficulties encountered in burning coke are due to too large size. In my own case ½- to 1-in. size would give the most satisfactory results. Small size coke is especially desirable for overnight banking. It will keep fire much better than larger sizes.

The house in which coke has been used has 10 large rooms, but only six of these are regularly heated. The heating plant is a one-pipe steam system. The boiler is of square shape with a fire box 21 in. high, 24 in. wide, and 30 in. long. The ashpit is 12 in. deep.

The method of firing is shown in Figs. 13 and 14. Before firing the burning coke is pushed with the coal scoop to the rear part of the grate, and the front is then filled with fresh coke as indicated in Fig. 14. This method of firing keeps the surface of the rear part of the fuel bed always glowing so that the combustible gases rising from the freshly fired coke can be ignited and burned as they pass over the glowing coke. Covering only part of the fuel bed with fresh coke also produces more uniform heating, because the uncovered glowing coke keeps on generating heat which is available for producing steam during the period when the freshly fired coke is brought up to ignition temperature. Covering the entire fuel bed with fresh fuel is always objectionable no matter what fuel is used.

[6] Research Engineer, Combustion Engineering Corp., New York, N. Y. Mem. A.S.M.E.

In case the fire has burned down too low, it is advisable to put in a piece of wood as shown in Fig. 15 before placing the fresh charge of coke into the furnace. The wood insures the ignition of the fresh coke. It is not necessary to put in much wood. One ordinary piece of kindling wood goes a long way to hold the fire and start the ignition. It is common practice among blacksmiths to bury a piece of wood in the glowing coke in their forge if they wish to keep the fire over an extended length of time. The wood upon heating distils combustible gases which ignite easily and start the coke burning. The wood also keeps glowing with very reduced air supply through the grate, and may be particularly useful with banked fire.

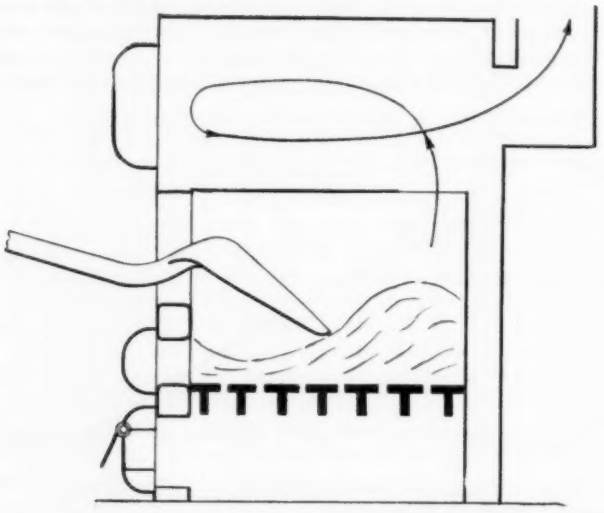


FIG. 13 BEFORE FIRING, BURNING COKE IS PUSHED TO THE BACK OF THE GRATE

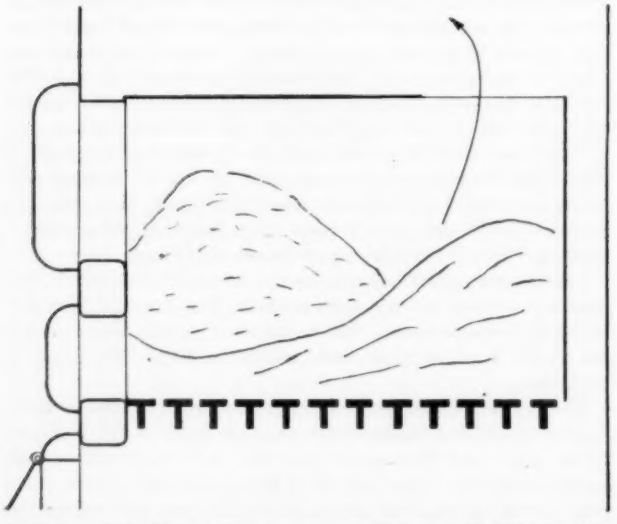


FIG. 14 THE FRONT SECTION OF THE FIREBOX IS FILLED WITH FRESH COKE

Banking fires for the night is accomplished in a similar way as the firing, with the possible addition that the pocket for the fresh coke on the front part of the grate is made somewhat deeper, the fuel with greater amount of small pieces is picked out for banking and piled up somewhat higher than for ordinary firing. It is very seldom that fire goes out overnight.

The frequency of firing depends on the weather. In mild weather two firings a day keep the house comfortable. In cold weather the number of firings may have to be increased to three or four or even five.

When burning coke it is of great advantage to have a boiler with a large furnace volume. Coke is much less dense than coal, and the pieces are more irregular and rougher, with the result that the coke makes a very loose pile in the furnace, and only about 50 to 60 per cent of the weight can be put in the same space as with anthracite. A large coal scoop greatly reduces the work of coke firing.

The coke used contains 8 to 10 per cent ash compared to about 20 per cent ash in anthracite available in this locality. The ash is generally fine dust that sifts through the grate bars into the ashpit by gently shaking the grate. Rocking of the grates is not desirable and necessary because it is sure to drop unburned coke into the ashpit. The total amount of ashes taken out is two to three bushel buckets a week. The amount of clinkers is very small and does not exceed 5 per cent. It is pulled out through the firedoor, and the labor connected with it is insignificant.

The regulation of the fire is done automatically by steam pressure. When the pressure exceeds 2-lb. gage the damper admitting air to the ashpit is closed, and the damper in the flue pipe is nearly closed. To prevent the flue-pipe damper from closing completely a small segment in the damper is cut out.

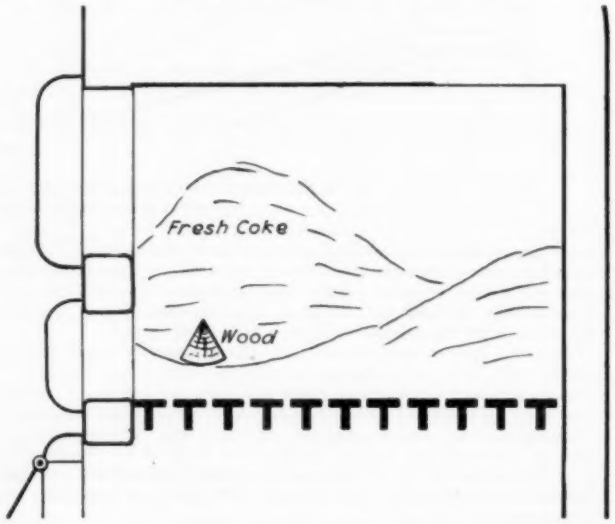


FIG. 15 A STICK OF WOOD HELPS TO IGNITE FRESHLY APPLIED COKE

H. J. ROSE.[7] There is a most important development which must not be overlooked in any discusion of coke as a domestic fuel. Domestic coke is a distinct type of coke, made to specification from selected coals, in order to produce the best possible fuel for the domestic trade. Less than 20 years ago (it is perhaps equally correct to say less than fifteen years ago) there was little coke offered as household fuel which could meet the above requirements. Much of the coke offered as household fuel was really not domestic coke at all; it was the rejects or small sizes of metallurgical coke. At some plants these rejects were, and today still are, good domestic fuel. But far too often such products were high in ash, had a low ash-fusing temperature, and were poorly sized. All three of these factors combined to cause clinker trouble and other complaints.

Coke from gas retorts was another variety which usually failed to raise coke in the public estimation. Perhaps the outstanding factor of genuine domestic coke today is its low ash content.

[7] Assistant Director of Research, The Koppers Co., Mellon Institute, Pittsburgh, Pa.

Certain coke plants are now producing domestic coke which contains less than 7 per cent of ash. We find that so long as low-ash coke is put into customers' bins, few customers are lost.

Proper merchandising methods and technical service are other factors in the outstanding success of those plants which are producing domestic coke that is made to specification. There are still many localities where it is difficult to buy the high-grade domestic coke that has just been described. Fortunately, it is being made and sold in continually increasing quantities.

HORACE C. PORTER.[8] The authors' statement that coke "is referred to as the ideal fuel from that standpoint" (smokelessness), might be challenged by the anthracite advocates since it is practically impossible to make smoke in burning anthracite coal of the better grades. Furthermore, anthracite has a small amount of smokeless volatile matter which in some respects improves its combustibility.

Among the objections or complaints listed by the authors as being applied to coke, they do not include (1) the additional bulk of fuel in storage, in other words, the extra space required for bins in the cellar; (2) the additional work of shoveling for the person handling the fuel due to its bulk, and the difficulty of shoveling caused by coke's peculiar way of falling from a pile or sticking in corners.

The curves given in Figs. 5 and 10, showing the relative amounts of CO and CO_2 in the gases over the fire from coke of different sizes, indicate clearly that small-sized coke gives much more CO than the larger sizes and therefore requires secondary air in order to maintain the best efficiencies. The same thing would apply no doubt in the case of small-sized anthracite. However, in practical operation of a household furnace, excess air is drawn through holes along the walls of the furnace which counteracts this difficulty. It would be of interest to have data on the amount of CO in the chimney gases under household burning conditions. Perhaps some one of the anthracite interests can furnish these.

The tests show that medium-sized coke 1 in. to $1^1/_2$ in. appears to work out the best. The coke producers have already found this out apparently, and are promoting sales of this size actively.

Low-temperature coke is well known to have in most cases a very high rate of combustibility. The unburned fuel in the ash should therefore be very low with this grade of coke, and the authors have called attention to the gain in efficiency that results from this factor.

Lykens coal (anthracite) has a much higher rate of combustibility than most of the other anthracites, and furnaces that burn Lykens coal or briquets made from it have much smaller losses of unburned fuel than those burning harder anthracites. Low-temperature coke would bear a similar relationship to high-temperature coke.

The factors listed by the authors as tending to cause lower efficiencies with low-temperature coke will not necessarily hold true if the right kind of furnace equipment is used, adapting it to the fuel.

The whole matter of improving solid fuels for household use is tied up with the new developments now going forward rapidly in automatic firing and automatic control. This development has been boosted by the increased use of oil and gas which are peculiarly adapted to such automatic control. There is some question whether the virtues claimed for coke in its low content of ash are fully justified beyond a certain point. A very low content of ash may give rise to a necessity for delicate and careful control of the fires that will make difficulties for the average householder. The designers of automatic control, however, are likely to find ways of overcoming this difficulty.

It would seem to be desirable for some agency like the Bureau of Mines to make comparative tests of efficiency and overall returns in house-heating by different fuels, using equipment and conditions especially adapted to the characteristics of the fuel that is being burned. Comparative tests heretofore have used equipment and conditions identical for the different fuels, and have thereby placed one or the other at a disadvantage.

[8] Consulting Chemical Engineer, Philadelphia, Pa.

AUTHORS' CLOSURE

The conclusion reached by Mr. Beebee that domestic users of high-temperature coke should be advised to keep the slides in the firing door closed is a rule which on the average should give the best results with the sizes of coke usually sold. There is no reason, however, why the companies that purvey coke in large quantities and have inspectors to assist their customers should not be in a position to discriminate more closely and advise some slot opening when it will be of benefit. The authors purposely avoided stating definite rules. The work which the Bureau of Mines is doing in determining the burning characteristics of fuels should be supplemented by some field surveys. Snap measurements of the composition of the flue gas of installations typical of those in a district would accumulate data which should be helpful to the coke companies in training their inspectors and would show how far the results of tests of commercial heaters made under test-block conditions are representative of regular use.

It is valuable to have the personal experience of Mr. Kreisinger. His choice of a 1- to $1^1/_2$-in. size for use in a heater which is amply large for the rooms heated agrees with deductions from the tests reported in the paper. It is doubtful, however, whether he would find the $^1/_2$- to 1-in. size as economical, unless bought at a lower price. The CO content of the gases from the fuel bed increases rapidly with decrease in size, and the loss of CO to the stack would be higher in the average heater; in addition, a greater quantity of clinker would be formed, and its removal causes some loss of fuel to the ashpit.

It is agreed that Mr. Kreisinger's practice of pushing back the live coke before adding new fuel is correct and will save some loss in efficiency. The loss, when the live fuel is smothered, from combustible gases passing to the flue between the time of firing and when the gases over the fuel bed are ignited has been determined by us, both for high- and low-temperature cokes. It varies with rate of primary-air supply and with size of coke, and is greater for the low-temperature coke because of its volatile matter. With high-temperature coke this loss increases rapidly for sizes below 1 in. and is negligible for sizes above $1^1/_2$ in. An inexperienced fireman would probably lose more fuel through the grate bars than he would save from the combustible gases; it would be safer not to recommend the practice for general use with high-temperature coke larger than 1 in.

The suggestion of Mr. Porter that coke may suffer in comparison with other fuels because of the larger bin it requires to hold a given weight is justified; also it is probable that most people will consider it more inconvenient to shovel than a sized coal, but opinions differ on this and womenfolk in particular have been found to prefer coke.

The authors, as far as possible, avoided comparisons between cokes and coals, and a paper to include such comparisons would be of much wider scope than that presented.

FSP-52-15

Effect of Fouling on Boiler Efficiency

By J. W. PIERSON,[1] CHICAGO, ILL.

The subject of this paper concerns a situation that confronts those in the Middle West who out of necessity are required to burn some of the low-grade fuels of Illinois.

The fouling conditions incident to the use of such fuel with chain-grate firing with high relative ratings are dealt with. Some of the objectionable characteristics of the fuel not shown by usual analysis are pointed out, together with the severe conditions which soot-blowing equipment of any kind would have to withstand.

The data presented show the efficiency characteristics of an operating cycle with this low-grade fuel compared to that which would exist when using a better fuel. Unfortunately the difficulties of the former cannot be solved by the simple expedient of using a better coal. The choice in such a situation is, then, to operate at high ratings with low overall efficiency or with lower ratings with improved efficiency. Lower ratings, however, would reduce station output, and is not a desirable solution.

OPERATION of a modern chain-grate stoker plant burning Central Illinois coal imposes ever-increasing difficulties when, due to the economics involved, still higher ratings than those of former practice are desired. Such a situation is exemplified in the boiler-room equipment of the Crawford Avenue Station of the Commonwealth Edison Company where the matter of maintaining clean heat-absorbing surfaces with such fuel conditions has developed into a major boiler-room problem. These matters are still far from a satisfactory solution.

Even though the coal burned has so many unfavorable characteristics, coal costs are such that the problem cannot be solved economically by the simple expedient of buying a better grade of fuel. The problem therefore is one of carrying on against the difficulties as they are, and will require considerable thought and development on the part of both designers and operators.

A typical analysis of the coal on the "as fired" basis may be expressed as follows:

B.t.u.	10,026
Per cent Moisture	16.95
Per cent Ash	11.95
Per cent Sulphur	4.04
Per cent Hydrogen	3.93
Per cent Carbon	53.10
Per cent Nitrogen	0.86
Per cent Oxygen	9.21

This analysis, however, does not show many of the characteristics of the fuel nor are they indicated by any of the usual forms of analysis. Due to its high volatile content it is a long-flaming coal, and requires liberal use of overfire air injection. Even under these circumstances some long flaming results. The ash has a fusion point of about 1950 deg. fahr., which on being carried as fly ash into the lower bank of tubes and superheater in a molten or viscous state, deposits and adheres firmly to the surfaces as a slag. This slag is composed largely of oxides of silicon, iron, and aluminum. Other portions of the ash were carried through to the economizer and preheater surfaces, where they present a problem of a different nature. The excess air required for complete combustion is higher than would be required for a better grade of fuel. This condition is also imposed by the necessity of limiting temperature conditions, to reduce refractory problems, and also somewhat to prevent worse fouling in the boiler passes.

The sulphur content is high, and while this may not directly affect the rate of fouling, it is important inasmuch as corrosion of economizer and preheater tubes requires equipment outages which impose an additional burden on the rest of the equipment.

There are 32 stoker-fired boiler units at this station, of the general arrangement shown in Fig. 1. The last extension of this

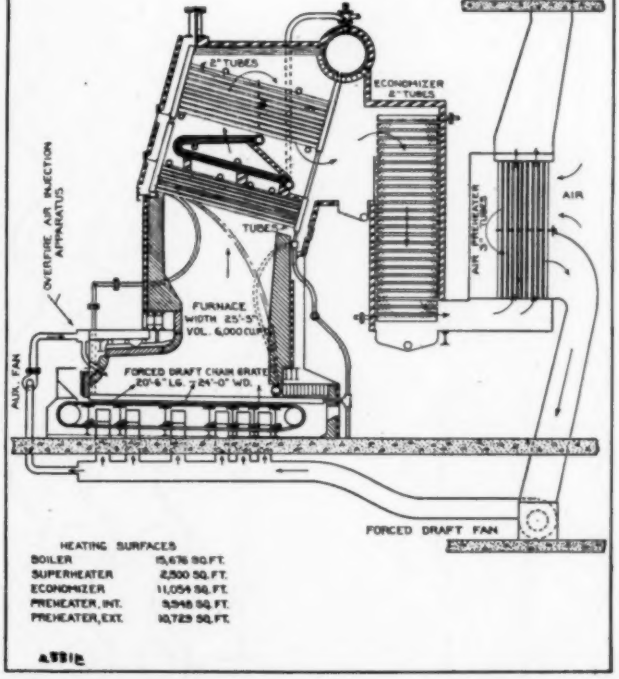


FIG. 1 GENERAL ARRANGEMENT OF BOILER NO. 2, UNIT NO. 1 IN CRAWFORD AVENUE STATION OF COMMONWEALTH EDISON COMPANY

boiler plant has 12 boilers equipped with water-cooled furnace walls. Previous experience with furnace walls indicated beneficial results of the cleaning problem together with reduced refractory maintenance costs.

Plant load conditions require a cleaning and maintenance program during the summer and early fall months which requires 26 boilers available for service. This means that each boiler unit must operate about 35 days, and is out about six days. At the end of the service period the boiler is badly fouled, and operating at a lowered capacity as a consequence. Winter conditions require 27 boilers for service, which means that each boiler unit must operate about 42 days, with the result that steaming capacity and efficiency fall still further before outage. The soot blowers are operated every twelve hours, and the boilers are air or water lanced every fifth night. It is obvious that

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[1] Boiler-Room Engineer, Commonwealth Edison Co.

Presented at the Third National Fuels Meeting, Philadelphia, Pa., Oct. 7 to 10, 1929, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

demands for still higher loadings only lead to a vicious circle, making the cleaning problem a matter of direct or even greater ratio to capacity demand.

The demand for higher superheat largely influenced the design of the interdeck superheater arrangement as shown in Fig. 1, which also shows the location of the blowing elements. It is obvious that the temperatures normally prevailing in the zone of the lower-tube bank and superheater subject blowing equipment to rather severe conditions, particularly to two central elements on the lower bank and the two in the superheater, all of which are in the direct line of maximum temperature. Where failures have occurred they usually take the form of warping or breaking, and in some cases burning off.

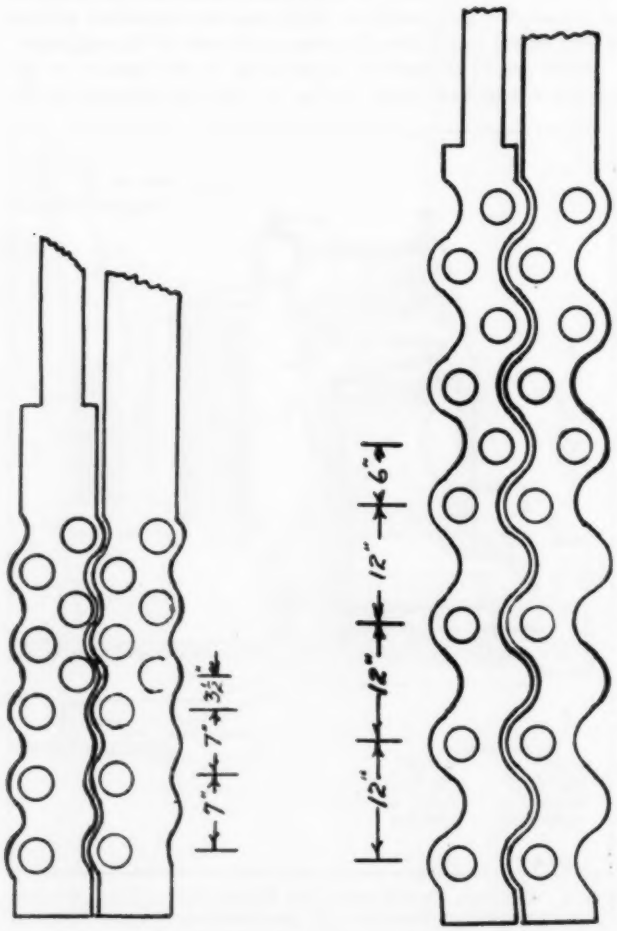


FIG. 2 TUBE SPACING ON LOWER DECK; AT LEFT, EARLY INSTALLATION; AT RIGHT, LATER INSTALLATION

Difficulties have been experienced with bearings which depend upon some form of clamping to hold them in position. Arc welding of steel strips to the boiler tubes to which are bolted alloy straps that hold the elements in place is apparently a solution inasmuch as the bearing is cooled more effectively.

The installation of more elements while seriously considered presents structural difficulties as well as increased steam use which has already reached undesirable proportions; consequently this does not seem to be a satisfactory line of procedure.

The nature of the soot deposit on economizers and preheaters is such that permanently installed blower elements were not considered as an effective means of removing it. Under conditions as stated washing is the only effective method so far found. This is done as the unit is being taken out of service so that excess moisture is evaporated quickly.

Experience with these general conditions has proved that cleaning by means of permanently installed blower elements must be augmented with hand lancing if the best results are to be obtained. One method is the use of the air lance on the lower-tube bank. The air has the effect of keeping the lance cool, enabling the operator to work on the slag deposits more effectively.

The tube spacing as shown at the left in Fig. 2 is that of the early installations comprising the first 14 boiler units. This tube spacing will not permit lancing or rodding from the side above the slagging rows. To meet this difficulty it was necessary to resort to a method of water lancing. While using the water lance the boiler is operated at a low rating, and a stream of water from a $^1/_8$-in. nozzle is directed against the slag from

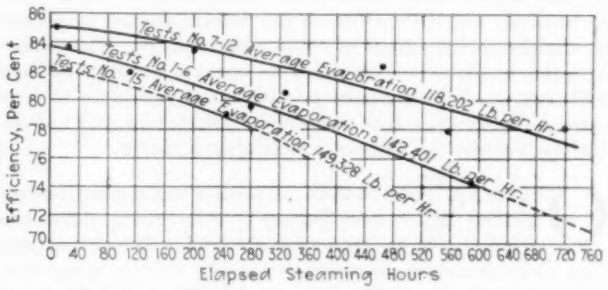


FIG. 3 GRAPHS SHOWING DROP IN EFFICIENCY DUE TO FOULING OF BOILER BURNING CENTRAL ILLINOIS COAL

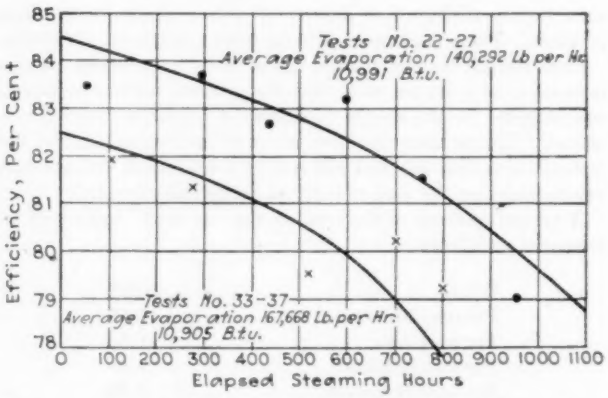


FIG. 4 GRAPHS OF DROP IN EFFICIENCY DUE TO FOULING OF BOILER BURNING WEST KENTUCKY COAL

above and below, which, due to sudden cooling, readily cracks off. This method is not entirely effective because of the inaccessibility of some of the surface.

The tube spacing as shown at the right in Fig. 2 is that of the later installations. It permits greater accessibility both from the front and sides.

Water-cooled furnaces have the effect of reducing the slag deposits by lowering the temperature of the ash before it enters the tube bank. The character of the deposit is different than that found on boilers with refractory furnaces, and is more easily removed.

All these conditions, as may be expected, have their effect on the overall operating efficiency of the boiler room. Evaporative boiler tests do not necessarily indicate the overall operating efficiencies that would be obtained by a given boiler room. As considerable difference occurs, particularly under the circumstances being dealt with, a series of tests representing a cycle of

operation were conducted to illustrate more clearly the true situation with this grade of fuel.

The data presented herewith represent conditions which exist in the operation of a boiler with a water-cooled furnace having the tube spacing as shown at the right in Fig. 2. It is reasonable to conclude that the situation here would be more favorable than that with refractory furnaces.

The boiler unit with its cleaning equipment was put in first-class condition, and operated at a fixed capacity during the day so far as operating conditions would permit. During the night it was operated at reduced rating in order to follow the cycle of boiler-room operation. Boiler tests of eight hours' duration were conducted weekly during the regular operation of the boiler, the coal and water being weighed, and all readings taken to conform to standard test code. The first series of tests were conducted at a capacity of 118,000 lb. of steam per hour, then a series at 142,000 lb. of steam per hour, and finally one at 150,000 lb. of steam per hour.

From Fig. 3 it will be seen that at a capacity of 118,000 lb. per hour the efficiency is 85 per cent at the start, decreasing to approximately 77 per cent at the end of 720 elapsed steaming hours. With a capacity of 142,000 lb. per hour the initial efficiency is slightly lower, but the rate of fouling is greater so that at the end of 590 elapsed steaming hours the efficiency has dropped to 74 per cent.

In the case of the series at 150,000 lb. per hour it was found after about 300 hours of service that it was impossible to maintain this capacity due to the fouling of the surfaces. The choice in operation is then either to operate the boiler units at high capacity with lowered overall efficiency or at lowered capacity with improved efficiency, the latter meaning reduced station output.

In order to show how much this situation would be improved with the use of a better grade of coal, a similar test program was carried out with results as shown by Figs. 4 and 5. The coal in this case was Western Kentucky with a heating value as fired about 11,000 B.t.u. The ash content is lower, and the fusing temperature of the ash is higher, all of which makes the whole cleaning problem easier. This was shown in the periodic lancing, and the condition of the heating surfaces at the end of the service period. Fig. 4 shows the efficiency at two capacities, one at an average evaporation of 140,000 lb. of steam per hour and another at 167,000 lb. per hour. The interpolated curves of Fig. 5 show the general situation as it appears to be for the two different kinds of fuel. While only a hypothetic picture, it is based on actual data.

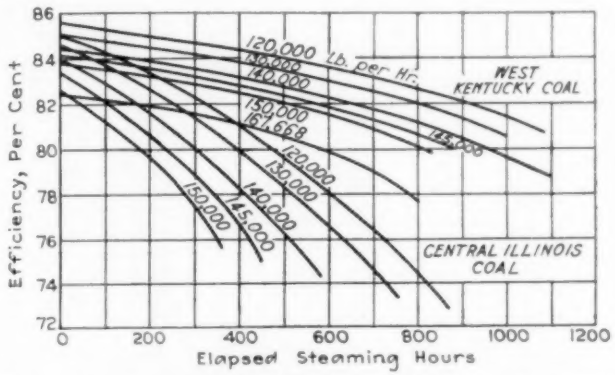


FIG. 5 BOILER AND STOKER EFFICIENCY AT VARIOUS CAPACITIES AND DURING VARIOUS OPERATING PERIODS

In general the curves are similar but show that when using the better grade of fuel the boiler can be operated twice as long before the same drop in efficiency is reached as when using Central Illinois coal. Also with the same operating cycle much better overall efficiencies could be maintained with the better fuel. As the cost of coal, however, imposes the necessity of continuing to burn this low-grade fuel, the problem of efficient operation must take these fuel characteristics into consideration if high boiler ratings and efficiencies under such circumstances are an economic necessity. It is evident, under the existing conditions, that with higher boiler ratings the difficulty of burning this low-grade coal economically will increase in greater proportion.

FSP-52-16

Sampling of Pulverized Fuel

By HORACE C. PORTER,[1] PHILADELPHIA, PA.

IT IS NECESSARY to sample pulverized fuel particularly in order to determine its fineness. In every case where pulverized fuel is used, the fineness of division has a very large influence on the performance of the furnace or other utilization device. It is also necessary to know the percentage of ash, volatile matter, and moisture, and the character of the ash in respect to fusion tendencies. These characteristics can, however, be determined from coal samples taken in the ordinary way before pulverizing.

The American Society for Testing Materials in 1926 adopted a Tentative Standard Method of Test (D 197-26 T) for the fineness of powdered coal in which was specified a method of sampling from the air stream in a pipe. This method has stood as tentative for three years, largely for the sake of securing results of further investigations into the accuracy of this and other methods. Sub-Committee No. 1 on Methods of Testing of Committee D-5 of the American Society for Testing Materials has this problem under consideration.

In 1928 Dr. Lincoln T. Work of Columbia University presented a paper before the American Society for Testing Materials on sampling of pulverized coal from air streams in which he presented data showing how wide can be the variation between samples taken from air streams by different methods.

The use of pulverized fuel has grown with astonishing rapidity. In 1924 it has been estimated a total of 16,000,000 tons was used in this country for all purposes. Now, it is probable that close to 30,000,000 tons is being applied in boiler firing alone. While for marine use the low cost of bunker fuel oil is holding back American development, the British, who have less oil, are becoming deeply interested, and it is reported that $500,000 will be used by the government next year on this development.

Rudolph Pawlikowski, at Görlitz, has made his internal-combustion engine work successfully on powdered German lignite, developing one kilowatt-hour on about 11,500 B.t.u. He expects to better this figure considerably with further improvements in the fuel and the machine. Since this engine uses only $^1/_{10}$ to $^1/_{20}$ the amount of water used for steam-turbine plants, the economic possibilities of further development in this field (power making from solid fuel) are most stimulating to the imagination.

In all of these applications of powdered fuel, higher efficiencies and smoother operation are obtained by having a maximum of uniformity in the fineness of the coal. Also to a certain extent results are better the finer the pulverization, so accurate methods of checking plant operation are needed in this respect.

Coal, when it is pulverized to pass 90 per cent through a 100-mesh screen and 75 to 80 per cent through a 200-mesh screen, is made up of very many particles that are finer even than the 200-mesh opening. Dr. L. T. Work, to whom reference was made previously, gives figures for a microanalysis of particle size in two different commercial samples of pulverized fuel, one that he calls a rather fine grade, and the other somewhat coarser than the average. C. M. Bouton and J. M. Pratt, also, in "Cooperative Coal Mining Investigations," Bulletin No. 12, Carnegie Institute of Technology, U. S. Bureau of Mines, give a similar size analysis of commercial pulverized coal.

SIZE ANALYSIS FOR TWO SAMPLES OF COMMERCIAL PULVERIZED COAL, BY DR. LINCOLN T. WORK, COLUMBIA UNIVERSITY

Microns	Coal No. 4, per cent	Coal No. 12, per cent
100.00–75.00	20.06	14.87
75.00–50.00	25.85	22.66
50.00–37.50	16.00	13.57
37.50–25.00	18.26	11.68
25.00–18.75	7.43	10.77
18.75–12.50	5.00	12.80
12.50– 9.38	3.03	5.38
9.38– 6.25	2.38	4.29
6.25– 3.12	1.23	1.94
3.12– 0.00	0.76	2.04

SIZE ANALYSIS OF PITTSBURGH COAL DUST 75 PER CENT THROUGH 200-MESH (20 GRAMS TAKEN) (BOUTON AND PRATT)

Size, microns	Amount, grams	Per cent
0–15	1.15 (approx.)	6
15–26	1.00	5
26–32	0.44	2
32–51	2.92	15
51–74	10.40	32
74–300		20
Losses	...	20

When these data are arranged in somewhat different groupings of the various particle sizes, they appear as follows:

Microns	Inches	Work I	Work II	Bouton and Pratt
Over 75	Over 0.003 (200-mesh)	20.1%	14.9%	25.0%
25–75	0.001–0.003	60.1%	47.9%	58.8%
12.5–25	0.0005–0.0010	12.4%	23.6%	8.7%
Under 12.5	Under 0.0005	7.4%	13.6%	7.5%
		100.0%	100.0%	100.0%

It will be seen from the preceding that in some commercial powdered coal as fired, from 15 to 35 per cent of the particles are smaller than $^1/_3$ of the opening in a 200-mesh screen.

This has an important bearing on the sampling. The velocity in a stream of air in a pipe varies over its cross-section, and the density of the coal and air mixture will correspondingly vary over the cross-section. Friction of the wall retards the flow to some extent. The larger particles tend to hug the walls. It may be seen from these considerations how difficult a matter it is to obtain a representative sample as to fineness from a moving air stream.

Samples, of course, can be taken from other parts of the system than from the air stream. Henry Kreisinger, in his well-known tests made in 1924 of the Oneida Street Station of the Milwaukee Electric Railway and Light Company, took samples from the feeder bin, dipping out about one pint from every 800-lb. weighing hopper added to the bin.

There are three main systems in use for firing powdered coal, which are as follows:

(1) The unit system, in which a pulverizing mill stands near the furnace and sends by means of a direct-connected blower the powdered fuel directly into the furnace without intervening storage.

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[1] Consulting Chemist and Fuel Engineer.

Presented at the Third National Fuels Meeting, Philadelphia, Pa., Oct. 7 to 10, 1929, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

(2) The storage system of the indirect type, where the coal is pulverized some distance away from the furnaces, carried by fans through pipes to storage bins, and then drawn off from the storage bins, delivered to feeders, intimately mixed with air, and blown through the burners.

(3) The storage system of the so-called direct type, where the coal is pulverized and stored in bins, from which it is drawn off and conveyed by means of a fan through pipes as a combustible mixture with air.

In the storage systems, the pulverized coal may be sampled from the storage bins, the conveyors, or the pipes delivering to the bins, or from the air stream in pipes between pulverizer and cyclone separators, or between storage bin and burner. It is best, in sampling from pipes, to use a vertical pipe. In the unit system it can be sampled only in the pipe leading from blower to burner.

The author is indebted to Max Hecht, Chief Chemist, Duquesne Light Company, Pittsburgh, Pa., for the following detailed description of his company's procedure for obtaining pulverized-coal samples for fineness tests from the Lopulco system—this is an indirect storage system:

"*Sampling Location.* An opening is available in a vertical duct, connecting the bottom of the cyclone separator with the storage bins or screw conveyors. A quart Mason jar is inserted in the opening, and moved across the cross-section of the duct, as the pulverized coal discharges from the separator.

"*Collection of Sample Aggregates.* Ten-gallon milk cans with tight-fitting covers are used for holding the aggregate collections. The sampling interval is five minutes for tests of two and three hours' duration and routine samples. The sampling interval is lengthened to 10, 15, or 20 min. when the test runs are correspondingly longer.

"*Mixing of Gross Sample.* When the can is approximately two-thirds filled, the lid is fastened tight, and mixing is accomplished as follows: The can is rolled back and forth five times and is then up-ended on its bottom. This is repeated and the can then up-ended on the cover. This is also repeated. This cycle is repeated 20 times. The sample can is emptied on a sampling cloth, and the coal is mixed by rolling the cloth.

"*Preparing Laboratory Gross Sample.* The amount of coal is reduced by passing it over a large riffle sampler. When the final sample volume is reduced to 10 lb., the coal is spread on canvas, and Mason jars are filled in duplicate by taking selected portions of the flattened pile.

"*Laboratory Preparation.* The sample is air-dried and then well mixed by rolling, and five 50-gram samples are selected from various portions of the flattened pile. The screens, after placing 50 grams each thereon, are nested with receivers. U. S. S. screens Nos. 16, 30, 50, 100, and 200 are used with a mechanical shaker. After an initial shaking of fifteen minutes, the coal remaining on the No. 200-mesh screen is weighed. The screens are again assembled in the shaker, and after a five-minute shaking, the weight is again checked. The coal remaining on each screen is then weighed. Results are reported as passing No. 16, 30, 50, 100, and 200 U. S. S. sieves."

At the Detroit Edison Company's Trenton Channel plant, which is also an indirect storage system, several different methods and locations for sampling have been tried. The author is indebted to W. A. Carter, technical engineer of power plants, and V. S. Barretta, boiler room engineer at Trenton Channel, for the following information:

It has been found most satisfactory to take samples by means of a "thief" sampler of a double-barreled construction arranged so that the open slot can be closed before taking from the coal stream. There has been some difficulty with the corroding of the cylinders with coal, but they are now making up a new sampler designed to overcome this trouble.

This is used in the coal-discharge pipe from the cyclone separators. As long as the cylinders remained clean, the results were consistent, varying only 3 per cent in fineness as against 15 to 20 per cent obtained when using other methods. A previous method was to insert a quart jar into the coal-discharge pipe from the cyclone at different points, but this proved unsatisfactory because the duct had to be left open to the atmosphere, which created a disturbing velocity condition different from that existing when the duct was closed.

Fairly satisfactory samples were obtained by taking portions periodically from the bunker, using a cup on the end of a pole, but this could not show the results obtained with each mill.

Sampling from the coal-and-air stream in the feeder pipe above the burners has also been tried at Trenton Channel, using a small pipe with the nozzle pointed in the direction of the coal stream. Results were unsatisfactory with this method due to large variations depending upon the position of the nozzle in the coal stream.

As was brought out in an earlier paragraph, the difficulties in sampling from an air stream in a pipe arise from the varying velocity, the varying density of coal, and the varying fineness of coal over the cross-section of the pipe. The varying velocity of the stream at different points causes not only a variation in the quantity of coal in suspension, but also a variation in the fineness of the particles at different points. When a sampling tube is inserted, the stream must be withdrawn through the sampler tube at the same velocity as that of the main stream in the duct at that point.

These factors have been carefully worked out by Prof. P. A. Willis of the University of Missouri and are explained by him in an article on "Accurate Sampling From Unit Pulverizers," in *Power*, December 18, 1928, pp. 1003–4. Professor Willis has worked out a method which he reports as satisfactory for taking samples from the coal-and-air stream in the duct leading from the pulverizer to the burners, 4 ft. above the fan. This duct was rectangular in cross-section, 18 in. × 23 in. In his preliminary study, he made a traverse of the area by dividing it into 12 equal squares, and taking a sample from the center of each for one minute, making these up into a composite. He used for this a specially prepared ell of $^1/_2$-in. pipe, and found that by having an opening in the end of this measuring $^{21}/_{32}$ in. he secured a velocity condition such as to withdraw in his composite sample the same amount of coal per unit area per unit time as passed through the entire duct. In other words, he withdrew a proportionate quantity, and this indicated an equal velocity condition in his sampler tube.

He then found that, by placing the opening of his sampler at a point $7^1/_2$ in. from one end on the center line of the duct, he obtained a sample of the same fineness test as that of the composite. He has not proved by this, however, that his composite sample truly represents the fineness of the whole coal stream. The degree of fineness and the density of the stream both vary from point to point in the cross-section, and it is incorrect to composite equal portions from every point.

This method of Professor Willis is being used daily in a large power plant on six large unit pulverizers, with results that are said to be entirely satisfactory. Each sample is taken during a period of five minutes. The precaution as to velocity in the sampling tube is especially important when the pulverizer parts have become worn and their performance is more or less irregular. It is necessary to have a different opening in the sampler pipe for each different operating rating of the pulverizer. If the rating is changed, the velocity in the duct changes, and therefore the velocity in the sampler tube must be changed to conform with it.

It appears to the author that both Professor Willis' method and the Tentative Standard Method of the American Society for Testing Materials have some questionable features. The A.S.T.M. method provides that the sampling tube (made of a thin-walled brass elbow) shall be moved through a stuffing-box "uniformly back and forth across the diameter of the pipe." It would seem from the considerations outlined that an error is introduced by withdrawing the sample uniformly, that is, for equal periods of time, at the various points in the cross-section.

On account of the varying density of the coal stream and the varying velocity in the pipe, it is hardly correct to withdraw the sample for equal periods at different points. Professor Willis draws from only one point, selected so as to conform to a composite, which may not be right. L. T. Work, in his paper previously referred to (American Society for Testing Materials, Part 2, 1928, Technical Papers, page 815) made a comparison of results obtained with the A.S.T.M. type of sampler and a certain type of "fixed sampler," and found more satisfactory results by use of the former. The fixed sampler consisted of a cross of pipe, placed in the duct, and having small holes bored at intervals facing the coal stream. Dr. Work's comparison, however, does not prove that the A.S.T.M. method gives accurate results, and it is undoubtedly true that further experimental study is needed to obtain a large amount of data on which to base an improved sampling method.

The following precautions are essential for securing good samples:

1 The sample should be taken from a vertical pipe, if taken from the air-coal stream
2 The sample should be taken at least 10 ft. away from an elbow, branch, or any other irregularity in the line
3 The sample should be taken as far away as possible from the disturbing action of fans or blowers
4 A stuffing box should be used through which can be moved freely a sampling pipe with right-angled bend at its inner end, facing the air stream, and without irregularities on its inner surface. This tube should be moved systematically to different positions in the cross-section of the pipe so as to correspond in its time schedule with the known variations in density of the coal-and-air mixture
5 Samples should be taken at a number of different places in the piping system, and be mixed in order to obtain an average.

Automatic or fixed samplers consisting of, say, a 1-in. pipe or a spider of crossed pipes inserted into the air stream, and bearing openings that face toward the stream, are not usually accurate samplers. The size of the openings and the placing of the openings must be chosen with care so as to take out the coal in relative proportion from different parts of the stream, as the density of the stream varies. The openings quickly become more or less clogged and thus interfere with accuracy.

Dr. Work mentions a sampler of this type used in a large power plant, which left a considerable part of the cross-section of the stream near the walls of the pipe unsampled. This part of the cross-section was the one that carried the larger proportion of coarse sizes.

On the other hand, when a sampler like the A.S.T.M. Tentative Standard is used, the results will depend in a considerable measure upon the personal equation. The operator has to move the tube in and out in accordance with a specified schedule of time intervals, leaving it for a definite period at each point in the cross-section. The accuracy will depend on how well he sticks to this prearranged schedule and how uniformly he maintains it at the different times and places of sampling.

The accuracy of sampling, as a general rule, can always be increased by taking a larger number of increments at a larger number of points in the system, combining these increments, mixing, and quartering down.

The American Society for Testing Materials, Committee D-5, which is studying these problems would welcome cooperation from committees of the American Society of Mechanical Engineers engaged in similar work. They would like especially to have reports of sampling methods now being used by plant operators and reports of any tests made upon them as showing their accuracy. The matter is one of large importance as affecting the determination of pulverizing-mill performance, pulverizing characteristics of different coals, and efficiency of performance of the furnace or burner.

Discussion

LINCOLN T. WORK.[2] In this paper Dr. Porter has discussed the problems of sampling in both the storage and the direct-fired systems. His own presentation, as well as other literature data, make it evident that the problem of greater moment lies in the sampling of powdered coal from air streams, as in the case of the direct-fired system. With reasonable precautions and with but slight adaptation of equipment, coal may be sampled accurately when prepared for firing from storage bins.

The writer's paper to which reference has been made dealt with the sampling of coal from air streams. On analysis of the principles, the following were shown to be the requirements for accurate sampling: (1) Minimize the effect of separation of coarse and fine coal. (2) Withdraw the stream at exactly the velocity which normally prevails at each point of sampling.

The reasons for minimizing the effect of segregation of sizes are apparent. With due care these effects can be avoided. The sampler should be placed at a point in the line where segregation is at a minimum. The first three precautions cited near the end of Dr. Porter's paper are sound means for doing this. However, it is seldom if ever possible to reach that objective in view of the present design of conveying equipment. Hence, small samples are taken at many points in the cross-section, and a composite of these is tested. The last two precautions cited in Dr. Porter's paper bear on this. Procedures differ, but none can be called final until it is possible to control the velocity of the sampling stream at each point. Good field evidence can be secured by sampling one cross-section of the pipe and then another after a few bends. If these samples are of similar fineness, the procedure may be considered acceptable.

The control of velocity into the sampling tube is much more difficult, and yet it is vital to the accuracy of the sample. The coarse particles moving in the line of the sampler are projected into the opening while the fines are diverted with the air stream into the sampler if the velocity in the tube is the greater, and away from the sampler if the velocity in the pipe is the greater. Static and velocity pressures, and resistance to flow in both burner and sampling circuits control this distribution. These problems have not been solved with even the best sampling devices in common use at the present time, and the limitations imposed thereby must be recognized.

Dr. Porter has discussed the proposal of Professor Willis for the simplification of sampling methods with pulverized coal in air streams. It must be recognized that a sample taken at one point may give the true percentage through one sieve, say 200 mesh, but that the sample so taken may give erroneous values in the coarsest sizes, and may be entirely unfit for testing in the sub-sieve sizes.

[2] Department of Chemical Engineering, Columbia University, New York, N. Y.

H. P. Reid.[3] In the first part of this paper the author refers to size analysis of pulverized-coal samples as determined by several different investigators. It would be interesting to know the methods used for controlling the fineness when grinding these samples. One would not expect the same general relation between the undersizes in coals pulverized by roll mills having screen fineness control as might obtain in coals pulverized by similar mills using air-separator methods for fineness control; nor again as might obtain in coals pulverized by ball or tube mills which discharged over a partition, and in which the fineness was controlled entirely by the rate of feed. It would seem that the sample taken would be much more representative when there was but small variation in sizes or in other words when there were comparatively few of the undersizes.

In obtaining a sample from an air-coal stream the paper states that this sample should be taken (1) from a vertical pipe, and (2) that it should be taken at least 10 ft. away from an elbow, branch or any other irregularity in the line. Often the set up is such that one can find but very little if any vertical pipe carrying this air-coal stream. Is it to be inferred that representative samples cannot be taken from horizontal pipe sections? Or is it possible to maintain sufficient velocities to carry the pulverized coal in such a way that representative samples may be taken from horizontal pipes. Again might it not be possible to insert a venturi throat in the pipe, using this both to mix the coal of differing sizes through the stream and also to increase the stream velocity so that the mix after the venturi restriction would be quite regular from which to sample. Does not the author mean ten diameters of pipe distant rather than 10 ft. distant from elbows, etc.?

When sampling pulverized materials from hoppers or from flowing streams of the pulverized material, is it not necessary to guard against segregation of particle sizes due to the rolling action of the larger particles? We have all seen how in piling coarse coal, for instance, the coarser particles tend to ride over the smaller ones, and slide down to the sides of the pile. If this is true with the visible particle sizes, may it not also be true with the smaller sizes.

In sampling from the discharge air-coal stream from a unit pulverizer will there not be a difference in the particle sizes at any given point in the stream due to fluctuations in air volumes passing the mill? And will there not be decided fluctuations in these volumes from time to time as effected by temperature, moisture content, and amount of coal load in the mill?

A. R. Mumford.[4] The sampling of pulverized fuel in power plants is not required to be of the utmost accuracy unless other than regular operation is the purpose. Coal is charged against a plant generally by checked railroad weights, and in order to determine how much coal is charged, it is necessary to sample the fuel as it enters the plant, and in every instance within the writer's acquaintance such a sample would cover raw coal only.

The sampling of pulverized fuel is necessary for operating information, and to follow in a general way the performance of mills and boilers. During technical studies of operation it may be necessary to obtain samples of pulverized fuel which accurately represent the total fuel used, but normally if the samples are relative so that variations from day to day can be determined, the results are sufficient for the purpose. The writer finds that variations of several points in the amount of pulverized coal passing through a 200-mesh screen does not appreciably effect the plant operation or efficiency, and he believes that this finding has been confirmed by others who have investigated the effect of normal variations in the sizing of pulverized coal. The writer gets the impression from Mr. Porter's paper that extreme accuracy is the aim of his committee's work, but as far as power-plant work is concerned extreme accuracy does not seem at present to be warranted by improvements in efficiency.

Sampling systems have been designed which would undoubtedly give a very representative sample of pulverized fuel, but the building volume necessary to house the sample machinery and the cost of the equipment does not seem to be justified over the somewhat less accurate but cheap method of relative samples.

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[4] New York Steam Corp., New York, N. Y. Mem. A.S.M.E.

Author's Closure

The discussion by Dr. Work brings out clearly the principles governing sampling from moving air streams. His statement, however, that "the sampler should be placed at a point in the line where segregation is at a minimum" leaves a little to be desired from the practical standpoint as to just where this point is best located. There is some segregation of sizes always in a moving stream of air and coal, and hence the precaution that a number of samples should be taken at different points in the conveying system, and compounded.

Mr. Reid's discussion brings out several interesting points. He may be right as to placing the sampler beyond bends, etc., a distance depending on the pipe diameter, but the author is inclined to hold still that a minimum of 10 ft. is advisable. He asks for more data as to how the coals, whose fineness is quoted as examples of common practice, were pulverized in the different cases. The author cannot give this detail, except that the samples were obtained from different plants. He asks whether correct samples cannot be taken from horizontal pipes. This would be difficult, but probably is not impossible if some such method of agitation as he suggests is made use of. It is to be hoped that he will experiment along this line.

It is necessary, as Mr. Reid suggests, when sampling pulverized coal out of a bin or from a flowing stream of the coal, to guard against segregation of sizes due to rolling action of the larger particles. This action, however, is not as great in powdered coal as in the coarser grades, and its effect can very largely be eliminated by taking a number of small portions from different parts of the bin or moving mass. His point about the effect of fluctuations in air volumes in a unit pulverizer is well taken, and endeavor should be made to take samples for comparative purposes under like conditions. Mention is made in the paper of the need for taking into account the capacity rating at which the pulverizer is running.

In answer to Mr. Mumford's question as to the need for any such accuracy as the Testing Materials Society is working for, it is sufficient to say that experiments have shown a variation between two samples taken by different but common methods amounting to 100 per cent. In other words, the percentage held on a 100-mesh screen in one case was twice as great as it was in the other case sampled by a different method. It may be all right for some plants to use rough methods for their own purposes, that is, to get relative conditions from day to day, but unless, in these methods, the place of sampling, the operating rating, the velocity of coal stream and of sampling stream, the choice of sampling point in the cross-section, and certain other conditions are kept uniform or allowed for in reasonable measure, there will be so large a variation in the results that it were better not to attempt any sampling at all.

FSP-52-17

Use of Fuel in the Manufacture of Portland Cement

By H. P. REID,[1] CHICAGO, ILL.

Heat is required in the manufacture of Portland cement for evaporating the water from the coal and from the raw materials used in making the cement; for preheating these materials to temperatures necessary for calcination; for breaking up the carbonates and oxidizing the undesirable elements, principally organic matter, from the raw materials; for further raising the temperature of the material to the point necessary to effect the desired chemical reactions; and for producing the power required for mechanical operation. All of these requirements except that of removing the moisture and the production of the power are met inside the cement kiln, and are commonly included in the term "burning." Furthermore, with the wet process the moisture is evaporated in the kiln also.

THE USES of heat in the manufacture of Portland cement can be classified as for (1) drying, (2) burning, and (3) power. For the generation of this heat fuel is required. In some cases separate fires are maintained for each of these three requirements, while under other conditions power is generated from the waste heat in the gases leaving the kiln. Generally speaking, of the fuel used for drying and burning 10 per cent to 25 per cent is required for the former and 75 per cent to 90 per cent for the latter. In all dry-process plants and in many of those of the wet process, sufficient heat may be reclaimed from the waste kiln gases to generate the necessary power required to operate the mechanical equipment. As this power is generated from heat produced by fuel burned in the kiln, it is proper to credit the "burning" with the fuel equivalent of the power generated. Such an adjustment would charge the power with from 25 per cent to 35 per cent of the total plant fuel, "burning" with 45 per cent to 60 per cent, and drying as given. These percentages are to be considered as representative of the cement industry as a whole, and not necessarily actual for any definite plant.

Natural gas, producer gas, blast-furnace gas, oil, and pulverized bituminous coal are usable as fuel in the Portland-cement industry. According to a bulletin of the U. S. Bureau of Mines recently published, of the cement produced in 1927 in this country, over 80 per cent was from plants using coal as fuel. The selection of the kind of fuel is made on the basis of cost of heat units delivered at the burner. It is therefore logical to expect to find oil and natural gas used in such places as are distant from suitable coal supply but near the oil and gas wells. Blast-furnace gas is used where the cement mill is operated in connection with blast furnaces, and a surplus of gas is available. Referring again to the same bulletin of the Bureau of Mines, the average fuel consumption in 1927, per barrel of cement produced with the various fuels, was with coal 135.8 lb., with oil 0.2743 bbl., and with natural gas 2450.5 cu. ft.

Practically any bituminous coal when prepared properly can be used as fuel in the cement kiln. Bin systems rather than unit mills are commonly used, and therefore the coal is usually dried before pulverizing. There are, however, a number of unit mills successfully operating at installations made in the past few years.

The cement kiln considered as a furnace is radically different from the pulverized-coal fired boiler furnace, and because of this difference it has been found necessary to pulverize the coal finer than is the usual power-plant practice. Uniformity of fineness is essential, and the kiln must have reasonable guarantees against shut down, even for short periods, to insure uniform quality of the product, and to preserve the life of the refractory lining. To insure even fires the coal as fired should be of nearly uniform moisture content, which condition is somewhat difficult to maintain in winter when snow and ice often exist in the coal as received. The given requirements account largely for the installations of the bin system in the majority of the cement plants. On the other hand, the development of the modern unit pulverizer with its air separator to control the fineness, with the use of preheated air in the mill to absorb moisture from the coal, and the improvements in design and selection of materials of construction to reduce operating delays will undoubtedly result in greater use of this type of mill by this industry.

When using the bin system the moisture in the coal must be reduced to such an amount as will permit economical pulverizing-mill operation, and such that the prepared coal may be conveyed to the bins and be drawn from them without blocking. The permissible amount of moisture depends largely on the character of the coal and on the type of mills, transport system, and burner feeders. Where air separation is used for fineness control considerable moisture may be removed by the mill air. Especially is this true when the coal fed to the mills has been preheated by partial drying or when the quantity of air vented from the system is large. Additional mill drying may be accomplished by introducing preheated air to replace that vented from the system. A small amount of moisture is removed from the coal in the screen type of mill, but usually the air circulation through such mills is insufficient to carry away much moisture. There is, however, the problem of recovery of the fine coal from mill-vent air, and also that of maintaining the temperature of this vented air high enough above its dew point to keep the ducts clean, and to remove fire hazards from such ducts.

It is customary in the cement industry to dry Pittsburgh and West Virginia coals down to 1 per cent or 2 per cent moisture as they leave the mills, and Illinois, Indiana, and Kentucky coals to from 2 per cent to 4 per cent. These percentages are often more a matter of practice than absolute necessity, as it is much more economical to take care of the moisture of the coal in the kiln than to remove it in dryers, provided that this moisture is low enough to permit uniform feeding, and such as will not make it difficult to hold the kiln fires.

Precautions and good housekeeping are as essential in the coal-preparation buildings of these plants to insure safety as they are in similar buildings of other industries. There is, however, a coal fire hazard peculiar to the type of machines used in cement plants. The discharge end of the cement kiln, where the fuel is admitted and ignited, is open except for a shielding hood. As this hood is stationary and as the kiln rotates, there is of necessity an opening between the two. In some cases this opening is closed by a rotating seal, but more often it is not. Furthermore, openings are necessary for the admission of secondary combustion air, and

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

for the burnerman to watch his kiln operation. In starting up a cold kiln the coal is liable to burn very irregularly until the kiln walls become sufficiently heated to hold the fire; also irregularities in the coal feeders may result in more or less violent explosions within the kiln. While the force of such explosions is largely dissipated within the kiln, yet ignited fuel and hot gases are forced out of these openings, and may be dangerous for some distance from the kiln end. Pulsations in the flame are quite common in rotating kilns. These may be caused by irregular discharge of coal from the feeder, by too high a percentage of primary air, or by too strong a draft.

High ash content in the coal is probably less troublesome in the operation of a cement kiln than in most other industries. Generally speaking high ash indicates relatively low B.t.u. content, and ash purchased as coal is expensive cement raw material. As high percentages of silica and alumina are found in Portland cement, the ash of the coal may be absorbed by the clinker in the kiln. The chemist in his control must, however, consider this ash as its weight amounts to approximately a quarter of a per cent of the clinker weight for each per cent of ash in the coal.

Drying Processes Used

The dry and the wet processes are both widely used in this country for the manufacture of Portland cement. Both processes have, in general, similar operating problems. In the dry process the moisture is driven from the raw materials before they are ground, while with the wet process more water is added, and the materials are ground in a slurry form. This slurry is pumped or conveyed to the kiln feeder. In some cases part of the moisture is filtered out before entering the kiln, while in others all is fed to the kiln. Whatever moisture there may be in the kiln feed must be evaporated in the back end of the kiln by the hot gases.

With the dry process it is the usual practice to pass each of the raw materials through its separate rotary dryer. Such dryers are cylindrical tubes made from steel plates, mounted on two concentric steel tires, and rotated about their longitudinal axis. To effect material travel through these tubes, they are usually pitched $^1/_4$ in. to $^3/_4$ in. per foot of horizontal length. Some dryers have no pitch but depend on the construction of internal lifters or flights to effect the forward travel of the material.

Heat as applied to the drying material is obtained by burning fuel either in furnaces in front of the dryers or in air suspension within the dryers. It is difficult to obtain efficient combustion, and at the same time large output with these machines. If a furnace is used in order to effect complete combustion of the fuel before the gases enter the dryer, the heat transfer from the gases to the material and the moisture therein, must be largely by convection rather than radiation. Such transfer requires large drying surfaces, which results in very large units in order to obtain capacity. If, on the other hand, the fuel is burned in suspension, the cold dryer walls and the heavy concentration in the dryer atmosphere of fine material particles so chill the flame as to quench many burning fuel particles. Consequently, combustion is not completed and losses by unburned fuel run high. In order to hold a fire at all with such a system, ignition is effected in a small refractory-lined chamber whose walls are held at an incandescent heat. However, with such a flame properly directed over the material large amounts of heat are transferred to the material by radiation, and the output of such a machine is generally greater than it would be were a furnace used.

Some dryer manufacturers build a small furnace before the dryer for use with powdered coal or oil. Such a furnace is little more than a large ignition chamber for the fuel. By the time the blast of air and fuel has passed through the furnace, the fuel is well ignited and partially burned, combustion continuing in the front end of the dryer. With the use of such a furnace better combustion is obtained, and considerable radiant heat transfer occurs also. Greater fuel economy should be expected than is possible when burning entirely inside the dryer, and the capacity should be greater than when combustion is complete in the furnace.

Some raw materials have low moisture content, and are of such a nature that this moisture is readily driven off. With such the drying problem and the dryer efficiency is of relatively small importance. Other raw materials contain high moisture or are of such nature that the moisture is difficult to remove. With the latter the efficiency with which the fuel is burned may reflect heavily on the production costs.

At many of the older plants, most of which are dry process, the cost of fuel was so low when they were built that combustion efficiency was then of small importance. In the construction of such plants the dryers were usually so bound in by other mill machinery that there is not sufficient space now to rebuild with machines that would be more economical in the use of fuel.

The application of heat at high temperatures in amounts sufficient only to evaporate theoretically the moisture will not effect complete drying in most raw-material dryers. Such application of heat would result in the gases leaving the dryer not only saturated with moisture but also carrying considerable superheated steam over and above the moisture absorbed by the gases. Consequently, excess air must be added, usually in very large percentages, to the combustion gases to provide sufficient moisture-carrying vehicle. From the thermal viewpoint, such a dryer is of necessity quite inefficient. As has been pointed out, good combustion is difficult, heat losses in the stack gases are great due to the large volumes of drying vehicle necessary, and as the material must be heated up in order to evaporate the moisture, considerable heat is carried away by the dried material. Much work has been done in efforts to improve the thermal efficiency of these machines, yet it can be appreciated that for this type of work such a machine can not be made highly efficient from the thermal viewpoint. If heat otherwise lost around the kiln could be reclaimed and used for this drying considerable economies might be effected.

The Rotary Kiln and Its Operation

The rotary cement kiln used almost exclusively in the U. S. A. is a cylindrical tube fabricated from steel plates, and resembles the rotary dryer in general appearances excepting that usually it is much larger. Early kilns were made about 6 ft. in diameter by 60 ft. long, some even smaller, while modern kilns are as large as 12 ft. in diameter by 350 ft. long. To protect the steel shell from the heat, refractory lining is used. This lining is usually 6 to 9 in. thick in the front or hottest end, and 4 to 6 in. thick further back where the temperatures are less.

The prepared material, wet or dry, is fed into the back or high end of the kiln, and is moved forward by the rotation and pitch of the kiln. For this pitch $^1/_4$ in. to $^3/_4$ in. per foot of horizontal length is used. The speed at which the kiln is rotated varies widely, some kilns running as fast as 2 r.p.m. while others make but 1 revolution in $2^1/_2$ to 3 minutes. It can therefore be appreciated that the time required for material to pass through the kiln is a factor of length, pitch, diameter, and rotary speed.

Fuel in blast suspension is admitted at the front end of the kiln, and the combustion gases travel against the forward moving feed. Both the surface of the feed and the exposed lining of the kiln absorb heat from the flame and the gases. While a small amount of the heat absorbed by the refractory lining travels through the bricks and is radiated from the outer shell, most of it is transferred to the feed as the kiln rotates.

With the wet process the moisture in the slurry or in the filtered feed is evaporated in the back end of the kiln. In some kilns this back end is an open tube similar to the rest of the kiln except that metal rather than refractory lining is used. With other

kilns various devices are employed to facilitate evaporation. These devices in general use some system designed to increase the area of surface exposed to the gases. One such device employs a large number of chains so fastened to the kiln shell that they drop in loops from that portion of the shell above the kiln center, and lay on the shell below that center. The hanging loops present large metallic areas to the gases, and thus absorb heat from the latter. As the kiln rotates and the wet feed rolls over these chains, the moisture and material absorb the stored heat, and leave the chain surfaces wet as they again come in contact with the gases.

From the time the moisture is evaporated in the wet-process kiln the two processes are similar. The dry feed is gradually heated to higher and higher temperatures as contact is made with hotter and hotter gases as it travels forward until the carbonates of calcium and magnesium begin to break down, and the sulphur and organic matters ignite and burn. This calcination of lime and magnesium begins at temperatures of about 1500 deg. fahr. and is completed by the time the material has reached 1750 or 1800 deg. fahr. In completing this process an average of about 950 B.t.u. per lb. of clinker produced is absorbed by the material. There are wide variations from this average depending on the composition of the raw materials used. The material then enters a second stage of heating until its temperature is raised to approximately 2600 deg. fahr. As the higher temperatures are approached the calcium, silica, and alumina regroup themselves into complicated chemical structures. The exact temperature necessary for the foregoing reactions will depend on the fineness of the particles of material, their proportions in the mix, and also quite possibly upon the presence in small amounts of other metals. The time required for the transaction has not been determined to the satisfaction of the average operator. It is the general belief that safety for quality lies in a relatively long zone in the kiln where these higher temperatures are maintained. Forward from this hot zone there are several feet where the clinker, formed by incipient fusion of the materials under the high heat, gives off some of its heat to the incoming combustion air and fuel.

Clinker leaving the kiln will be from 1800 to 2200 deg. fahr. according to the length of this cooling zone, which varies widely in the same size kilns at different plants, and to some extent on the same kiln according to the temperature of the air entering and to the amount of draft used.

In many of the older plants the clinker as it leaves the kiln is sprayed with water to cool it sufficiently to permit the use of metallic conveyors or elevators for removing it to cooling or storage piles. Further cooling may be effected by water spray, by exposure to air, or by a combination of the two.

In plants of more recent design, mechanical coolers are used. The most popular type of these clinker coolers appears to be a rotating cylindrical shell through which atmospheric air is blown against the clinker travel. Such quantities of the air heated in cooling the clinker as are required for combustion are delivered into the kiln. Some of the air may be passed through the coal mills to assist in removing the moisture when unit mills are used. There yet remains a large volume of air containing considerable heat which must be wasted. Such air would be of value in drying coal and raw materials were it possible to transport it to the dryers without too great a heat loss.

In describing the combustion within the kiln, pulverized coal will be considered as the fuel; such description can be applied readily to other fuels by those interested in them.

The pulverized coal is blown into the kiln with blast or primary combustion air at such a velocity as to insure its being carried by the gas streams until combustion has been completed. This velocity may vary according to the open inside diameter of the kiln, the fineness of the coal and the draft used. While many attempts have been made to use improved burners which would mix the coal and air intimately, and produce gas turbulence for scrubbing the films from the burning particles, most such efforts have not proved a success, and the open pipe is almost exclusively used as a burner with the rotary kiln.

The fuel is ignited, combustion takes place, and the gases deliver their heat to the material all within a straight, nearly horizontal rotating tube, and the burning fuel can not be allowed to impinge on the refractory. Therefore, the time required for complete combustion of a given size particle must be much longer than would be necessary were turbulence of the gas stream possible. It is practically impossible to maintain refractories with their surfaces directly exposed to the high temperatures of the flame, and at the same time to the erosive action of the clinkering materials rolling over these surfaces. No attempt is made to burn on the exposed lining, but it is coated in the hotter zones by material fused to it. As the material mass traveling through the kiln is heated to a state of incipient fusion it will adhere readily to surfaces which are slightly cooler than it. To enable the refractory to present and maintain this cooler surface to the semi-fused material, heat transfer must take place outward through the refractory to the shell and atmosphere. The rate of this transfer is governed largely by the thickness and nature of the refractory. If too thick the heat flow is too slow, and the face of the refractory will burn away until the heat equilibrium is established, and the protecting coating fuses to it. If at any subsequent time the fire becomes too intense or localized, this coating will be melted away. The ease with which such a coating may be formed and held is largely a function of the raw materials used for the cement, and also of the skill of the burnerman.

Sufficient heat must be generated to raise the temperature of the material to, and maintain it at approximately 2600 deg. fahr. As the chemical reactions take place a small amount of heat of reaction is liberated at the higher temperatures. The temperatures in this hot zone must be held within the limit of the life of the fused coating or the refractory will be ruined in a few hours. Normally such a refractory lining should last from nine months to two years.

With turbulent or short-flame burners there is not sufficient radiant-heat absorbing material around their short combustion zone to cool the flame to safe temperatures. With the open-pipe burners, because of the lack of turbulence, the combustion is sufficiently slow to permit enough heat transfer to keep the temperature of the flame within safe limits. Also this slow-burning flame may be made so long that a safe length of clinkering zone can be maintained readily for the proper chemical reactions. The length and position of the flame with a given installation may be altered over a wide range by the fineness of coal pulverization, by percentage and temperatures of primary air, by temperature of secondary air, and by the draft on the kiln. The operator usually has control of the rate of coal feed, the volume of primary air, and the draft.

Very wide ranges of the percentages of primary-combustion air are used in the industry. Some installations use as little as 20 per cent while others use more than 50 per cent. Generally speaking, high percentages of primary-combustion air tend toward short flame and short refractory life. The permissible percentages of this primary air depend largely upon the characteristics of the coal used.

While preheating of the primary and secondary air can be used with economy in many installations, it should be introduced with care on existing equipment as it tends likewise to shorten the flame and thus create excessive zone temperatures. Because the fuel must be burned in suspension in a long horizontal gas stream, the necessity of fine pulverization of the coal is evident. The exact value of a coal particle which drops out of the gas stream

into the material traveling through the kiln is unknown. However, as the heat requirements of the kiln are the maintainence of certain temperatures, it may be that the B.t.u. of such coal particles are not as effective as when they are dissipated at higher temperatures found in the flame. Certainly delayed combustion of coal particles which remain in the gas stream is undesirable, as under such conditions their heat is liberated in that portion of the kiln where an excess amount of heat already exists.

Balling of the pulverized-coal particles is probably much more prevalent with the bin system than is commonly appreciated. After having expended power to grind the coal to a given fineness, more attention should be given to the conveying and mixing equipment to see that the full advantages of the fine grinding are obtained, and not largely counteracted by feeding the coal to the burner in lumps or balls. Uniformly mixing this fine coal with the primary air should also be accomplished. It may be possible that by using proper precautions against balling, by maintaining proper air and coal mix, and then with the use of preheated air obtained from the clinker, economies might be obtained in coal preparation, as these features should overcome to a certain extent the drawbacks of coarser coal and higher moisture content.

Heat-insulating materials can be used between the refractory lining and the steel shell in that portion of the kiln from the feed end forward to within a few feet of the clinkering zone. However, there are no refractories economically available that would stand the clinkering-zone temperatures were heat-insulating materials placed behind them. As the refractories are held in place within the steel shell solely by the arch action of the complete circle, insulation must be placed with extreme care behind the brick of this circle. To reduce relative motion between the brick and insulation, and to minimize the crushing effect of the load on the insulation, key brick are placed against the shell at frequent intervals.

While different brands of Portland cement may vary in calcium and magnesium content depending upon the raw materials used, a cement having a CaO equivalent of 65 per cent and a MgO equivalent of $2^1/_4$ per cent may be said to be representative. Assuming that all this lime and magnesium existed in the raw materials as carbonates there would be about 195 lb. of CO_2 liberated in calcining sufficient of these materials for one barrel of cement. For burning such a cement approximately 100 lb. of coal would be used in the kiln with the dry process. This coal should produce about 275 lb. of CO_2, making a total of 470 lb. of CO_2 per barrel of clinker produced or 4.7 lb. of CO_2 per pound of coal burned. Because of the slight turbulence in the gas stream as it flows, it is difficult to maintain 0 per cent CO with less than 2 per cent of O_2 in the gases leaving the kiln. To insure proper oxidation of the undesirables in the kiln feed, this amount of O_2 which corresponds roughly to 11 per cent excess air is probably necessary. However, to maintain a definite percentage of CO_2 as indicated from the foregoing is impossible with many raw materials because the lime is not always in the form of carbonates, but may exist as other calcium compounds, thus varying the amount of CO_2 evolved. Also the quantity of coal burned per barrel of clinker is not constant, and very slight changes in the draft radically change the amount of induced air at the front end. Again, the thickness of the coating on the refractory and consequently the free-gas area is variable, thus altering the resistance to gas flow through the kiln. It is therefore evident that a constant draft on the kiln will not result in uniform operation.

Control meters on the gas content, whether CO_2 or CO, have not proved a definite aid in control of the kiln. Pyrometers and draft gages when properly located and interpreted may give some valuable information to the operator. However, because of the adhesive properties of the fine particles in the gases, all such instruments require considerable attention to function properly.

Waste Heat Power Generation

The temperature of the gases leaving a dry-process kiln will be from 1000 to 1700 deg. fahr., depending largely on the length of the kiln; 1400 deg. fahr. seems to be accepted as a fair average temperature. With the wet kiln this temperature is much lower, being anywhere from 550 to 1100 deg. fahr. The longer the kiln of either type the greater the heat transfer to the feed, and consequently the lower the exit-gas temperatures. With the modern type of kilns one would expect exit temperatures of around 1000 deg. fahr. from a dry kiln, and 500 deg. fahr. from a wet kiln.

With the average dry-process kiln there is recoverable heat in these exit gases sufficient to generate power to operate the entire plant, and in some instances to generate a surplus for other purposes. With the modern wet-process kiln the amount of heat possible to recover from the waste gases may be so small that waste-heat boilers would not give sufficient return on the necessary investment to justify their installation.

FSP-52-18

Recent Developments in Stoker Design

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In this paper recent developments in stokers are discussed. The presentation covers all stokers manufactured by the firms who are members of the Stoker Manufacturers' Association. Stokers for the larger size of steam-power boilers have undergone a different development from the smaller stokers. There has been a distinct advance in size, in practical combustion rates, and in the convenient arrangement of stokers in general. The problems of preheated air and the development of small stokers for domestic and industrial uses are also presented.

EACH manufacturer of stokers has made improvements according to his conception of the art of stoker firing and the requirements of the market. Research and test work has been undertaken in diligent effort to improve not only the design of stokers themselves, but also the conditions under which they will render the most satisfactory service. In addition to this work on stokers themselves, the problems of furnace design have been the subject of an immense amount of study.

General Development

The development of stokers has taken place along two radically different lines, in each case synchronizing with a positive commercial demand. Stokers adaptable to large steam-generating units have developed principally along the line of size; those of small boilers principally along the line of convenience.

All are aware of the well-defined trend toward steam-generating units of larger and larger size. It has been found economical to build steam-turbine generators in sizes of 150,000 kw. and 200,000 kw. The same factors of economy are responsible for a demand for steam-generating units, ranging in their larger sizes from 500,000 to 1,000,000 lb. of steam per hr.

Pulverized-fuel firing has proved somewhat more popular for the bulk of these very large modern steam-generating units. It has been comparatively easy to expand the combustion principles, and multiply the mechanical apparatus of pulverized-fuel systems. Mechanical stokers have also been increasing rapidly in size, and it is quite certain that during their recent growth no obstacle so far encountered has indicated any specific limitation, either mechanical or economical.

This growth in the size of stokers is perhaps more remarkable when the mechanical difficulties that confront the designer are fully comprehended. In either the multiple-retort or the chain-grate stoker, a material increase in size may render its former structural formation entirely impractical. The larger stokers may require almost 100 per cent new design in order to secure the necessary strength of structure, resistance to heat, and the feeding and distributing of the fuel and air.

An interesting feature of this development work in large stokers is the fact that both stoker manufacturer and customer join in the responsibilities that are involved. Models are out of the question, and a newly designed larger stoker must make good as a practical operating machine, and must meet its predicted performance tests in the customer's plant, without the benefit of preliminary trial installation.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The fact that such stokers practically always equal or better the expectations of their sponsors is a splendid tribute to the accurate knowledge of the art of combustion shared in general by both stoker and power-plant engineers. There is, furthermore, a certain elasticity in the underlying principles of stoker firing which accounts for the success attained by various stoker manufacturers in working out their particular development

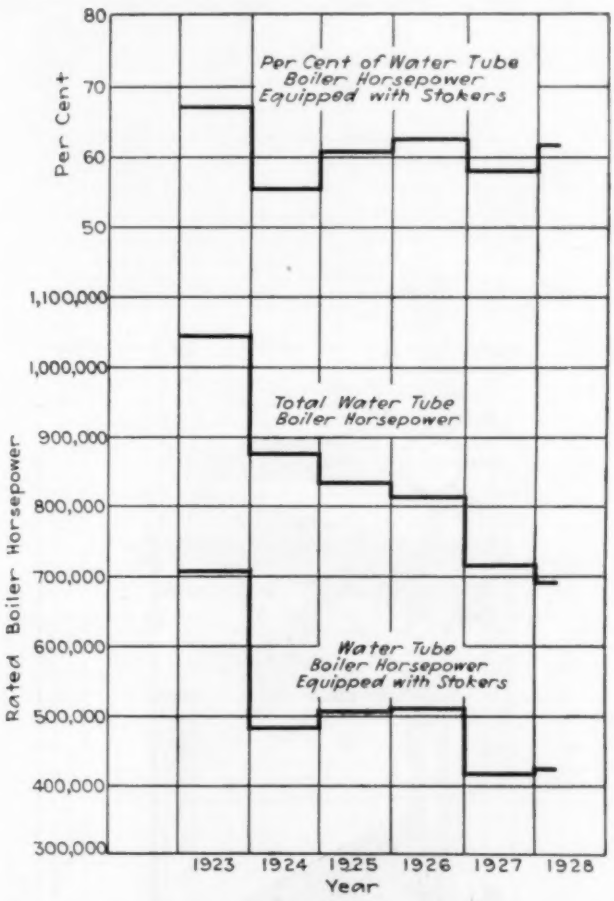


Fig. 1 Chart Showing Department of Commerce Figures on Total Horsepower of Water-Tube Boilers, Water-Tube Boilers Stokered Each Year and Percentage of Total Horsepower of Boilers Represented by the Horsepower Stokered

programs along lines which are not always parallel, but along which each manufacturer attains a definite measure of success.

Development of Size

From a production of, say, 100,000 lb. of steam per hr., stoker installations have gradually increased in size till today the production of 400,000 lb. per hr. is both feasible and economical. For one installation 500,000 lb. of steam per hr. is expected. The projected areas of stokers required for such steam production are of the order of 700 sq. ft., and a number have been or are being built of approximately this size.

In the multiple-retort type, single-end stokers ranging up to 713 sq. ft. of projected area and double-end stokers up to 716 sq. ft. have been or are being built. A steam production of the

order of 400,000 to 500,000 lb. is contemplated from these installations.

The chain-grate type of stoker has developed from the 420 sq. ft. of grate surface, furnished at Crawford Avenue some years ago by several manufacturers, to 528 sq. ft. in one installation and 684 sq. ft. in another, both in the United States, with 700 sq. ft. the maximum in England. The contemplated steam production with these stokers is appreciably lower than with the multiple-retort underfeeds, due chiefly to the application of the chain-grate stokers to the burning of low-grade fuels.

Considerably large stokers can be built, and modern high-combustion rates can be consistently maintained upon them.



FIG. 3 TAYLOR MULTIPLE-RETORT STOKER AT DELRAY STATION OF DETROIT EDISON COMPANY—THIS STOKER IS 15 RETORTS WIDE, 57 TUYÈRES IN LENGTH, AND HAS AN ESTIMATED COAL-BURNING CAPACITY FOR PRODUCING 500,000 LB. OF STEAM PER HOUR

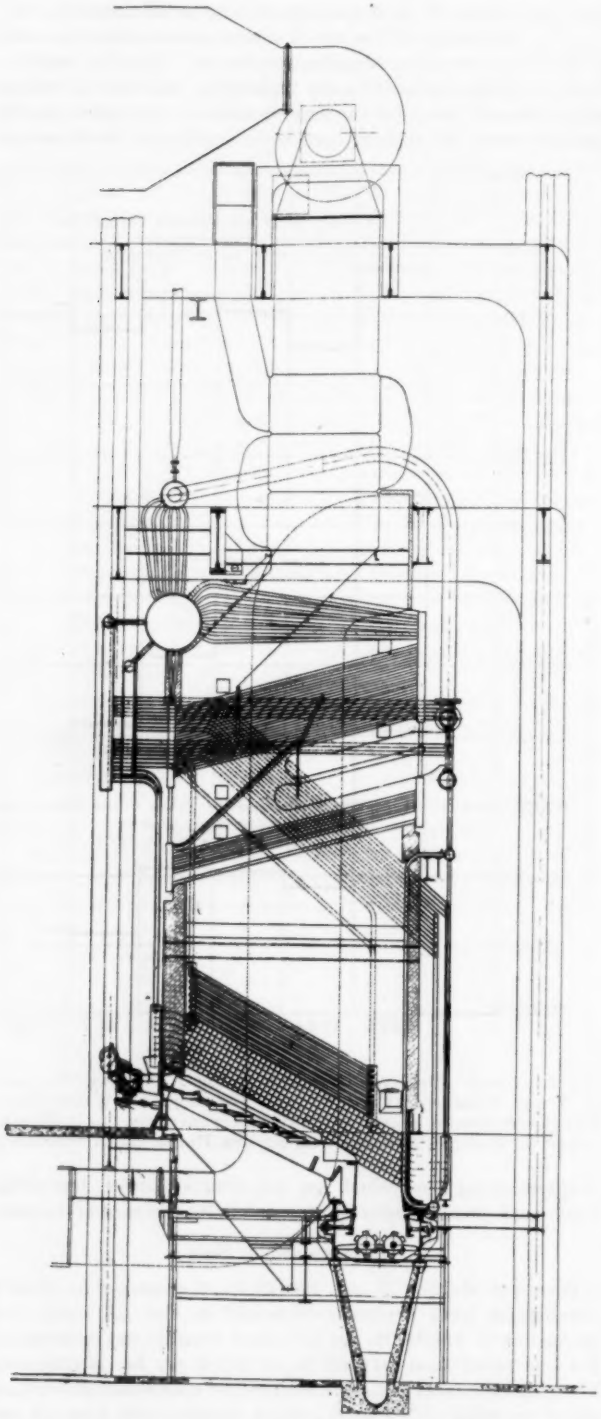


FIG. 2 TAYLOR MULTIPLE-RETORT STOKER INSTALLED AT THE WESTKRAFT PLANT IN BERLIN UNDER 2580 HP. BORSIG BOILERS—STOKERS ARE 20 RETORTS WIDE, 49 TUYÈRES IN LENGTH, HAVE A PROJECTED AREA OF 713 SQ. FT. EACH, AND ARE DESIGNED FOR 500 DEG. FAHR. AIR TEMPERATURE

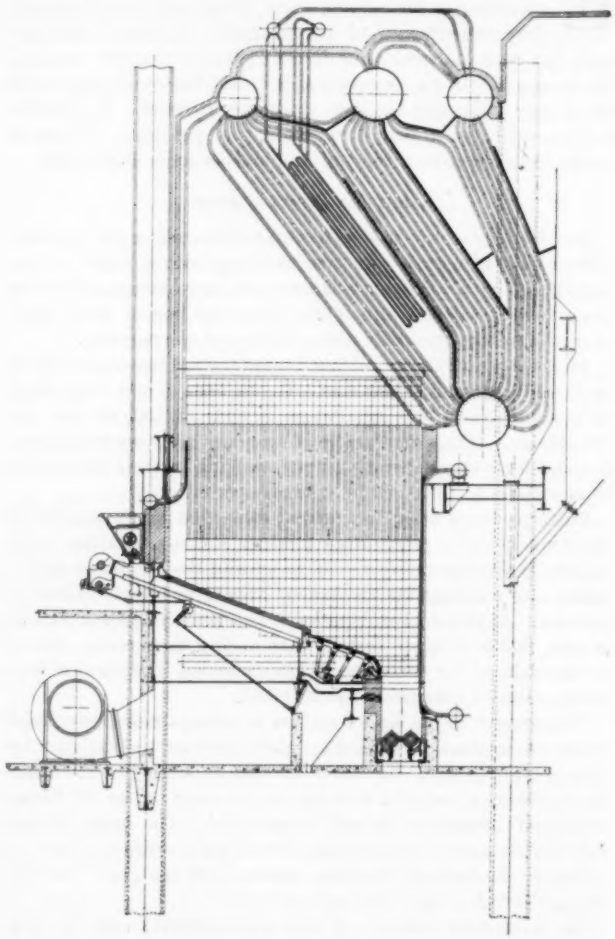


FIG. 4 WESTINGHOUSE MULTIPLE-RETORT STOKER FOR REED STATION OF DUQUESNE LIGHT CO., PITTSBURGH, UNDER 3180-HP. BOILER—STOKERS ARE 18 RETORTS WIDE, 33 TUYÈRES LONG, WITH LINK GRATES AT THE LOWER END OF RETORTS MAKING A LENGTH EQUIVALENT TO 45 TUYÈRES

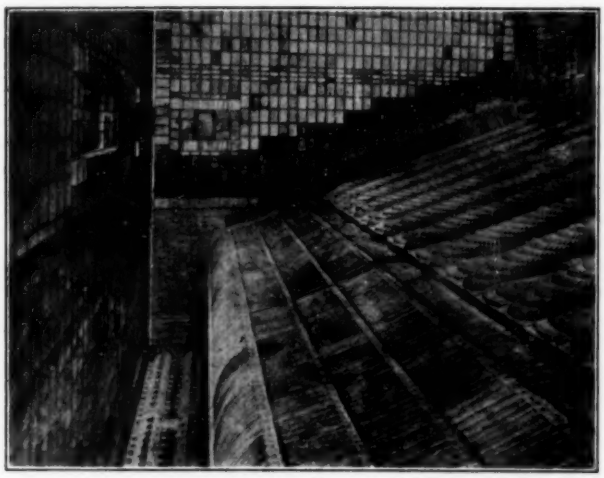


FIG. 5 INTERIOR FURNACE VIEW OF WESTINGHOUSE MULTIPLE-RETORT STOKER AT HUDSON AVENUE STATION OF BROOKLYN EDISON COMPANY UNDER 2380-HP. BOILER

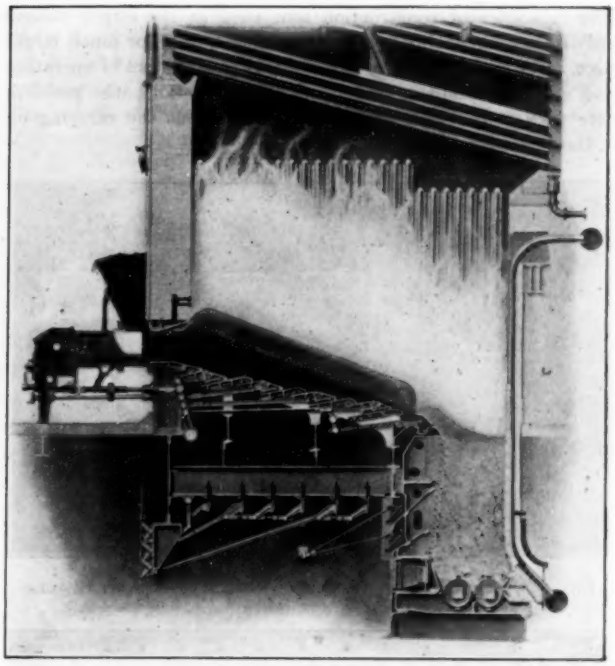


FIG. 7 COMBUSTION ENGINEERING CORPORATION MULTIPLE-RETORT STOKER

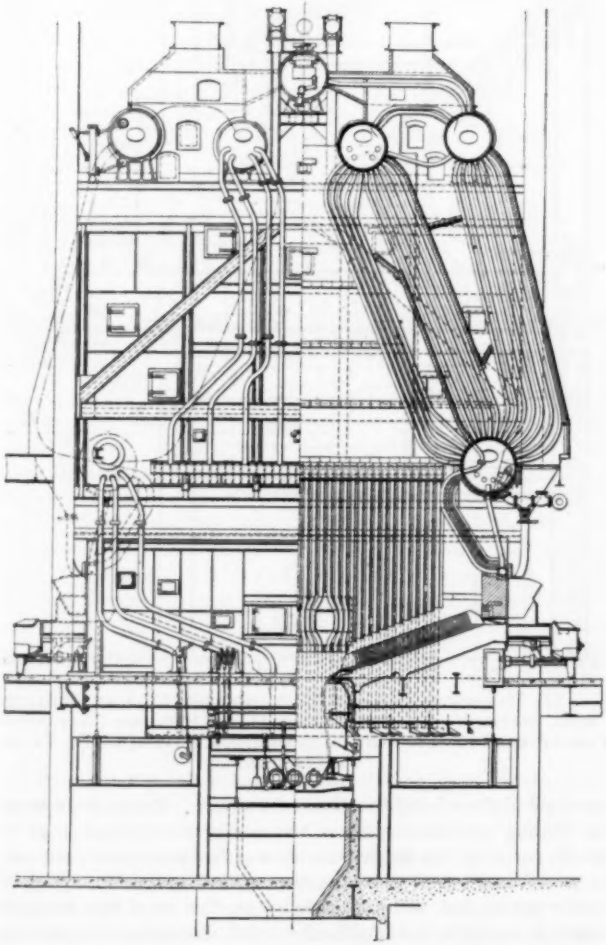


FIG. 6 COMBUSTION ENGINEERING CORPORATION MULTIPLE-RETORT STOKER FOR ALLEGHENY COUNTY STEAM HEATING COMPANY AT PITTSBURGH UNDER 3670-HP. BOILER—THIS STOKER IS 14 RETORTS WIDE, AND MEASURES 27 FT. 7 IN. FRONT WALL TO FRONT WALL, GIVING A PROJECTED AREA FOR THE DOUBLE STOKER OF 716 SQ. FT.—THE UNIT IS DESIGNED FOR 400,000 LB. STEAM PER HOUR

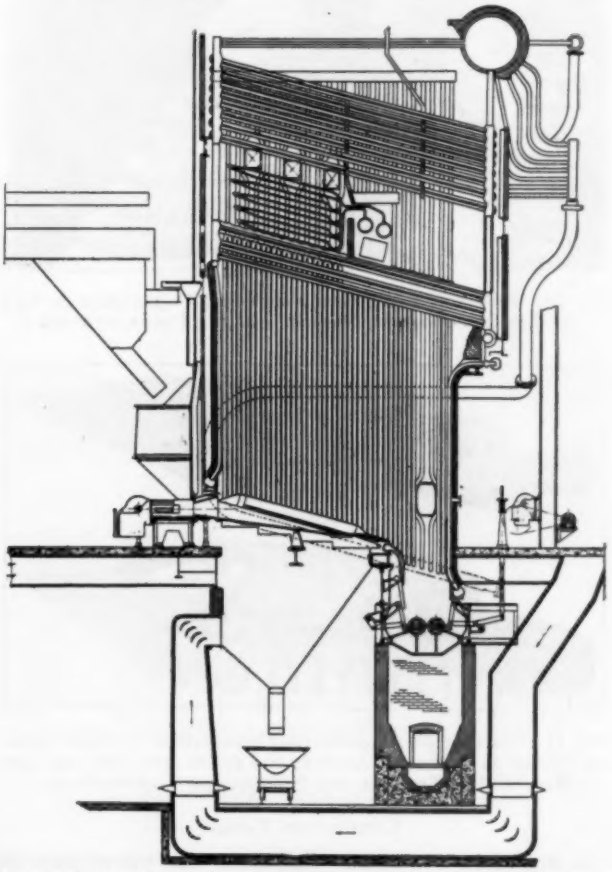


FIG. 8 RILEY MULTIPLE-RETORT STOKER AT KEARNEY STATION OF PUBLIC SERVICE ELECTRIC AND GAS COMPANY UNDER 2360-HP. BOILER—THE STOKER IS 15 RETORTS WIDE AND HAS 479 SQ. FT. AREA

The principles of construction now used in the larger stokers are directly applicable to the structure required for much larger sizes. High average combustion rates are a product of operating skill and refinement in stoker and furnace details; the problem here with respect to still larger stokers is but the carrying on of the best practice now in effect.



FIG. 9 SHOP VIEW OF ASSEMBLED RILEY 15 RETORT STOKER

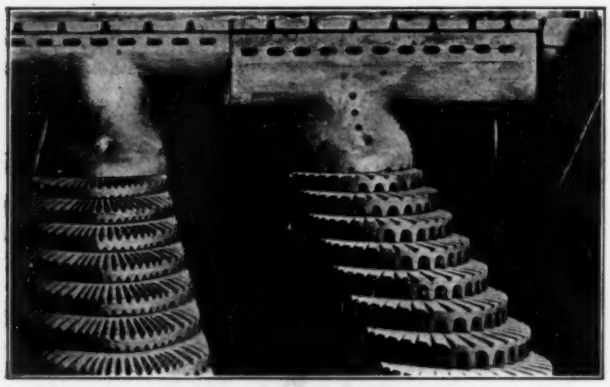


FIG. 10 WESTINGHOUSE TUYÈRES SHOWING COMPARISON IN SIZE OF AIR OPENINGS FOR NORMAL AND FOR PREHEATED AIR

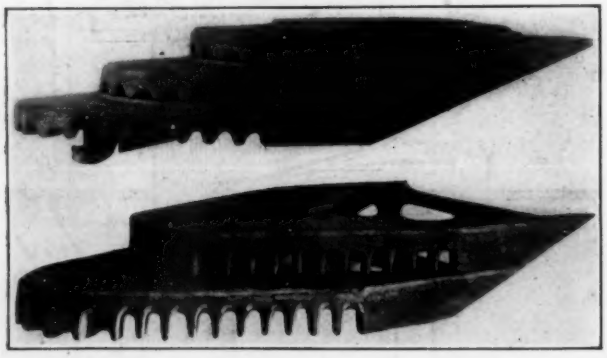


FIG. 11 COMBUSTION ENGINEERING CORPORATION TUYÈRES SHOWING SIZE OF AIR OPENINGS ADOPTED FOR PREHEATED AIR COMPARED WITH OLDER TUYÈRES FOR NORMAL AIR TEMPERATURES

COMBUSTION RATES

It is difficult to make any compilation that will evidence an increase in combustion rates. It may be perfectly true and somewhat obvious that combustion rates have been and are gradually increasing. Various comparatively old records of 70 to 75 lb. of coal per sq. ft. of stoker area per hr. would seem to rob any recent achievements in this line of all claim to recent development. The more prosaic figures of maximum operation under ordinary plant operating conditions would be much more important, but they are somewhat too elusive for accurate compiling.

There is, however, improvement to report along this line which will have recognized, if not quantitative, value. The problem of increasing combustion rates for regular operation involves pretty much all of the related functions of a stoker. Particularly must the feeding and distributing of the fuel be worked out so as to handle correctly the larger quantities of fuel per square foot, and to secure its economical combustion. Air distribution, speed of stoker mechanism, and fire thickness are factors of this problem which do not offer much difficulty. Ash elimination is a serious problem only in the case of coals having a low fusion ash, or extraordinary clinkering tendency. Furnace design is of extreme importance, and its problem with respect to high-combustion rates has been solved successfully by the work of stoker engineers in combination with the manufacturers of water-cooled walls

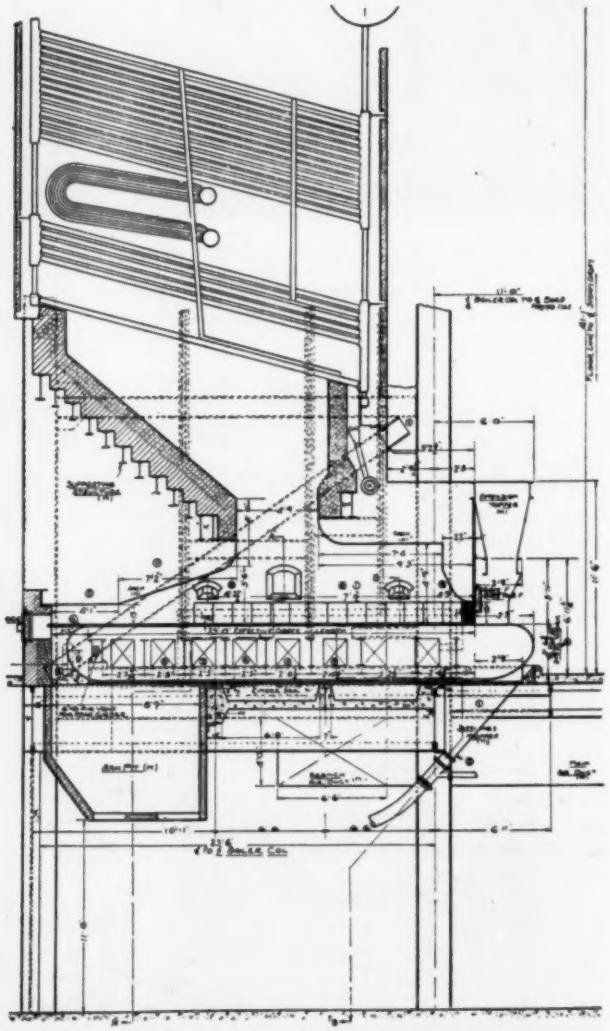


FIG. 12 HARRINGTON STOKER FOR ANTHRACITE COAL AT HUNLOCK CREEK STATION OF LUZERNE COUNTY GAS & ELECTRIC COMPANY—THIS STOKER IS 21 FT. WIDE BY 23 FT. LONG, HAVING 483 SQ. FT. OF STOKER AREA

and air-cooled refractory walls. This subject, however, is not regarded as within the scope of this paper.

Various stoker manufacturers have attacked the problem of development individually, as they visioned the requirements. This has led to a different stressing of the fundamental details in the various cases. In each case, the manufacturer has achieved some material measure of success, and in the large sense of a craft report, the total of these improvements and developments is along the line of higher consistent combustion rates forstokers.

In multiple-retort underfeed stokers, one manufacturer has developed particularly the length of retort, together with special control of secondary rams; the object being to secure high combustion rates for his large-size stokers. Another manufacturer has developed a deep ram and deep retort, and also special functioning of secondary rams; the object being practically the same, although the method is different. Still a third manufacturer has centered his development particularly on the overfeed section of this type of stoker, enlarging and articulating it according to his view of the requirements of proper combustion, and this also is a still different method of attacking the common problem. In these three developments the object is to provide for practical handling of the fuel as a fire, over greater distances from the point of feeding than formerly. The problem has been difficult and very serious, but by no means insurmountable, and in each case a certain measure of success has resulted.

For the chain-grate type of stoker, higher combustion rates have been largely the result of improved furnace design. The various modifications of the rear arch have lifted the old 45-lb. per sq. ft. rate up to the present 60-lb. rate. Of course, there are instances of higher rates. Special high rates are not to be emphasized, but the gradually increased rate of coal burning in every-day practice should be.

Along with improved furnace design, there has been an increase in speed-of-grate travel. For bituminous coals the grate speed has increased from an old limit of about 30 ft. per hr. to a present-day usage of about 60 ft. per hr. For anthracite coal the older limit of about 45 ft. has been superseded by present maximum speeds of about 100 ft. per hr.

The development in higher combustion rates has always been gradual. For all the various fuels, and with all the various stokers, there has usually been a pioneer installation that demonstrated some principal point of improvement. This point has then been tried out and varied in other subsequent installations, with the result that its advantages have been gradually but thoroughly worked into the whole practice of the art. In time the discoveries and improvements, whether originating in operating skill or vision of stoker owners, or finding their inception within the organization of any particular manufacturer, are absorbed and distributed widely throughout the whole field of combustion of fuel on stokers. This, while far from specific, and incapable of mathematical precision of expression, does constitute a real and practical advance in the art.

Preheated Air

Preheated air has caused all stoker manufacturers to build stoker bodies, and all parts in contact with the preheated air, so as adequately to provide for structural expansion. There is a trend toward making the air openings through the various kinds of grate surface in the different types of stokers generally larger for preheated air than for air at normal room temperature.

The allowable limit of preheated temperature of air to stoker fires depends partly on the characteristics of the fuel. Certain coals with preheated air have a tendency to fuse or mat over, thus becoming almost impervious to commercial forced-draft air pressures. In certain cases, 300 deg. fahr. has been found to be a limit on this score.

A temperature of 350 deg. fahr. for preheated air with stokers seems to be regarded as safe both by stoker manufacturers and the public which they serve. There is a divided opinion as to the practical effect on stokers of air at higher temperatures. The burden of objection to higher temperatures is on the score of excessive maintenance of grate-surface elements.

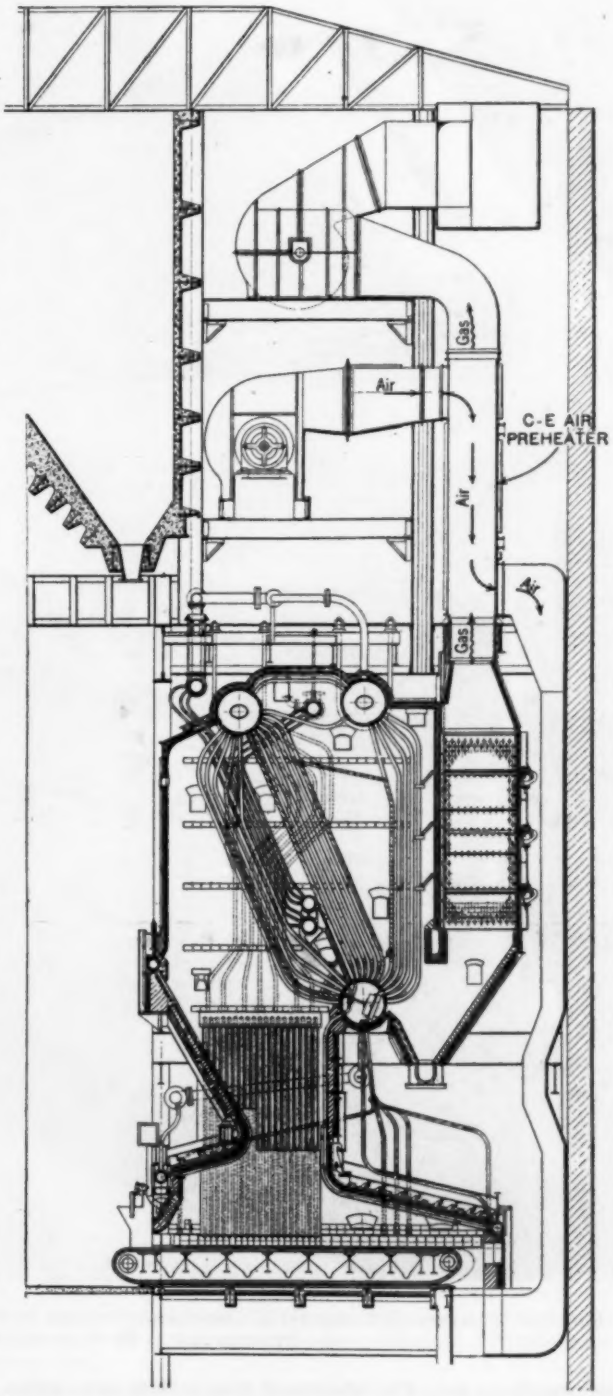


Fig. 13 Combustion Engineering Corporation Stoker for Midwest Bituminous Coal at Grand Avenue Station of Kansas City Power and Light Company Under 1516-Hp. 700-Lb. Pressure Boiler—The Stoker Is 24 Ft. Wide by 22 Ft. Long, Having 528 Sq. Ft. of Stoker Area

Small Stokers

The single-retort stokers, usually considered for boilers of from 200 to 600 hp. in size, have, as a class, developed recently only with respect to convenience in the arrangements for feeding coal and supplying air. The interiors of these stokers have undergone practically no change. But a rather definite demand for electric drive and self-contained source of air supply has caused the development of compact combinations of geared drive and fan worked out for simultaneous driving from a single motor.

In still smaller sizes a large market has recently opened up for stokers to fit under small heating boilers. Practically all of these machines use the combination of speed-reduction gear for coal feeding, together with a small fan for forced-draft air, both driven from a common motor. In fact the development of this style of combined-fuel feeding and source of forced-draft air in these very small stokers has set the style for this arrangement with the small stokers for power boilers.

The stokers for heating boilers are almost all of the screw-feed

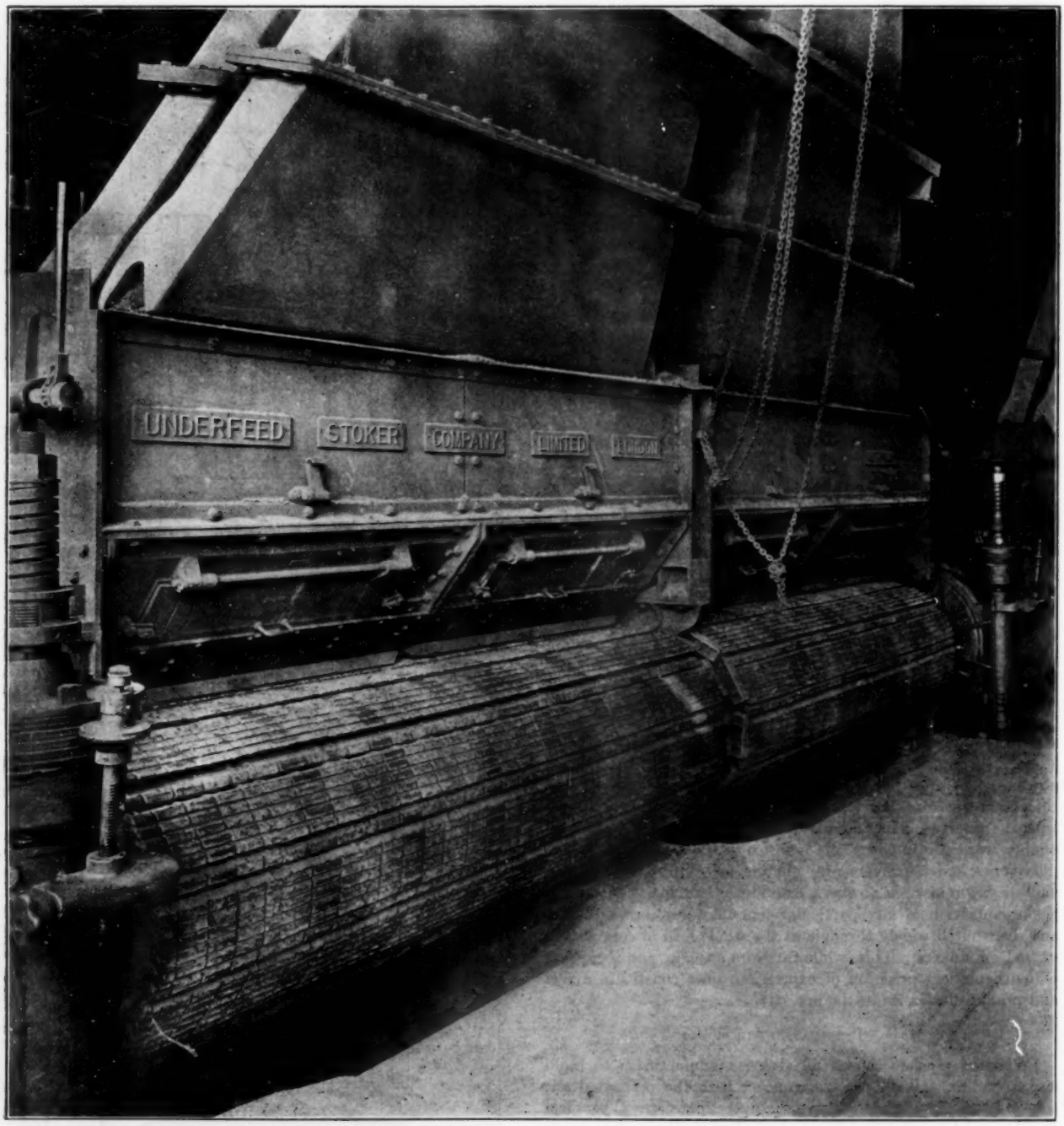


Fig. 13A Combustion Engineering Corporation Stokers in Deptfort Power Station of the London Power Co., Ltd. These Stokers Are 31 Ft. Wide and 700 Sq. Ft. of Stoker Area Each

type, where the coal is fed from the bottom of a hopper, through a delivery pipe, into the bottom of a retort, by means of a slowly revolving screw. Single-retort stokers for small-power boilers

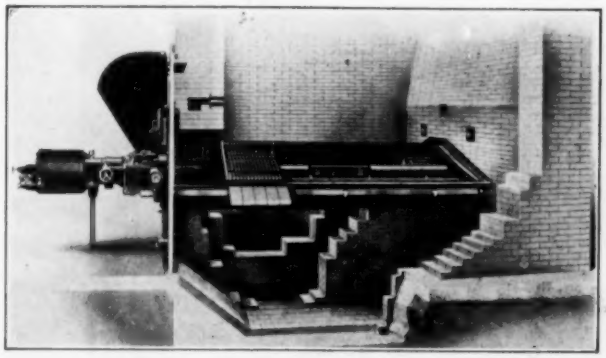


FIG. 14 TYPICAL VIEW OF WESTINGHOUSE SINGLE-RETORT STOKER, SHOWING THE LINK-GRATE FOR AGITATION OF FUEL BED

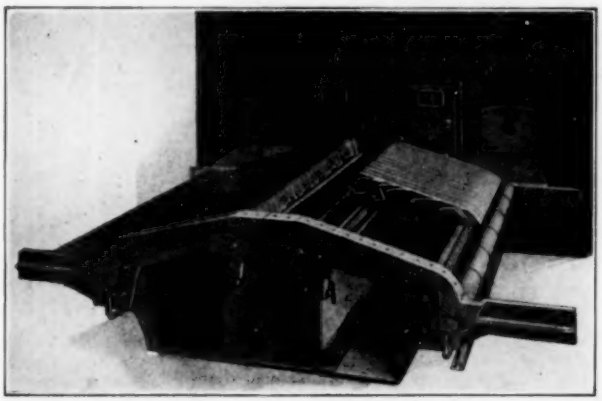


FIG. 15 TYPICAL VIEW OF COMBUSTION ENGINEERING CORPORATION TYPE E SINGLE-RETORT STOKER, SHOWING ALTERNATE MOVING GRATE BARS FOR AGITATION OF FUEL BED

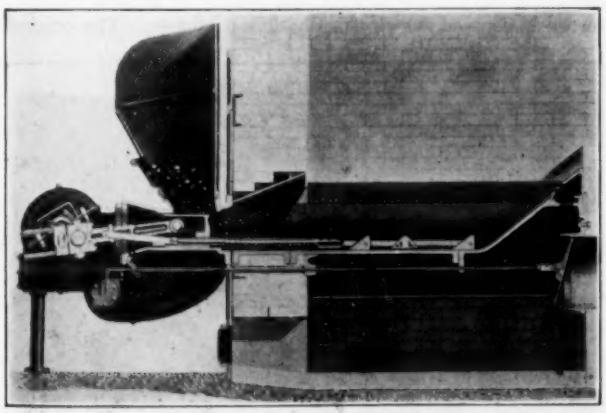


FIG. 16 TYPICAL VIEW OF DETROIT SINGLE-RETORT STOKER, WITH RAM TYPE OF COAL FEED AND AUXILIARY PUSHERS

are usually larger in size, and utilize rams for feeding the fuel from hopper to retort.

As a class, the heating-boiler stokers do not provide any agitation of the fire, whereas the small-power boiler stokers have means for agitation in order to break down the coke masses, and in this point they resemble the action of the large multiple-retort stokers.

THE FIELD FOR STOKERS

A study of the statistics compiled by the Department of Commerce has yielded information that indicates the continued popularity of the stoker as a means for burning fuel under steam-power boilers. These statistics include a summary of the horsepower of water-tube boilers sold each year in the United States,

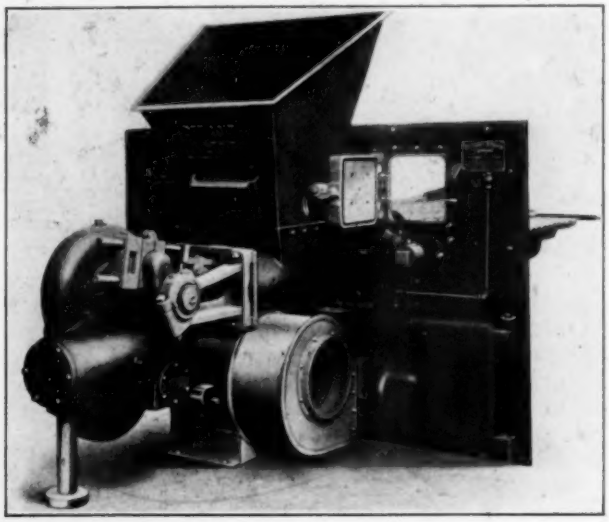


FIG. 17 FRONT VIEW OF DETROIT UNISTOKER, SHOWING THE COMBINATION OF GEARED DRIVE AND FAN, DRIVEN FROM A COMMON MOTOR

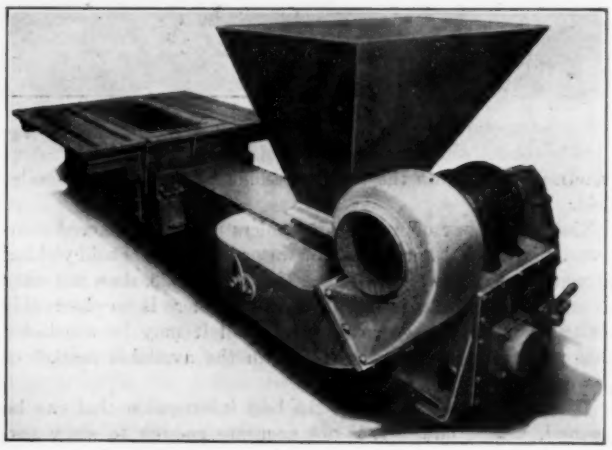


FIG. 18 COMBUSTION ENGINEERING CORPORATION SMALL INDUSTRIAL STOKER WITH COMBINATION GEARED DRIVE AND FAN, DRIVEN FROM A COMMON MOTOR

and reported separately from the horsepower of fire-tube boilers. It may be assumed without serious error that the horsepower of water-tube boilers represents the potential market for all methods of firing steam-power boilers. This set of figures dates from 1919.

There are also further figures for the number of stokers purchased each year, and these are divided into totals for fire-tube and for water-tube boilers. In addition, the horsepower of boilers fitted with stokers is given. These figures date from 1923, and while not entirely inclusive will serve with sufficient



FIG. 19 FRONT VIEW OF HUBER STOKERS

exactness to indicate the trend of stokering in the power-boiler field.

The horsepower of water-tube boilers that are stokered compared to the total horsepower of water-tube boilers sold yields a percentage figure which, commencing with 1923, does not vary to any great amount from 60 per cent. There is no observable tendency in this curve to drop off, and it may be concluded that stokers are holding their own in the available market of power boilers.

It may also be added that the best information that can be secured, which, however, is not accurate enough to show as a curve, indicates something like 20 per cent of the horsepower of all water-tube boilers are being fired by the pulverized-fuel method. This supplementary market for mechanical firing is doubtless largely in the field of exceptionally large units. The final residue of about 20 per cent of this boiler-firing market is probably being sold for the methods of gas and oil firing, and for hand firing. It is very interesting to note that stokers are maintaining their popularity as the major method for firing steam-power boilers.

Discussion

JAMES W. ARMOUR.[2] Among small stokers, the importance of which the author has pointed out, there is a decided tendency toward a demand for electrically driven stokers. The original

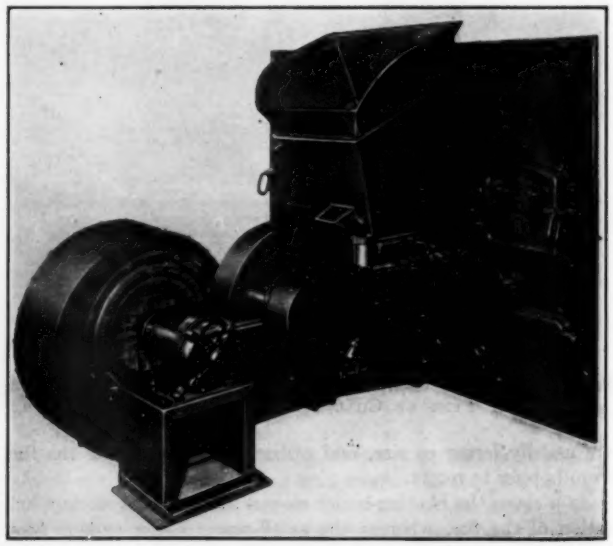


FIG. 20 RIFLEX STOKER DRIVE

[2] Vice-President and Engineer Manager, Riley Stoker Corporation, Worcester, Mass. Assoc-Mem. A.S.M.E.

single-retort small stoker was steam-driven. This type of drive lends itself to automatic control and quite a variation in range of coal feed. Electrically driven stokers do not lend themselves so well to this wide variation in range. Alternating current usually is the available power, and it is very difficult to control the variation in speed of the load such as is imposed by the plunger of a single-retort stoker. Direct current works very satisfactorily, but there is still the objection that the mechanical drives driven by a worm reduction cannot be run very fast. Since these stokers are quite frequently installed for heating loads, such as in schools and apartments, the operator must get on the job early in the morning in order gradually to build up his fire, and he must start early in the evening to build up a bank to last him through the night. It would therefore be highly desirable to have a stoker which can be operated extremely fast for short periods in the same manner as a steam-driven stoker can be operated.

Such a stoker drive, known as the Riflex, was recently put on the market by the company with which the writer is associated. (See Fig. 20.) Speed variation up to 6000 lb. of coal an hour can be obtained. A constant-speed squirrel-cage motor is used, thereby getting away from difficulties encountered with brushes and commutators.

R. A. Foresman.[3] The author has stated that the trend in boiler-plant development is to larger and larger sizes of boiler units and that the major method of generating steam from coal is with stokers. During the past decade the economic problems arising from this development have been of an extremely varied character. The indications are they will continue so and that stoker manufacturers must be prepared to develop their equipment in accordance with these constantly changing conditions.

It was further pointed out that the manufacturer was given no opportunity to determine what the performance of a major development would be by trial installation. With users demanding performance that can be secured only by a major change to existing equipment or by designing a complete new machine, the stoker manufacturer is confronted with a most difficult problem. Not only must the mechanical design be sound and the performance results good, but these should be produced without any more delay and interruption to the service than would occur with a completely developed commercial machine. Under these circumstances there have occurred notable examples of failure to obtain predicted results or to secure them only after protracted effort and great expense.

In any industry that depends upon empirical science to guide it, as does the stoker industry, this is to be expected, and instances of this will continue to occur. This is a cost that the industry must pay if it is to progress.

The record that has been made in saving coal and reducing steam-power costs in recent years is a very gratifying one and proves the sound economic value and good commercialism of the research and experimental work performed to overcome these difficulties and reduce to common practice the improvement it was found possible to make.

Without dwelling on the developments that have been reduced to practice, the writer will mention some that will have a major influence on the economics of steam generation of the future.

(*a*) Notwithstanding that the double-end multiple-retort stoker is not a popular type, it has inherent characteristics that make it superior to the single-end stoker. Comparing these two types, each having the same area and burning the same amount of coal, the principal advantages of the double stoker are:

1 Feeding and distributing coal from the two ends of the furnace
2 The lineal travel of the coal and its speed reduced to one-half
3 Radiant heat from either fuel bed is directed upon the other, which will minimize to a substantial degree the effect of water-cooled walls upon the ignition and combustion of the coal
4 A stream of rich gas from each end of the furnace and one of lean gas from the center, converging at the boiler uptake and causing a turbulence that produces an ideal combustion condition
5 It is more responsive in every respect to varying load
6 It is capable of producing more steam per boiler unit than has been required up to the present
7 It eliminates the first cost and maintenance of an expensive bridgewall.

The question immediately arises as to whether these advantages will not be offset by the double-stokered unit requiring a greater amount of space per unit of power developed, the additional coal bunkers, and the operating cost of looking after two stokers instead of one. With a steady increase in the size of the boiler unit and the capacity that is required, it is possible the first two may prove to be an advantage. The large coal consumption of the boiler unit of the future may make it possible to reduce this cost by using a bunker over each boiler aisle when distribution of weight, space utilized, coal handling, and reserve capacity and performance results are carefully considered. The last will certainly be assured by the improved performance and lower cost per unit of power developed.

Therefore, there is a strong probability that among the major advancements in the economics of steam generation in the large plants of the future the double-end stoker will be important.

(*b*) As to sustained high-combustion rates, the responsiveness of the multiple-retort underfeed stoker to variable demands has always been one of its most valuable characteristics. Originally, performance at or near its maximum was of relatively short duration. This provided a considerable margin between normal and maximum operating performance. The development of the large boiler unit has materially reduced this margin because of the economic necessity of operating continuously for relatively longer periods at combustion rates more nearly approaching the maximum. Much progress has already been made in this respect, but more is going to be required of the future. It is necessary to amplify this thought somewhat.

There is still a relatively big gap between test and operating results. With respect to capacity the measure of a boiler plant's value is the greatest amount of power it can be depended upon to produce, for a given time, without interruption.

It is not a very difficult thing to operate a single unit on a test at or near maximum combustion rates for a period of six to eight hours or longer, but it is an entirely different matter for the operators to take a number of identical units or a group of boilers comprising an entire station and duplicate this performance on each and every one of them for a like period. There are still too many variable factors in actually burning coal for this margin to be entirely eliminated. However, there has been steady improvement in this respect in recent years. The time is rapidly approaching when the highest combustion rates it is economically sound to operate at can be maintained on all the boiler units comprising a group as long as may be desired.

It is not expected to see our present maximum coal-burning rates per unit of area of stokers greatly exceeded. What the writer has endeavored to bring out is that the stoker of the future will be a machine which under the normal service conditions will enable the operator to maintain higher and more uniform combustion rates on each and every unit with absolute dependability and for longer periods than can be secured at present.

[3] Chief Engineer, Stoker Department, Westinghouse Electric & Manufacturing Company, Philadelphia, Pa. Mem. A.S.M.E.

(*c*) As to stoker maintenance, up to the present time a stoker is usually taken out of service for reconditioning in not to exceed 90 days. In but few cases does the service period exceed 60 days. Considerable progress has been made in reducing stoker maintenance, but the stoker of the future will have to operate for still longer periods.

Instead of a service period of 90 days, the manufacturer will have to look forward to producing stokers which will operate consistently for service periods approximating six months at the very least.

(*d*) In general, the appearance of the boiler unit has not been particularly attractive, and the stoker equipment has added but little, if anything, to this. Frequently after being in service for some time the evidence is only too conclusive that no effort is spent upon the stoker beyond what is an actual necessity.

The stoker of the future will undoubtedly present a more pleasing appearance, prevent dust and dirt from escaping from it, and by its design and finish become sufficiently attractive so that it will be kept in good condition.

This phase of the development may not be considered an economic development, but it is a foregone conclusion that the better appearance the equipment makes the more effort there will be in keeping it up, and the more effort there is in this respect the better it will be maintained, Therefore, it is quite likely to prove an economic benefit to both the user and the manufacturer.

RALPH A. SHERMAN.[4] Neither the author nor the discussors have brought up the question of overfire air for the underfeed stoker. All are familiar with the improvements made in the design of furnaces for traveling-grate stokers in which secondary air is admitted at the front of the stoker, at the throat of the front and rear arches, or that which comes through the rear of the fuel bed and is brought forward to mix with the combustible gases by means of a long rear arch.

The fuel bed of an underfeed stoker is deep; therefore, gas with a high combustible content rises from various parts of the bed, particularly in the front where the volatile matter is being driven off. Free oxygen finds its way through the middle of the bed through cracks formed by the shrinkage and movement of the coke, and through the thinner bed on the extension grates. The walls of the furnace generally rise vertically toward the boiler tubes, and no provision, other than the natural turbulence and diffusion, is made for mixing these gases. That the underfeed stoker is an admirable mechanism for firing coal is unquestioned, but the combination of these combustible gases with free oxygen above the fuel bed seems to be more a result of chance than of the design of the stoker or furnace.

That stratification does exist in the furnace and that carbon monoxide, and even hydrogen, pass unburned to the uptake was proved during tests on two underfeed stoker-fired furnaces in the course of the investigation of furnace conditions which affect the life of refractories.

Some experimental work, which has not been published, has been done by one operating company on the admission of overfire air at the front wall of an underfeed stoker with apparently good results. It is understood that overfire air is being provided in one of the installations pictured in the author's paper.

However, much work remains to be done. A thorough investigation of the process of combustion in the underfeed stoker has never been made. Such an investigation with a modern stoker should give valuable information on the proper use of the controls of shape of fuel bed and of the zoning of air with which stokers are now equipped. It should show whether the design of the furnace generally used is correct. Perhaps a front arch would be helpful; the available water-cooled arches do away with the difficulties inherent to the refractory arch. It would indicate whether overfire air is necessary or desirable and the proper points for its admission. Furthermore, it should show why, when at least one-half of the process of combustion occurs within the relatively small volume of the fuel bed, the average rates of heat release for the entire furnace are little greater than for powdered-coal furnaces.

[4] Fuel Engineer, Pittsburgh Experiment Station, U. S. Bureau of Mines, Pittsburgh, Pa. Assoc-Mem. A.S.M.E.

FSP-52-19

Electrically Heated Hardening and Tempering Baths Compared With Fuel Methods of Heating

By WIRT S. SCOTT,[1] MANSFIELD, OHIO

Necessity for uniform products and improved working conditions led the Warren Axe and Tool Company to replace its fuel-heated furnaces with electrically heated types. A reduction in costs of manufacture has also resulted, so that, based on actual book profit, the new equipment will pay for itself in 18 months. The new equipment, its control and operation, is described in this paper.

PRIOR to electrification the axes manufactured by the Warren Axe and Tool Company were heated in semi-muffled gas furnaces. Individual treatment for each axe was thought to produce the ultimate in heat treatment. The possibility for error in judgment on the part of the heat treater was appreciated, but this was minimized as much as possible by employing only experienced high-grade heat treaters.

The "Sager Brand" axe is of a dark blue color, the natural color of the tempered steel. Securing of this color in tempering was the aim of the heat treaters, but the accomplishment was a matter of judgment alone, and subject to many variables, resulting in different shades of tempered steel and different degrees of hardness. The desirability of improving the quality of product and of securing the most dependable uniform quality was apparent, and a decision was made to eliminate some of the variables and reduce others to a minimum through the use of electric heat, regardless of the operating cost.

Electric furnaces were put in operation Feb. 1, 1929. The entire method of heat treatment was revised. Batch-type, semi-continuous operation was used; bath-type furnaces replaced hearth type; separate hardening and tempering furnaces were used; one hardening and one tempering furnace replaced five combination hardening and tempering furnaces. This was the program decided upon, based upon experience gained and results produced in other fields.

The electric furnaces are of the rectangular bath type, consisting of a box-like steel-plate shell riveted and welded together, and lined with heat-insulating and refractory brick to form a heating chamber. Heating elements formed of return-bend coils of nickel-chromium resistors are suspended from hooks anchored in the sidewalls, and are placed in the bottom in suitable refractory spacer blocks.

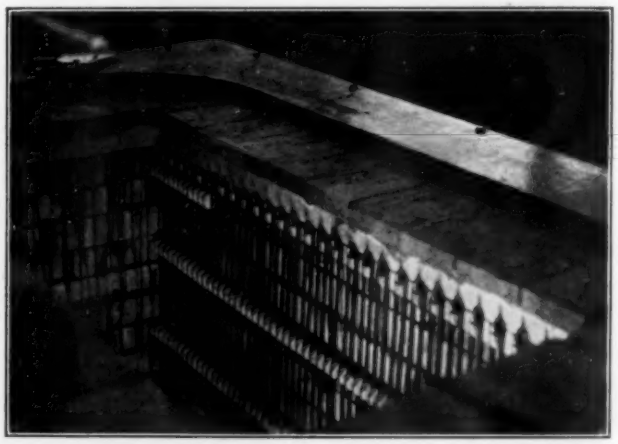
The vessel for the bath is made of cast nickel-chromium alloy, supported in the heating chamber by means of a flange resting on the brickwork. A nickel-chromium protective apron is placed around the flange, and over this is laid a steel top plate so as to protect thoroughly the brickwork from damage, and to prevent the molten bath from entering the heating chamber.

The axes were to be suspended by suitable hooks in heated lead and salt baths, held at the proper temperatures for the hardening and tempering operations. Due to the fact that lead is heavier than steel, means had to be provided for holding the axes down in the lead bath.

The hardening-bath furnace is operated at a bath temperature of 1405 deg. fahr.; has a vessel, the inside dimensions being 22 in. wide × 60 in. long at the top × 14 in. deep. The electrical capacity is 65 kw., two phase, 60 cycles, 220 volts. The tempering-bath furnace is operated at a bath temperature of 580 deg. fahr.; has a vessel, the inside dimensions being 12 in. wide × 46 in. long at the top × 12 in. deep. The electrical capacity is 57 kw., two phase, 60 cycles, 220 volts.

Each furnace is divided into two equal circuits, each circuit operated by separate magnetically operated contactor switches, which in turn are operated by means of temperature-recording and controlling pyrometers.

Two thermocouples are used with each furnace, one of the protected type located directly in the bath, and the other a bare type within the furnace chamber. This double protection and control is necessary fully to protect the heating elements against excessive temperatures, particularly during the heating-up periods, at which time the temperature of the bath does not rise as rapidly as that of the heating chamber, and the chamber temperature tends to rise much faster than the heat can be absorbed.



CONSTRUCTION OF HEATING CHAMBER USING ROD-TYPE ELEMENTS

The production consists of two distinct classifications of axes, the Sager Brand or all-blue tempered, and the various other brands under distinctive trade names. These later brands are finished in various colors of enamels, and are not required to have the blue finish. The production in one month consisted of 6400 blue-finished single-bit axes, 9500 blue-finished double-bit axes, 10,900 single-bit in other finishes, and 525 double-bit in other finishes. The total number of bits or cutting edges to be heat treated was 37,350 in the month referred to.

The axes are made of two different kinds of steel. The body is forged from low-carbon steel, and to this is welded the cutting edge, consisting of high-carbon crucible steel. Only the blade or

[1] Special Representative, Westinghouse Elec. & Mfg. Co.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

cutting bit need be heat treated, hence it is unnecessary to heat the entire axe, except for the purpose of obtaining the blue finish on the Sager Brand, which is done in the tempering bath.

In the hardening bath, 23 axes are suspended point down, and immersed in the lead bath up to within one inch of the hole for the handle. On top of the molten lead is placed a blanket or cover of granular carbon, which has the effect of reducing the radiation losses from the surface, reducing the oxidation of the lead, and giving a wiping action for removing lead globules from the axes. After the axes have been in the bath a definite length of time they are removed, quenched in brine, rinsed in clear water, and are ready for tempering.

A tempering salt is used, which if held at the proper temperature for a definite length of time produces just the shade of blue desired. The temperature, of course, had to be that required for a proper draw, and at the same time had to be sufficient for thorough saturation and no more.

In bluing, the Sager axes must receive the finished polishing after hardening and before tempering. With the process now used a beautiful dark-blue shade is obtained uniformly over the body of the axe, and a slightly lighter shade over the cutting edge or bit. Just so long as the temperature remains constant within certain limits, and the work is kept submerged in the salt for just a definite length of time, the same shades of blue are reproduced exactly.

One of the most interesting phases in the use of electric heat is the possibilities of accomplishing results heretofore unattainable, and often the securing of results unlooked for, which in themselves more than justify the installation. The possibilities with equipment now available are not always appreciated, and not used to the fullest extent until confronted by conditions that call for further refinement of control beyond that previously used.

An illustration of this fact was brought out in an attempt to obtain a much closer temperature control of both the hardening and tempering bath than had previously been obtained with electric furnaces. With bath-type furnaces, due to the inherent sluggishness of the bath to respond to changes in heating-chamber temperatures, the temperature of the bath does not follow immediately that of the heating chamber. For example, should the temperature of the bath begin to drop below normal, and by doing so cause to be liberated or turned on an increased amount of heat energy, the temperature of the bath will continue to drop from 30 to 50 degrees below normal (depending upon the type of furnace used) before it begins to rise again. This increased heat energy or source continues adding heat to the bath until normal temperature is regained. However, by this time the entire furnace chamber, consisting of refractory firebricks, has been heated to a much higher temperature than the bath, and has absorbed a considerable quantity of heat, which it begins to liberate to the bath just as soon as the heating medium is reduced or cut off entirely. The net effect is to continue to supply heat to the bath after the proper temperature has been reached, with the result that the temperature of the bath continues to rise from 30 to 50 deg. above normal.

By using an electric furnace having one thermocouple in the bath and one in the heating chamber, it is possible to make the bath temperature respond more nearly to that of the furnace chamber, cutting off the current when the heating chamber



GENERAL VIEW OF ELECTRICALLY HEATED LEAD-HARDENING FURNACE

(Double-bit axes are being heated. The bit only is immersed in the lead. There are three lamps on the wall at the extreme right with the white light indicating proper temperature and correct rate of production. The brine quench tank is directly back of the furnace operator. After finishing the day's production the rack for holding the axes in the bath is removed, and the insulated cover is placed in position over the lead bath so that the lead is still in a molten condition next morning.)

reaches a predetermined temperature, thereby preventing an excessive storage of energy, and later when the heating chamber has cooled off somewhat, cutting on the current again. This action anticipates to an extent the demands for heat within the bath. Such a system of control can be depended upon to hold the temperature of the bath within 20 deg. plus or minus.

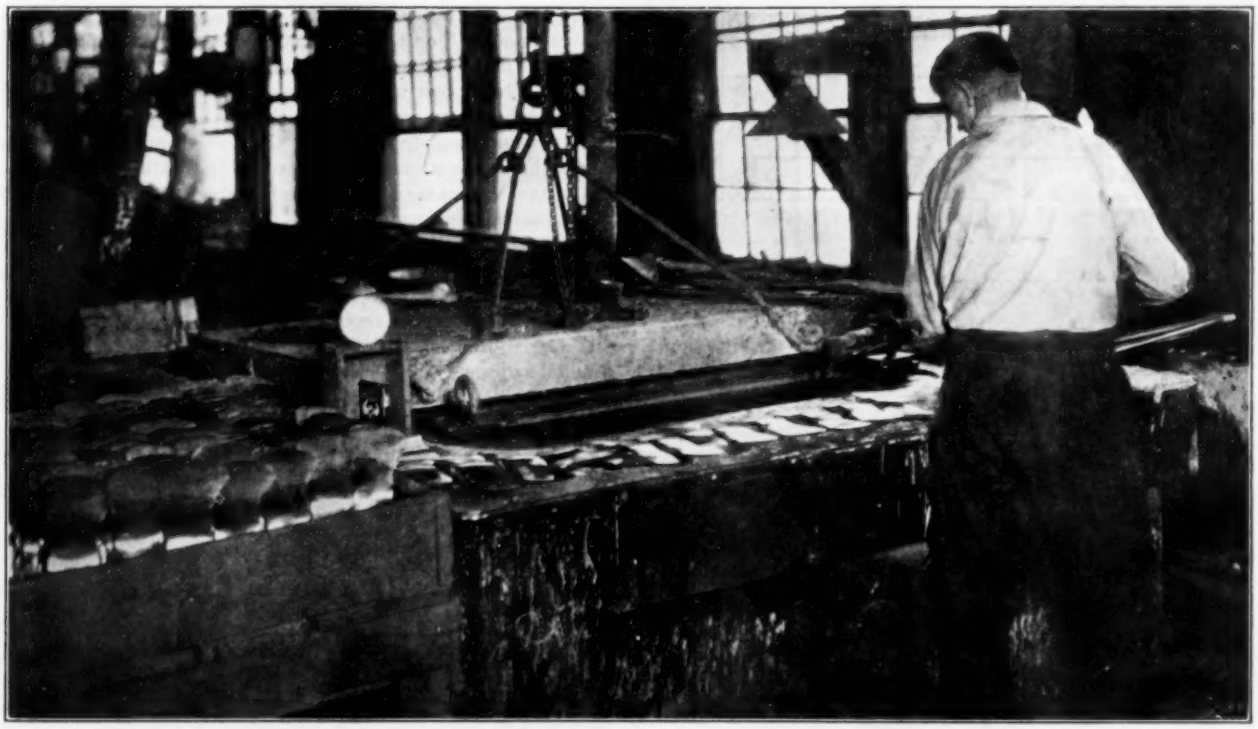
Theoretically, if a definite amount of heat be supplied each minute to a bath, it should be possible to put just enough work through the furnace each minute to hold the temperature constant. This principle in combination with the electric furnace described was put into effect. Since the axes had to be placed on and removed from the hooks by hand, some positive means had to be employed to indicate to the operator when he was operating at the production rate which would tend to keep the furnace at a uniform temperature. This was accomplished by placing three contacts on the pyrometer. These operate three lamps placed over the corresponding furnace. A white light indicates correct temperature; a blue light, temperature too low; a red light, temperature too high; a white and blue light, a downward swing in temperature; a white and red light, an upward swing in temperature.

With 23 hooks suspended over the hardening furnace, 22 of which always hold axes, the operator's job is to remove one heated axe and replace it with a non-heated axe at such a rate as to keep the white light burning. Should the blue light come on with the white, the operator must slow up; should the red come on with the white, he must work faster. Should the white light go off entirely, the operator must cease work until conditions become normal. The extent to which this is accomplished is shown graphically by the recorded-temperature chart in the control room.

In practice it is impossible to comply entirely with this theory. The weights of axes vary; consequently, the rate of heat absorption is not constant; hence variable amounts of power are required at different times. This condition necessitates a furnace having a capacity sufficient for heating the largest charge at a given production rate, and as a factor of safety, an excess capacity for unusual or unforeseen contingencies. The temperature variation of this combined system under the worst conditions was sufficient to hold the temperature within 15 deg. plus or minus of 1405 deg. fahr.

Control of the hardening temperature within 15 deg. was a tremendous improvement over any results anticipated by the Warren Axe and Tool Company, and the elimination of 80 per cent of the field rejects was assured provided the operator could be depended upon to watch his lights. A difficulty arose in this respect, however. One two-point control instrument was used on each furnace, the instrument, automatically every 30 sec., changed from the thermocouple in the bath to the thermocouple within the heating chamber. Since the indicating lights had to be connected to show the bath temperature, there was therefore a period of 30 sec. out of each minute during which the lights did not burn. This resulted in much uncertainty and confusion. The situation was remedied by replacing the two-point instrument with two single-point instruments, one for the bath and one for the heating chamber, thereby permitting the lamps to burn uninterruptedly.

The use of two control instruments, two separate contactor switches, and two circuits in each furnace opened the way for an investigation for further refinement of temperature control, insuring a closeness of control that would positively give a still better product, and eliminate field rejects due to improper heat treatment. As previously explained, the surging of the temperature had been reduced from 30 deg. to 20 deg. by controlling the temperature of the heating zone. If the amount of energy necessary to control on and off intermittently could be reduced, the corresponding fluctuation or surging in temperature would be reduced. This was accomplished by taking one circuit of the

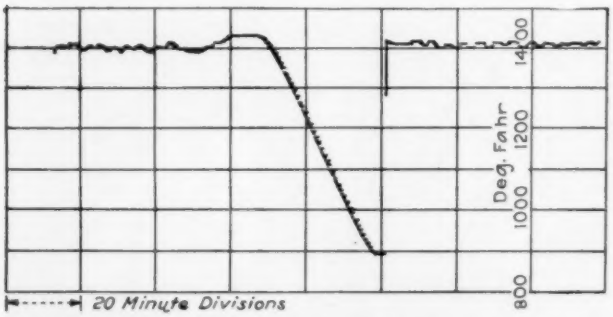


TEMPERING DOUBLE-BIT AXES IN A SPECIAL SALT SOLUTION THAT TEMPERS AND BLUES THE AXE AT THE SAME TIME

furnace entirely off the recording-control pyrometer, and placing it on the control pyrometer for the heating chamber, with the net result that one-half of the furnace capacity was in constant service and one-half in intermittent service.

Since the electrical capacity of one-half the furnace was more than that necessary to supply the radiation losses, during the noon-hour period when no work was being heated the constant-heat circuit is cut out, and the temperature-control circuit alone used for maintaining the temperature. Furthermore, at all times whether one circuit or both circuits are on, they are under the control of the heating-chamber thermocouple and pyrometer which cuts off the current entirely should the temperature of the heating chamber rise to a predetermined point, and cuts it on again as the temperature is lowered.

The building up of excessive temperatures and excessive heat storage within the heating chamber is thereby effectively elimi-



WEEK-DAY CHART OF THE TEMPERATURE OF THE LEAD-BATH HARDENING FURNACE

Shutting down at 2.30 P.M. the bath temperature dropped to 890 deg. fahr. over night, and recovered to normal temperature again in 30 minutes.)

nated, and the bath temperature is under independent and positive control. With this scheme the temperature fluctuation from normal was reduced to $7\frac{1}{2}$ deg. fahr.

The furnaces are never permitted to cool off sufficiently to cause the bath liquid to solidify, except during the week end. Heavily insulated covers are provided which completely cover the entire furnace top. At the end of the day's run the covers are lifted from their stand by a chain hoist, and lowered to the furnace. With the current completely cut off the furnaces, the excellent heat storage and response to current applied are shown by the following data taken from temperature charts of the baths:

Hardening Furnace

Operating temperatures	1405 deg. fahr.
Hours operated per day	7
Operating hours	7 a.m. to 12 noon 12.30 to 2.30 p.m.
Number of hours each day furnace is shut down before current is again applied	$15\frac{1}{2}$
Temperature of bath when current is applied	840–900 deg. fahr.
Time required to recover or reach 1405 deg. fahr. in the bath	30–35 min.

NOTE: Furnace is held at 1405 deg. fahr. for 30 min. in the morning before the operation is begun to permit a more thorough heat saturation of the furnace chamber.

Tempering Furnace

Operating temperature	580 deg. fahr.
Hours operated per day	7
Operating hours	7 a.m. to 12 noon 12.30 to 2.30 p.m.
Number of hours each day furnace is shut down before current is again applied	$15\frac{3}{4}$
Temperature of bath when current is applied	450–480 deg. fahr.
Time required to recover or reach 580 deg. fahr. in the bath	20 min.
Time furnace is held at 580 deg. before beginning operations	20 min.

The method of operating one of the furnaces is worth recording in its sequence of operation, showing as it does the manner in which a batch-type hand operation can be made practically continuous.

SEQUENCE OF OPERATING HARDENING FURNACE

1 At the beginning of the working day, the furnace operator removes the insulated cover, and prepares to resume operation.

2 The furnace being up to temperature, a charge of 23 axes is placed on the hooks.

3 The bath drops in temperature due to the placing of the entire charge at one time in the bath. The lights change from white to white and blue, and then to blue alone.

4 The bath temperature drops to 1380 or 1385 deg. fahr.

5 Within eight minutes from the time the charge is placed in the bath, the temperature is back to 1405 deg. fahr., and the white light shows alone.

6 The operator is ready to begin regular production, and makes a note of the time on a time sheet. (A clock is provided.)

7 The first two axes are taken off hooks No. 1 and No. 2, and quenched blade first in a brine solution, and hung on hooks in the quench bath to continue to cool.

8 One non-heat-treated axe is then placed on hook No. 1 in the bath, leaving a vacant hook for readily spotting the position.

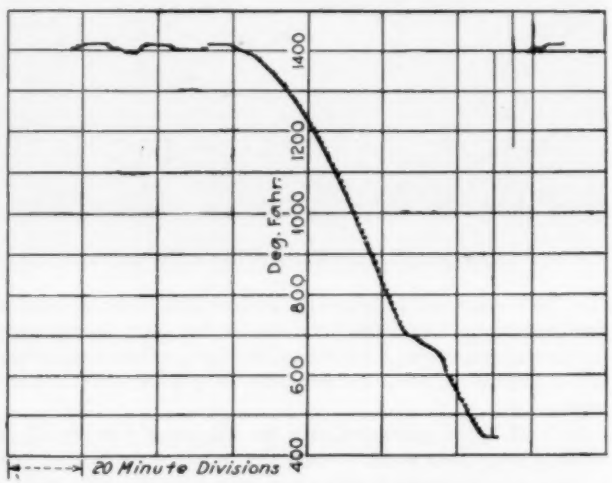
9 Axe No. 3 is taken out of the furnace, quenched, hung on a hook, and the first axe previously quenched is dipped in a rinse tank containing clear water, and then placed in a box on a truck.

10 The second non-heat-treated axe is then placed on the next vacant hook in the furnace.

11 This operation is repeated, so that by the time the last axe has been quenched, and the last hook refilled by a non-heat-treated axe, the furnace has been entirely refilled with axes.

12 Twenty-three non-heat-treated axes are laid out on the furnace plate alongside the bath, ready for resuming operation.

13 A record of the time is again made on the time card, and the quenching of the second batch started as before.



WEEK-END SHUTDOWN AND RECOVERY MONDAY MORNING

(Furnaces were shut down at noon on Saturday and started up again Monday morning, recovering normal temperature in one hour, ten minutes.)

With four-pound single-bit axes, five minutes is allowed for this complete cycle, permitting the operator to work at a very comfortable pace, without hurrying. With $6\frac{1}{2}$-lb. axes, an eight-minute schedule is required. Batch after batch was put through with only the white light showing, indicating a temperature variation of less than 5 deg. fahr. Occasionally, the white and blue or the white and red lamps would come on, in which case the operator

gaged his speed so as to cause only the white light to show. At no time did either the blue or red lights come on alone.

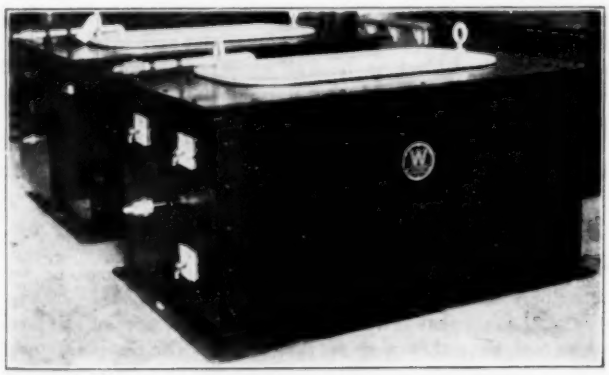
During the course of this investigation, the plant engineer was demonstrating the action of the pyrometers in control of the lamps, and in doing so opened the case of the instrument, disconnecting the control lamps. The control room is in another room parallel to the heat-treating department, but cannot be reached directly from that department, necessitating walking the length of the heat-treating room, into the connecting room, across the room, and back an equal distance to a point directly opposite the furnaces. In less than two minutes after the pyrometer case was opened, the furnace operator appeared in the control room to learn why his lamps had gone out.

The operation of the tempering furnace is substantially the same as the hardening furnace, except that the entire axe is submerged in the liquid when tempering the Sager Brand. With the double-bit axes of the Sager Brand, the entire axe being submerged, only one tempering operation is required per axe, whereas with the hardening furnace each double-bit axe is run through twice, once for each blade. The timing is worked out on the same basis as the hardening furnace, that is to say, each weight axe has a definite time cycle.

When tempering $4^1/_2$-lb. double-bit axes, both points at one time, the length of time any one axe was in the bath was eight minutes. The tempering bath holds 26 double-blade axes, or 29 single-blade axes by the changing of the jig.

The two furnace operators are on piecework, each man turning out from 1400 to 1500 axes per day, with an average of 1435 each, for the past four months. The two men operating the electric furnaces are turning out a larger production than five heat treaters previously turned out on the fuel furnaces, are not rushed as they were in the past to average a certain wage per day, and have infinitely better working conditions.

The reasons ascribed by the Warren Axe and Tool Company for electrifying are dissatisfaction with results obtained with fuel-



General View of Bath-Type Furnaces Before Shipment

fired equipment, both as regards uniformity of product and the amount of labor required. Fool-proof equipment was wanted also.

A survey of the conditions prior to electrification as compared to the present results shows the extent to which these objectives have been attained and the economics of the investment in so far as is determinable to date.

Cost of Fuel Versus Electricity

There were no meters on the gas furnaces nor on the electric furnaces, hence accurate costs were not readily obtainable. A survey of the electric bills since the electric furnaces were placed in operation and a similar survey of the gas bills for six months previous to discontinuing the gas furnace furnished a basis for a sufficiently accurate check.

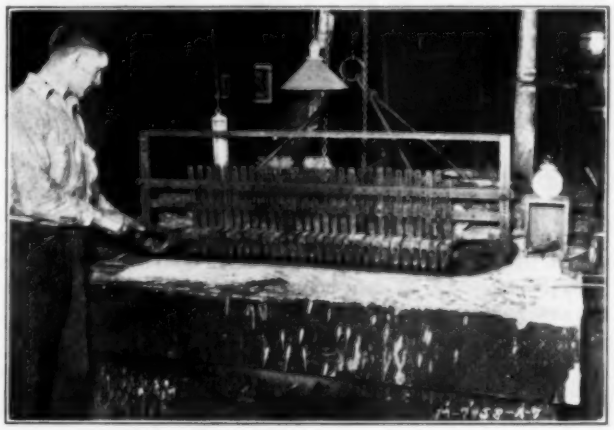
Natural gas was used, the price paid being 43 cents per 1000 cu. ft. The gas bills showed a reduction of approximately $150 a month since using electric heating. The date of reading the gas meter not coinciding with the date of beginning the operation of the electric furnaces, that month's reading had to be approximated and checked against production.

Electric power is purchased from the American Gas & Electric properties at an average rate of $0.02 per kw-hr. including the demand charge. The total demand for the entire plant, over a 15-min. continuous reading is 312 kw. (month of April). The power bills for the past nine months for the entire plant are as follows:

October	$665.63
November	690.26
December	678.87
January	687.32

Electric furnaces added:

February	$924.97
March	875.40
April	993.55
May	1018.98
June	967.08



Salt Tempering Bath, With 29 Single-Bit Axes in the Bath. Axes Other Than the Sager Brand Are Not Immersed Completely

The production was constant up to Mar. 1, 1929, when it dropped slightly and then increased again slightly above normal. Averaging the last four months prior to installation of the electric furnaces gives an average power bill of $680. Averaging the last five months of operation gives a power bill of $956. The difference in these bills being a reasonably accurate result, shows that the electric furnace costs $276 a month, making no deduction for the increased production of the past three months over last year.

With a gas cost of $150 a month and an electric cost of $276, the increased cost of electricity over gas fuel is $126 a month.

Labor Cost. Using the gas furnaces, five heat treaters were employed on piecework to get out the production. The average daily production per man for hardening and tempering was as follows:

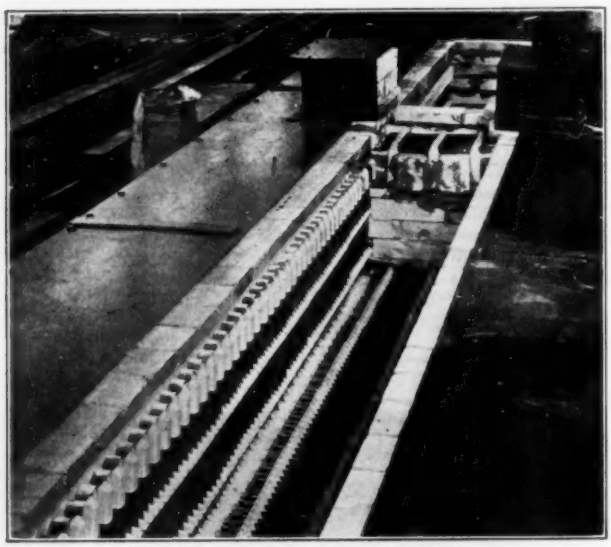
Common axes, single bit	400 per day
Sager axes, blue, single bit	325 per day
Common axes, double bit	200 per day
Sager axes, blue, double bit	165 per day

With electric furnaces, one man hardening and one man tempering, the combined production is as follows:

Sager axes, blue, single bit and Sager axes, blue, double bit and	1500 hardened 1500 tempered 750 hardened 1500 tempered

The direct labor cost per 100 axes hardened and tempered, using gas fuel, was as follows:

Single bit	$1.73 per 100 axes
Double bit	3.46 per 100 axes



Construction of Heating Chamber Using Ribbon-Type Elements

Since using electric furnaces, the scale has been adjusted so as to give the men the same wages in a seven-hour working day as formerly received in an eight-hour day. The direct labor cost per 100 axes hardened and tempered is as follows:

Single-bit axes	$0.85 per 100 axes
Double-bit axes	1.24 per 100 axes

On the basis of a production of 8320 double-bit and 25,350 single-bit axes the comparative labor cost is as follows:

Direct labor gas furnace operators:

83.20	×	$3.46	= $288
253.50	×	1.73	= 438
Total			$726

Direct labor, electric furnace operators:

83.20	×	$1.24	= $103
253.50	×	0.85	= 215
Total			$318

Average monthly saving in direct labor since using electric heat, $408.

Quality of Product. With gas heat, an attempt was made to hold the hardness between 53 and 57 Rockwell, "C" Scale (518 to 568 Brinell). With electric heat, the results are coming uniformly between 54 and 56 Rockwell (530–555 Brinell).

With gas heat, the difference in hardness of the inside and outside corners of the same axe bit frequently amounted to the low and high limits set. With electric heat, the difference is never more than 1½ points, Rockwell test.

Each axe is stamped with the year of manufacture. The principal outlet being the lumber or logging industry, approximately 50 per cent of the output is put into use almost immediately, and it is safe to assume that 90 per cent of the axes are in use six months from the date of manufacture. It is, therefore, possible to form a sufficiently accurate estimate of the period of manufacture corresponding to a given number of field replacements.

From the performance to date of the electric furnaces it appears certain that the field rejects due to heat treatment will be entirely eliminated. It is also certain that some of the rejects may be due to other causes, such as flaws in the steel and abuse of the axe. Since each axe is given a hammer test calculated to show up any flaws due to forging or welding, it is felt that but a very small percentage is due to this. To insure a high-grade uniform-quality product when using fuel furnaces, a considerable amount of added expense was necessary which has been eliminated in some cases and very materially reduced in others. The yearly value of this item in the reduction of testing, inspection, reduction in field replacements, etc., will amount to $8000 a year.

Improved Working Conditions. The two operators now on the electric furnaces had also been heat treaters on the gas-fired furnaces. One had been with the company 10 years, the other 15 years. An interview with these men disclosed their opinions of the relative difference in their working conditions before and after using electric heat.

With the gas furnace they worked eight hours a day, and had to work hard every minute of the day in order to get out the production to give them a certain wage. They had to be on the job practically every second, or they would have some overheated axes. The heat given off from the furnace was intense, and since they had to work directly in front of the open furnace all day, the working conditions, particularly during the summer months, were as severe as a person could stand.

With electric furnaces they work seven hours a day, and have plenty of time to do a good job, and in addition have from 1½ to two minutes rest out of every eight minutes. They are making as much money in seven hours as they formerly made in eight hours. There is no discomfort at all from heat, and they scarcely know or realize there is any heat in the baths. (The hardening bath, being covered with one inch of granular carbon, the surface of which is dead black, is, from all appearances, cold.) Instead of having the hardest and most uncomfortable job in the plant, they now have the easiest job, and just as comfortable as any. The electric furnaces have so improved their working conditions that it would be worth accepting less money rather than return to the old method.

Maintenance

Gas Furnaces. The maintenance of the gas furnaces was amazingly small, due to the simple construction of the furnaces, and to the fact that the heat-treaters supplied most of the labor at no cost, since they were on piecework. The records showed that an expenditure for materials amounting to $179 a year total, or $14.90 a month.

Electric Furnaces. No expenditures for repairs to the electric furnaces have occurred to date. The vessel of the hardening furnace depreciates in proportion to its hours of use, and there are other expenditures, such as for lead, granular carbon, and salt for the tempering bath, which must be considered as part of the maintenance. These figures should be of interest to those having applications for similar furnaces or as a comparison to results being obtained by means of fuel equipment.

Vessel Life. From data obtained from one installation having 24 large hardening and tempering baths, and from a number of other installations having a lesser number of baths, the average life of a nickel-chromium vessel, lead hardening, using electric heat, is from 5000 to 7000 hours. Tempering vessels have been in use for 13,000 hours with still no signs of depreciation. On the

basis of a vessel costing $800 installed in the furnace, and on a basis of 200 hours of use per month, the monthly depreciation is $23.20.

Lead Consumption. A total of 280 lb. of lead and 28 lb. of tin are used each month in the hardening bath to make up the loss.

280 lb. at $0.0875 =	$24.50
28 lb. at $0.2325 =	6.50
Total	$31.00

Granular Carbon. One ton of granular carbon will last nine months, and costs $50 delivered. Average monthly cost $5.60.

Salt Consumption. One barrel of tempering salt, of special mixture to Warren Axe and Tool Company's specifications, costs $75 and lasts from three to 3½ weeks. On the basis of three weeks, this amounts to $100 a month.

The monthly cost of maintaining the electric furnaces is as follows:

Vessel	$ 23.20
Lead	31.00
Carbon	5.60
Salt	100.00
Total	$159.80

Sales Value. It is the practice in most plants to take the direct labor cost and multiply this by a fixed amount to arrive at the total labor cost. The factory cost multiplied by a fixed amount gives the selling price. A reduction of $1 in direct labor may therefore permit a reduction of from $1.50 to $2 in the selling price. However, if in making this reduction in direct labor, a much higher quality of product is obtained, and at the same time, the trade distinction or appearance such as the Sager Blue is greatly enhanced, the sales value has been improved tremendously and if sold at the same selling price as before, gives a wider margin of profit. This added sales advantage can be estimated only, but on an estimated small value of one cent per axe, the average monthly value is $350, a figure considered very conservative by the management.

Investment Charges

Gas Furnaces. The investment tied up in the gas furnaces was relatively small, and in any event, since it had been charged off the books years ago, no monthly charge can be made against this equipment.

Electric Furnaces. The cost of the electric furnaces and control equipment, installed complete, was $9038.85. With an interest and depreciation charge of 15 per cent, the annual investment charge is $1355.83 or $133 a month.

COMPARISON OF MONTHLY OPERATING COSTS BETWEEN GAS AND ELECTRIC FURNACES AT THE WARREN AXE AND TOOL CO., WARREN, PA.

	Electric furnaces	Gas furnaces
Electric or fuel bill	$ 276.00	$ 150.00
Labor for heat treatment	318.00	726.00
Testing, inspecting, field rejects, etc.	167.00	667.00
Maintenance	159.80	14.90
Investment charge	133.00	
Total	$1053.80	$1557.90
Net actual book profit per month using electric heat		$504.10
Added sales value, per month		350.00
Total savings, per month		$854.10

Based upon the actual book profit shown for electric heat of $504.10 per month the equipment, completely installed, will pay for itself in 18 months, or a return on the investment of 66⅔ per cent. Such a situation is most gratifying to the Warren Axe and Tool Company, whose entire purpose in electrifying was the production of a uniform high-grade quality product, and who would have been satisfied had the operating costs increased by the same amount they have decreased, considering the quality of product now being obtained.

FSP-52-20

A Study of Electric House Heating

By A. R. STEVENSON, Jr.,[1] and F. H. FAUST,[2] SCHENECTADY, N. Y.

The use of oil and gas as fuels for domestic heating furnaces has been developed to avoid the inconvenient features of coal. Electricity as a source of heat has been given little consideration, particularly because of the idea that it is too expensive, although otherwise it is an ideal heat source. Two principal ways of using it are described—the direct conversion of electricity to heat, and the use of electricity as a source of energy to drive a heat pump. A plan is described based on using electric energy off peak in order to obtain low power rates and of storing the heat generated to be used as required.

IN THE attempt to overcome a number of inconvenient features in the use of coal, such as shoveling it into the furnace and taking out the ashes, various methods of using oil or gas have been evolved. Very little attention, however, has been paid to the use of electricity as a source of heat, principally because it has been considered too expensive, although in every other way it may be considered as a nearly perfect heat source.

It is hardly necessary to point out the cleanliness and convenience of electricity. Anything that can be said concerning the cleanliness of gas is obviously even more true for electricity.[3] With electricity, it is unnecessary to provide space for fuel storage; and by automatic control all labor can be eliminated.

In view of all these advantages it would appear that the use of electricity should be carefully investigated to determine its economic possibilities. It is the purpose of this paper not only to present the results of such an investigation or survey, but to show what has been done in the past and what is being done at the present in the way of electric house heating. No attempt will be made to discuss electric heating for industrial purposes, but attention will be given only to electric heating of homes and buildings.

There are two principal ways by which electricity can be utilized as a source of heat—the direct conversion of electricity to heat and the use of electricity as the source of energy to drive a heat pump. The direct conversion of electricity to heat may be accomplished in a number of ways, such as:

1 Using a metal resistance in an electric circuit to convert electricity into heat, which may be dissipated to the air directly from the resistance.

2 Using a fluid resistance such as water between two electrodes of an electric circuit and storing the heat in the water.

3 Using a copper tube as the secondary of a transformer. By passing water through the tube the heat formed in the tubing may be carried off in the water.

Any of these methods of directly converting electricity to heat may be accomplished at an efficiency of 100 per cent. If, however, electricity is used as a source of energy to drive a heat pump, as was suggested by Lord Kelvin as early as 1852,[4] it is possible to obtain an apparent efficiency greater than 100 per cent. For instance, a refrigerator is one example of a heat pump.[5] If it is driven by an electric motor, the heat output will be the sum of the heat equivalent of the electrical input and the heat pumped from the heat-absorbing element (the evaporator) to the heat-dissipating element (the condenser); consequently, the apparent efficiency, or the coefficient of performance of a refrigerating machine as it is called, is greater than one.

Since the use of a refrigerator as a heater has not been applied commercially, so far as we know, attention will be confined to the direct conversion of electricity to heat. As already mentioned, this conversion is consummated at an efficiency of 100 per cent, but it is possible in most localities to obtain electric power off peak at lower rates than on-peak power. By properly controlling the time of the conversion of electricity to heat, it is possible to obtain an apparent efficiency greater than 100 per cent compared with on-peak power costs. First there will be reviewed some of the methods of heating where the use of the electricity is not confined to off-peak periods.

The first method that suggests itself is resistance heating by radiation or convection from hot wires or grids. Such a method is extremely adaptable and flexible, because heating elements can be placed in every room and can be regulated manually or automatically to give any desired temperature. The heating elements are also of simple construction so that the first costs are low and there is very little that can go wrong.

In the usual types of radiant heaters a hot wire is placed in front of a polished metal reflector or a refractory material, and the heat is radiated to the walls of the room and to objects in the room. The advantage of this type of heater is that a quick heat may be obtained locally, and it is possible to feel warm in front of a radiant heater even though the air itself is cool.

The convection type of heater is usually operated at a much lower temperature than the radiant type, and it is not at all necessary that the heater be visible in the room to be heated. The heat from the hot wires warms the surrounding air, which sets up convection currents, thereby distributing the heat to the room. This method of heating gives a fairly even distribution of heat, but it has the disadvantage that the top of the room is heated first.

The method of heating water by passing an electric current from one electrode to another through the water has been used in Sweden, but it has had no practical application in this country.

The use of a copper tube as the secondary of a transformer to heat water has not been used practically. It is entirely feasible, although no great advantages over other methods are apparent.

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[1] Engineering General Department, General Electric Co. Mem. A.S.M.E.

[2] Refrigeration Engineering Department, General Electric Co.

[3] The economic value of cleanliness is illustrated by the following extract from "The Sky's the Limit," by Henry Obermeyer, Assistant to the Vice-President, Consolidated Gas Company of New York: "For example, the annual gas bill for the average family in Chicago is $30. Going back to our figure representing per capita cost of the smoke nuisance at $17 per year, the prorated annual cost to each family in Chicago is over $65 a year. The smoke nuisance therefore costs Chicago families on an average more than twice what they pay for gas. Chicago could pay over three times its present gas bill and still not be paying as much as its present gas bill plus the cost of the smoke nuisance."

Presented at the Third National Fuels Meeting, Philadelphia, Pa., October 7 to 10, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[4] "On the Economy of the Heating or Cooling of Buildings by Means of Currents in Air," by Lord Kelvin. Glasgow Phil. Soc. Proc., vol. 3, December, 1852.

[5] There are other kinds of heat pumps, such as a thermocouple with a cold and hot junction supplied with a direct current, and an arrangement of batteries maintained at different acid concentrations so that heat is transferred from one location to another, but neither of these methods is at all practical. See "Energy Problems in Electric Heating," by Otto Gunwolt, *Graz ETZ*, Sept. 27, 1928, vol. 39, pp. 1437–1441.

Another method of resistance heating involves the heating of water by immersion heaters. This method is mainly used in off-peak heating systems, and it will be discussed in that connection.

Unfortunately, resistance heating with present on-peak power rates is far too expensive to interest the majority of people in this country, although it is recognized that under special circumstances the advantages of electric heat would outweigh the higher cost. It is understood that in England several department stores[6] have been equipped to heat with electric-resistance heaters, but they have been able to get a rate of 1 cent per kw-hr., which is considerably lower than the present rates in this country for on-peak power to be used in homes. It is interesting to note that, although the actual cost of electricity at 1 cent kw-hr. was higher than the actual cost of coal, oil, or gas, the decision to use electricity was arrived at solely from financial considerations after the factors of cleanliness, depreciation, handling charges, convenience, etc., had been taken into consideration.

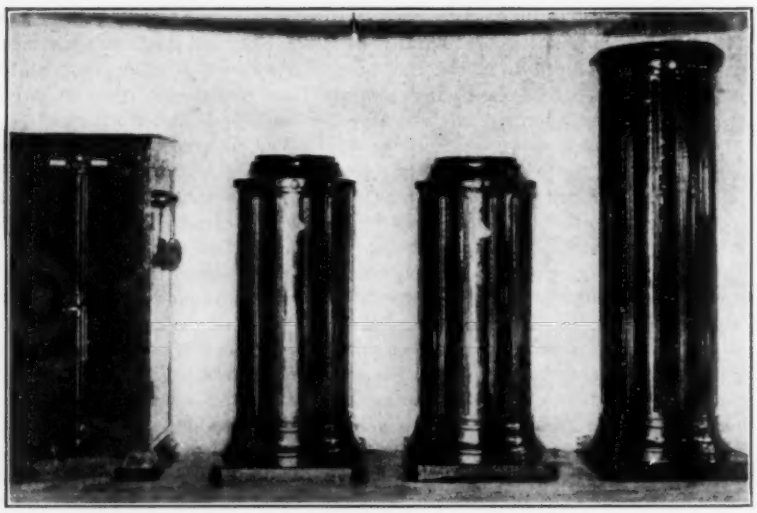


FIG. 1 THE FOUR STORAGE HEATERS USED IN THE CORK-INSULATED HOUSE

In this country, electric resistance heating has been installed in a few localities where power is cheap, principally along the Pacific Coast. It is a matter for speculation whether or not much progress will be made with this method of heating unless a rate of approximately 1 cent per kw-hr. for on-peak power should be established. Such places as gasoline filling stations, small stores, lodgerooms, clubrooms, churches, waiting rooms, etc., could probably use this form of heating most economically.

Within the past couple of years a number of experiments have been conducted to determine the feasibility of using electric energy off peak in order to obtain low power rates and of storing the heat generated to be used as it is required. The theory behind the use of off-peak power is as follows: The basic cost of electricity consists of fuel, labor, supplies, and investment costs. At low per cent load factors—i.e., when there are large peak loads and low average loads—the basic cost per kilowatt-hour is high. If an off-peak load can be introduced so that the off-peak demand never exceeds the on-peak demand, no additional investment is required and this item is excluded from the cost of generation of the electrical energy.[7] Consequently, the utility companies can afford to give a very much lower rate to customers using energy off peak.

There have been a number of actual installations of off-peak heating systems made recently. The results of a few experiments on off-peak electric heating and the results from a number of commercial installations were described in recent articles.[8]

Considerable work has been done in both an experimental and a commercial way by a new company[9] which was organized about a year ago to engage in the engineering, design, research, and sales of electrical devices for the heating of buildings, both of commercial and domestic forms, and for the heating of water for domestic purposes. Part of its program consists in cooperation with the electric utilities for the improvement of load factor and for the purpose of obtaining off-peak power rates sufficiently low to make electric heating economically justified.

One type of off-peak heating system developed in Germany and described by E. B. Dawson and I. F. Lamb[9] uses pea gravel as the storage medium. This type of heater was designed to be placed directly in the rooms to be heated. It apparently operated satisfactorily except that in the comparatively warm weather in the spring and fall it was difficult to keep from storing too much energy in the heaters. Fig. 1 shows the appearance of several of these heaters, and Table 1 gives some information from a German leaflet describing them.

The Hall Electric Heating Company has developed several systems of heating with off-peak power, and in the short time since it was organized it has successfully installed a number of various types of heating systems. These were in operation during the whole or part of last winter.

TABLE 1 STORAGE HEATERS IN INSULATED HOUSE

Rating, watts	Dimensions in inches. Height	Diameter	Net weight without gravel, lb.	Approximate list price[1]	Weight of gravel required, lb.
3000	51.7	23.6	353	$118	320
5000	71.5	23.6	467	157	485

[1] Based on an exchange rate of 23.80 cents per mark.

This company's system of off-peak heating consists of insulated water-storage tanks, electric immersion heaters, heat-transfer surfaces, and suitable control devices. The electric immersion heaters are automatically turned on during the off-peak period of the system, and the heat is stored in an insulated tank at 250 to 300 deg. fahr. The heat from the storage tanks is supplied thermostatically to the house as it is needed, either by circulating the hot water from the storage tank to conventional

[6] A very good description of one of these installations is given in a paper, "The Electrical Heating and Ventilation of Bourne & Hollingsworth Premises," by Arthur Henry Barker; excerpt minutes of the Proceedings of The Institution of Civil Engineers, vol. 22, 1928–29. "A Large Heating, Cooling, and Ventilating Scheme," the electrical installation which is running satisfactorily at the premises of Messrs. Bourne & Hollingsworth, *Electrical Review*, January 11, 1929.

[7] This is not quite exact, on account of the fact that the service connection with the transformer and sometimes a small portion of the primary system must be increased in capacity, if it is inadequate. Of course, the generating and main distribution system would not be involved at all.

[8] (1) "Electrically Heated Houses," by E. B. Dawson and I. F. Lamb, *Electrical World*, part I in Jan. 12, 1929, issue, pp. 91–96, and part II in Feb. 23, 1929, issue, pp. 389–395. (2) "A Customer's Experience With Electric House Heating," by V. M. Nussbaum, *Electrical World*, March 2, 1929. (3) "Off-Peak Electric Heating of Buildings," *Electrical World*, June 1, 1929, vol. 93, pp. 1105–1108. (4) "Sell Electric Heating to Banish Chimney and Coal Bin," *Electrical Record*, July, 1929, vol. 46, pp. 65–66.

[9] Hall Electric Heating Company, Philadelphia, Pa.

radiators in the rooms or by circulating air by forced draft through ducts to a finned-tube radiator placed near the storage tank.

The length of the off-peak period is determined by the characteristics of the load on the power station, but it usually lasts from ten to twenty hours and usually at night. In Fort Wayne, Indiana, for example, the power company allows customers to draw electricity at reduced rates from 9 p.m. to 7:30 a.m. and from 12 m. to 1 p.m. The off-peak period is determined by the power company, and it sets the control device in the heating system to give the desired results. The control contains a Telechron clock mechanism, which requires no winding, so that the heating system is entirely automatic.

Fig. 2 shows a typical warm-air installation by the Hall company in a home of average size, and Fig. 3 shows the interior construction of the warm-air heating system. It is claimed that it is possible to use only four warm-air pipes in this system as contrasted with twelve in a furnace installation. Also, owing to the forced circulation of air, no pitch is required, thus giving maximum headroom.

When a house is being constructed, a system can be adapted as shown in Figs. 4 and 5, thereby eliminating the necessity of a basement and reducing the duct work to a minimum.

Knowing the cost of fuel or electricity at any place and knowing the efficiency of the conversion into heat, it is possible to compute the amount of heat energy that can be obtained for a given sum of money. Of course, the price of fuel varies greatly

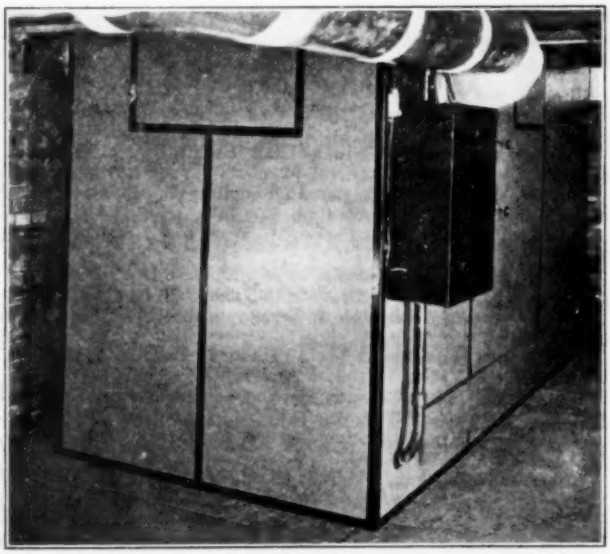


FIG. 2 TYPICAL HALL COMPANY WARM-AIR INSTALLATION

from place to place, and the efficiency of conversion into heat is never constant from day to day even in the same plant. The efficiency of conversion is influenced by the type and size of furnace, the frequency and method of firing, the size and cleanliness of the chimney, the weather, and many other factors, so that any estimate that is made will be subject to considerable variation.

Such figures indicate that electric off-peak heating systems using electric energy at 1 cent per kw-hr. are usually somewhat more expensive than coal or gas, from a heat energy standpoint. However, in considering this matter further, perhaps it may be shown that it is possible to make electricity more attractive from a cost standpoint.

When the Hall Electric Heating Company installs a heating system, it endeavors to do or to have done the following things:

1 Insulate the house with some form of insulating material and add storm doors and windows.

2 Automatically control the temperature in the house so that the temperature never becomes excessive at any time.

3 Reduce the temperature from 70 deg. fahr. in the day time to 55 deg. fahr. in the night time.

4 Instruct the occupants of the house to keep doors and windows closed as much as possible and to shut off bedrooms from

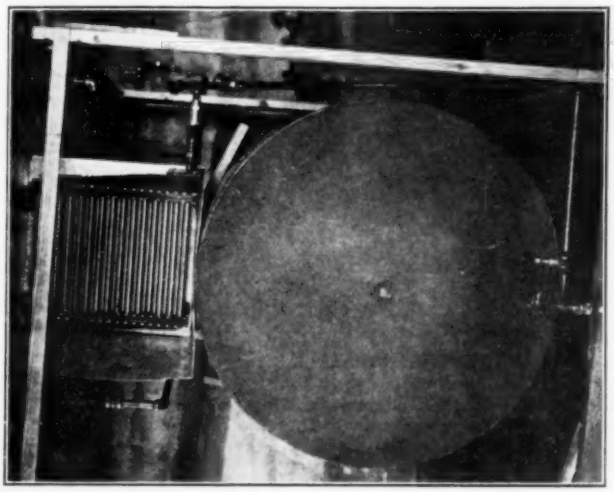


FIG. 3 INTERIOR CONSTRUCTION OF WARM-AIR HEATING SYSTEM (Storage tank, radiator, expansion tank, and sheet-metal work shown.)

the rest of the house when bedroom windows are opened during the night.

By thoroughly insulating a house it is possible to reduce the heating requirements by one-half, so that with the other savings effected the total cost of electricity for an off-peak heating system is not greatly in excess of the cost of coal, and perhaps somewhat less than the cost of gas. Of course, it may be reasoned that these savings could be applied to other heating systems and that consequently the electric method is unfairly favored. The fact remains, however, that in the majority of cases none of these savings would be effected with the usual type of heating system; therefore, if the company which installs electric heating can effect these savings, at least some part of the savings should be credited to the electric system.

The fact is or should be well established that the cost of heat energy is not the only factor entering into the operating cost.

When coal is used, a charge should be allowed for as handling and for labor involved in firing the furnace. The soot and dirt from a coal-burning installation will make it necessary to refinish the house and clean the draperies more often, and this should be taken into consideration. The factors of convenience, cleanliness, and reliability are important but hard to evaluate.

Besides the operating charges mentioned, there are the fixed charges, which include depreciation, interest on the investment, heating-plant insurance, and taxes.

With the electric method of heating the following advantages are claimed:

1 Absolute cleanliness throughout the house, as there are no products of combustion.

2 Absolute safety.

3 Complete automatic control.

4 Quick heating. It is said that the house can be raised from 55 deg. to 70 deg. at a predetermined time in the morning in about fifteen minutes.

5 No fuel storage.

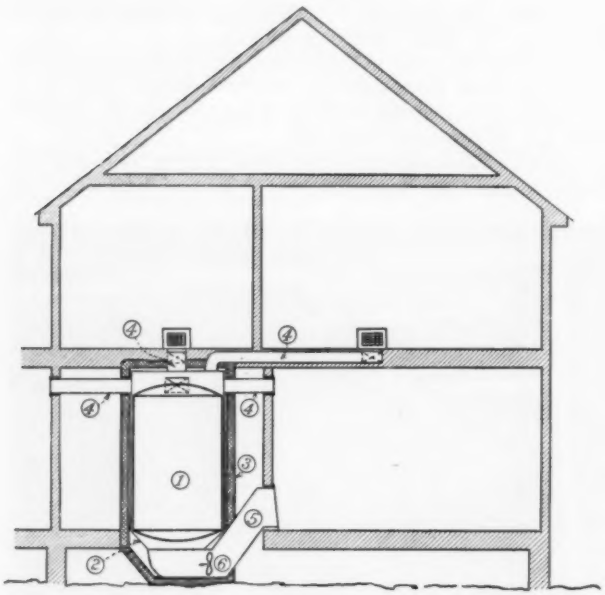


FIG. 4 ELEVATION DIAGRAM OF HOME OF SPECIAL DESIGN FOR APPLICATION OF ELECTRIC-HEATING WARM-AIR SYSTEM

(1, heat-storage tank; 2, sheet-metal casing to provide for passage of air over tank; 3, insulation inclosure around storage tank; 4, hot-air ducts to different rooms; 5, cold-air intake; 6, fan for positive circulation of heat.)

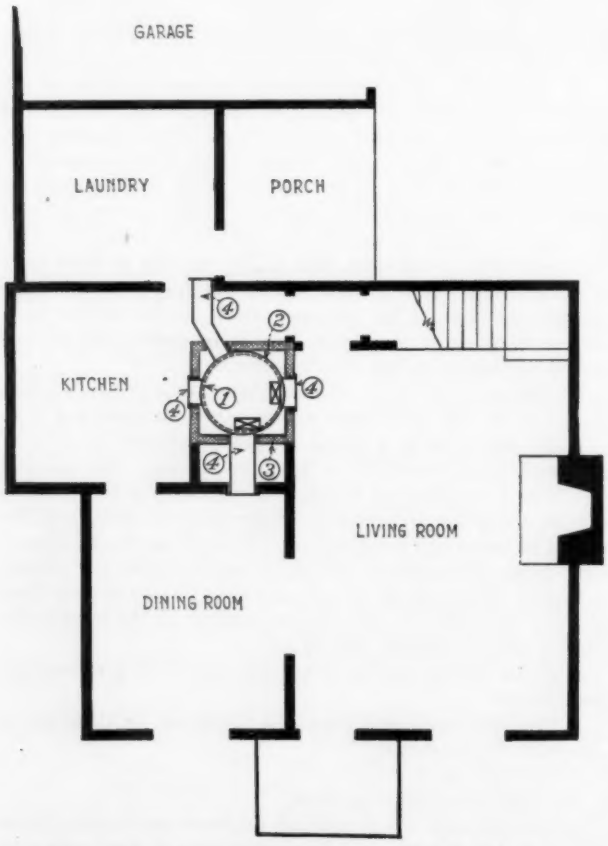


FIG. 5 PLAN VIEW OF HALL COMPANY HEATING SYSTEM

(1, heat-storage tank; 2, sheet-metal casing to pass air over tank; 3, insulation inclosure around storage tank; 4, hot-air ducts to different rooms.)

6 No labor required to operate the heating system.

7 Since electric heat uses no oxygen from the air and since there are no products of combustion, it should be the most healthful form of heat.

It is not claimed that off-peak electric heating is going to revolutionize present methods of heating with coal, oil, or gas. Even though this method were cheaper than all other methods, there is the limitation that a power company can supply only a limited amount of off-peak energy because beyond that amount the energy becomes on peak. This is easily illustrated by an example. The average utility company will have a load factor[10] of approximately 45 per cent. Any energy sold off peak which creates a peak higher than the normal peak cannot be sold economically at off-peak power rates. It is estimated that about 2 per cent[11] of all the wired homes could be equipped with off-peak electric heating systems before the load factor reached its maximum practical value. This gives a potential market of about 400,000 installations in the United States based on present statistics.

It is recognized that electric heating is more expensive from a heat-source standpoint than other methods of heating, but in view of the extra advantages obtained, it would appear that it should be possible to sell a large number of this type of heating system.

Off-peak electric heating of homes is new in the United States, although it has been used for a number of years in several European countries. On-peak electric heating has had a limited application in this country, although in one or two localities it has been used extensively.

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Discussion

CHESTER I. HALL.[12] The heating of our homes and our workshops has been a pressing need for a great many years.

[10] Yearly load factor is the ratio of the average load in a year to the peak load in a short interval of time, usually fifteen minutes.

[11] This is a hard figure to determine accurately. Others who have studied this subject thoroughly believe that 5 per cent is a more representative figure. The smaller figure is used in order to be ultra-conservative.

[12] President, Hall Electric Heating Company, Inc., Philadelphia, Pa.

Throughout history the various stages of development of civilization have been marked by significant changes in the home-fire.

The use of fire for its heat-giving qualities started with the use of the campfire back at such an early date that its origin is lost in antiquity. Even at a later date when more permanent homes began to be built, the fire was built in the middle of the room with only a hole in the roof to allow the smoke to escape.

Through ages the heat-giving fire has been merely transferred in succession from one type of primitive home to the next through a long period of gradual improvement in the home-building art. The introduction of the fireplace in England marked an important stage in the effort to improve home-heating facilities. Improvements in fireplace heating continued until in 1774, when Franklin invented the iron stove, which marked another era in the history of fire in the home.

The next major development marked the moving of the fire to the basement to eliminate soot, smoke, dirt, and obnoxious gases in an effort to make the home more healthful and more attractive. Most of the efforts of our present day have been devoted to the improvement of the basement heater, and have resulted in the development of the modern warm-air, hot-water, and vapor systems, all burning fuel in the basement, and still unsatisfactory from the standpoint of efficiency, cleanliness, safety, healthfulness, and ease of operation.

Along with the advent of the telephone, the radio, the automobile, the airplane, and many other modern devices have come the many appliances and developments which have improved the living conditions and removed the drudgery from the home. But our heating systems have not kept pace with this development. Raw fuel is still delivered to the home and burned within the house, even while industrial plants are eliminating the furnace and the boiler in an effort to obtain improved conditions in our workshops.

But a new era in the art of home heating is dawning. The recent development of a new system of electric heating, using cheap off-peak energy by means of a storage principle, is destined to free us of the primitive practice of combustion in the home, and we now have available heat in the fall by the simple snap of a switch, perfectly controlled by a small, rugged, attractive, accurate thermostat, placed on the wall in the living room, and there is no combustion, no soot, no dirt, no insidious gases, and no elimination of the health-giving oxygen and other air vitamins, no cares of coal and ashes. The pure, unadulterated heat in the electrical form is delivered to the home by wire from the waterfalls or the base of coal supply.

H. S. Vassar.[13] The authors of this paper have clearly indicated the two principal methods of applying electricity to domestic heating; i.e., (1) the direct conversion of electricity to heat, and (2) as energy to drive a heat pump. Somewhat naturally they have dismissed the second method with but a single paragraph because no commercial application of the principle as applied to domestic heating has been made up to this time. I also note that their bibliography contains no reference to the experimental work of a French engineer[14] tending to indicate not only the practicability of such an application but also a coefficient of performance, or apparent efficiency, so far in advance of the direct conversion method as fully to warrant thorough investigation.

Such a development might be based on the two following well-established facts: (1) That we are surrounded by an unlimited supply of heat which requires only a change in thermal level to make it available for man's use, and (2) that it is possible to "pump" heat from a lower to a higher temperature level.

The fact that the ideas on house heating advanced by Lord Kelvin so many decades ago, up to now have not been commercially applied, in itself indicates nothing. A similar period elapsed after the beginnings of mechanical refrigeration before there was a great flood of domestic refrigerating machines by which we are now surrounded.

If it is practicable to pump heat from a refrigerator, and discharge that heat against the higher temperature of the room in which the refrigerator is located, why is it not equally practicable to pump heat from outdoors and discharge it into the house? If the present limitation is simply one of apparatus, will not intensive development remove that limitation?

The French engineer already referred to states that it is theoretically and practically possible to obtain upward of 16,000 B.t.u. by the expenditure of one kilowatt-hour when used as energy to drive a heat pump, as against the maximum of 3145 B.t.u. obtainable by direct conversion, a ratio of more than four to one in favor of the heat pump principle.

Until disproved, this statement in itself should be sufficient to stimulate development along these lines, particularly in view of the fact that if suitable equipment should be designed to apply the heat-pump principle to domestic heating, it also could be used for house cooling.

One of the authors of this paper in a recent address before the Franklin Institute, Philadelphia,[15] made the following statement:

"Approximately four times as much heat on the average can be delivered to a house by a refrigerating machine pumping heat from outside than by using the same amount of electric power in resistance heating units."

Charles Lamb in one of his essays tells the story of some early experiments in the application of heat to domestic problems. Perhaps in years to come we may look back upon the direct conversion of high-grade energy into heat with something of the amusement with which we view Ho-ti and his neighbors burning down a home when they feel a yearning for roast pig.

Authors' Closure

This paper did not discuss in detail the use of electricity as energy to drive a heat pump for the reason stated that no commercial application has been made as yet. H. S. Vassar is quite right in saying that development of this principle should not be retarded or governed because of the fact that the principle was stated by Lord Kelvin as far back as 1852, just 77 years ago.

About a century ago Carnot stated the following law:

$$C = \frac{T_1}{T_2 - T_1} = \frac{\text{Refrigerating effect of } T_1 \text{ deg.}}{\text{Work to pump the heat to } T_2 \text{ deg.}}$$

where C is known as the Carnot coefficient of performance and the temperatures are expressed in degrees absolute.

Assume the condition of a refrigerator operating as a house heater when the house temperature is 70 deg. fahr. and the outside temperature 25 deg. fahr. In order to transfer heat it is necessary to have a temperature differential, hence assume the condenser at 100 deg. fahr., and the evaporator at 10 deg. fahr. The coefficient of performance is then

$$C = \frac{T_1}{T_2 - T_1} = \frac{459.6 + 10}{100 - 10} = 5.22$$

To this may be added the motor input, which will make the

[13] Bloomfield, N. J. Mem. A.S.M.E.

[14] "The Practical Possibilities of Using Energy for the Combined Production of Heat and Refrigeration," by M. Lebre, Society of Civil Engineers of France, vol. 80, no. 5, May, 1927, pp. 669–708.

[15] "Refrigeration," by A. R. Stevenson, Jr., Journal of Franklin Institute, August, 1929, p. 185.

ratio of heat delivered to the heat equivalent of the electric input equal to 6.22. It must be remembered that this ratio can only be obtained by an ideal machine, and cannot be obtained in practice. Assuming a reasonable efficiency it is apparent, however, that the ratio of heat delivered to the house will be considerably more than the heat equivalent of the electrical energy supplied.

The use of a refrigerator to heat a house is possible today with existing equipment, but it will not be commercially successful until the many practical difficulties are overcome. The principal reasons for the failure to use refrigerating machines as heaters may be noted as follows:

1 No suitable refrigerating machine with the necessary characteristics of quietness, lack of vibration, and freedom from service has yet been developed in large enough sizes (5 hp. to 25 hp.).

2 The cost of electricity in most localities has been too high until recently to make this method of heating compare favorably with existing methods. Perhaps also we have all been somewhat blind to the economic possibilities already here with the gradual introduction of lower rates for electric ranges, hot-water heaters, and domestic refrigerators.

3 Special precautions would have to be taken to prevent the refrigerant, in case of a leak, from getting into the ventilating system, and thus rapidly contaminating the air of the whole house, unless apparatus can be developed to utilize a really harmless refrigerant.

4 The development of a complete system for using a refrigerating machine both for heating and cooling is a task involving so many problems that it cannot be undertaken lightly.

Although some of the difficulties enumerated above appear formidable, it is probably not impossible to overcome them. It seems probable that the limitations of this type of heating and cooling system could be overcome by intensive development as Mr. Vassar suggests. However, on account of the fact that the coefficient of performance of a refrigerating machine decreases with increase in temperature differential between the evaporator and condenser, this system of house heating is not as attractive in the north as in the south.

FSP-52-21

Liquefied Petroleum Gases as Industrial Fuels

By R. W. THOMAS,[1] BARTLESVILLE, OKLA.

THE OBJECT of this paper is to show (1) the general availability and the assured continuity of supply of large quantities of liquefied petroleum gases, (2) the adaptability of these fuels for exacting industrial processing such as heat treating and other operations of a highly refined nature, and (3) the advantages and limitations of liquefied petroleum gases in industrial fields, particularly from the economic viewpoint.

Ideal materials are ofttimes regarded as imaginary, fanciful, or impracticable, but however nebulous most so-called ideal commodities may be, all engineers recognize certain characteristics of an ideal fuel.

Such a fuel must be gaseous so as to be easily confined, transported, and controlled, and to permit a maximum quantity of heat to be released in a minimum combustion space in the shortest length of time. It must be free from foreign or deleterious substances and from corrosive constituents. It must have combustion characteristics that adapt it to the multitudinous demands of hundreds of industries and thousands of processes. It must be uniform and unchanging in composition to insure like results under like conditions. It must be generally available and in practically unlimited quantity, regardless of the point of use. It must possess flexibility and universal adaptability, and it must be economical in use. Such a fuel will naturally produce results superior to those obtained with ordinary fuels, and will therefore find a large and honored place in industry.

Practically every one will concede that natural gas most nearly fills these specifications. However, it is not generally available, the supply is not always adequate, and impurities such as hydrogen sulphide are ofttimes present.

We must then search further for a fuel approaching this ideal. Coal has availability, distribution, and economy in its favor, yet one recognizes its limitations for the higher classes of thermal processing, upon calling to mind the enormous investments and the high capital and operating costs involved in converting it to coke, to coal gas, water gas, and other more adaptable and therefore more valuable classes of fuel. Producer gas has its serious limitations, coal-gas production lacks the flexibility demanded by industry, and water gas not only presents the carbon-monoxide hazard but usually requires carburetion. All of these gases are inherently variable in composition, specific gravity, and calorific value, although these handicaps are minimized by judicious mixing in holders or by frequent adjustment of the combustion apparatus in use.

Oil is generally available and is usually more economical per unit of heating value than the higher forms of fuel evolved from coal, but it suffers by comparison with the ideal. It is not gaseous, it requires a large volume of combustion space, it is variable in quality, it usually contains sulphur compounds, and its precise control offers difficulties.

Liquefied Petroleum Gas

A secondary product originating in natural gas, and in "casing-head gas" produced with crude oil, is now attaining prominence because it fulfils many of the specifications of the "ideal" fuel. It produces large-scale commercial results equal to and often better than those obtained with natural gas in every respect save that of cost.

Origin. There are certain of the paraffin hydrocarbons that occupy that zone intermediate between the so-called permanent gases and the liquids, and which carry the generic name "liquefied petroleum gases." These are too volatile, if liquefied, to be of much practical value when included in the motor-fuel fraction. If allowed to remain in natural gas in quantities, they are likely to cause difficulty by condensation under high pressures and low temperatures. Their very physical properties which make them undesirable as components of gasoline or of natural gas are capitalized when the various fractions are segregated, as their transportation in concentrated form as liquids in pressure containers is followed by their advantageous utilization as true gases.

TABLE 1 PRINCIPAL PHYSICAL AND CHEMICAL PROPERTIES OF COMMERCIAL PROPANE AND BUTANE

	Lb. per sq. in. gage	
Vapor pressure	Propane	Butane
at 70 deg. fahr	120	33
at 90 deg. fahr	165	53
at 100 deg. fahr	195	65
at 105 deg. fahr	210	71
at 130 deg. fahr	300	110
Temp. deg. fahr. at which press is 0 lb. per sq. in. gage	−44	15
Sp. gr. of liquid (water = 1)	0.509	0.576
A.P.I. gr. of liquid deg. A.P.I.	146.4	114.2
Initial boiling point, deg. fahr.	−45	12
Final boiling point, deg. fahr.	−40	30
Weight per gal. of liquid	4.24	4.8
Mean coef. of thermal expan.		
From 0 to 50 deg. fahr.	0.001316	0.000908
From 50 to 100 deg. fahr.	0.00174	0.00118
Sp. gr. of gas (air = 1)	1.523	1.95
Sp. heat of vapor at 60 deg. fahr. (C_p)	0.475	0.458
Dew point at 14.7 lb. absolute, deg. fahr.	−44	26
Cu. ft. of gas per lb. liquid	8.49	6.7
Cu. ft. of gas per gal. liquid	36	32
Limits of inflammability		
Gas per cent in gas-air mixtures		
Lower explo. limit	2.4	1.9
Max. rate flame propagation	4.7	3.6
Upper explo. limit	9.5	8.5
Max. rate of flame propagation cm. per sec. in 2.5 cm. dia. tube	82.4	82.5
Heating value		
B.t.u. per cu. ft.	2,550	3,200
B.t.u. per lb.	21,650	21,500
B.t.u. per gal.	91,800	102,400
Latent ht. of vaporization at b.p.		
B.t.u. per lb	186	170
B.t.u. per gal.	788	830

Properties. Chemically speaking, propane and the butanes (normal and iso-butane) constitute the predominant compounds in all of the definitely established grades of liquefied petroleum gas. Practical consideration of their properties has divided these products into two classes, the "propane" and the "butane" grades. There are shown in Table 1 the important properties of these two commercial grades of liquefied petroleum gas referred to as "propane" and "butane."

The properties shown in the table are those found by actual determination and careful research of these two products. For all practical purposes these grades can be considered as commercially pure propane and commercially pure butanes. Each of them is exceptionally uniform in composition and quality. The study of the properties of the products has been so complete that the behavior of either of the grades under the wide variety of conditions encountered in actual usage can be predetermined with a degree of accuracy much higher than that found in usual

[1] Vice-President and General Manager, Philfuels Company.

Presented at the Third National Fuels Meeting of the A.S.M.E. at Philadelphia, Pa., Oct. 7 to 10, 1929.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

engineering calculations. Fig. 1 shows graphically the temperature-pressure relations of several commercial grades.

Potential Supply. Propane and butanes are found abundantly in natural gases from gas wells and oil wells and are, of course, also present in large quantities in refinery gases resulting from oil-distillation processes. The natural-gasoline industry, however, constitutes the only source of liquefied petroleum gases worthy of consideration at present. This is due to the fact that the natural-gasoline industry has always disposed of the major portion of the volatile fractions, suitable for liquefied petroleum-gas manufacture, in motor fuel. There is a definite limit to the percentage of this volatile material which can be absorbed by motor fuel.

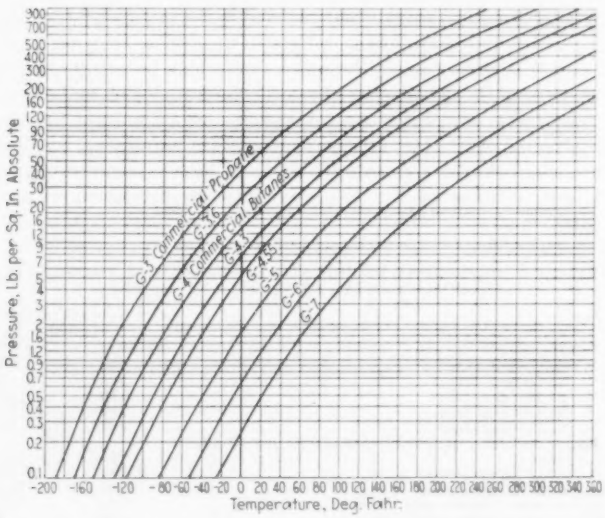


FIG. 1 VAPOR-PRESSURE-TEMPERATURE RELATION OF LIQUEFIED PETROLEUM GAS

The increased use of oil-cracking processes by the refiners is known to favor a demand for still more stable grades of natural gasoline, thereby augmenting the supply of liquefied petroleum gases available from the natural-gasoline industry. The average motor fuel contains 0.4 per cent of propane and 4.0 per cent of butanes. The average daily consumption of motor fuel during 1928 was approximately 38,500,000 gal. This means that the motoring public last year daily consumed about 1,700,000 gal. of liquefied petroleum gas in its motor fuel. Statistics indicate that as a very conservative estimate there is a daily potential production of at least 13,000,000 gal. of liquefied petroleum gases from the natural-gasoline industry. However, this large supply is not immediately available. To begin with, the design and operation of the average natural-gasoline plant are such that only a portion of the potential supply is extracted along with the raw natural gasoline. Another consideration in regard to the availability of liquefied petroleum gas lies in the fact that its production is accomplished more economically at the larger natural-gasoline plants. The amount of liquefied petroleum gas removed from the raw natural gasoline in preparing it for the market is about 1,500,000 gal. daily, of which about 50 per cent is propane. Probably 50 per cent of this immediately available supply exist at plants too small economically to justify equipment necesssary for the further separation of the liquefied petroleum-gas products. This means, then, that there is available a supply of about 750,000 gal. daily which will undoubtedly find markets at existing competitive prices. When economic conditions justify, the supply of liquefied petroleum gases will be increased in proportion to the price level of the commodity. Table 2 presents some pertinent data covering the growth of the natural-gasoline industry, and gives a picture of the rapidly increasing supply of the raw material from which the liquefied-petroleum gases are segregated.

TABLE 2 DATA REGARDING THE NATURAL GASOLINE INDUSTRY

Year	Gas treated billion cu. ft. (1)	Gasoline produced, million gal. (2)	Average yield per M cu. ft. of gas (3)	Per cent of natural gasoline to total motor fuel manufactured (4)	Gallons of natural gasoline per bbl. of crude oil produced (5)
1911		7			0.03
1912		12			0.05
1913		24			0.10
1914		43		3.4	0.16
1915		65			0.23
1916		103			0.34
1917	429	218	0.51		0.65
1918	449	283	0.63	7.4	0.79
1919	480	352	0.74	8.3	0.93
1920	496	385	0.78	7.4	0.87
1921	465	450	0.96	8.1	0.95
1922	545	506	0.93	7.6	0.91
1923	876	816	0.93	10.0	1.11
1924	1016	934	0.92	10.0	1.31
1925	1040	1127	1.08	9.9	1.45
1926	1206	1363	1.13	10.30	1.77
1927	1341	1641	1.22	11.51	1.82
1928		1776		10.92	1.98

Manufacture. Liquefied-petroleum gases are manufactured from the "wild" raw natural-gasoline feed stock by the application of the well-known principles of rectification to large-scale production, resulting in accurate separation of the original components. Fig. 2 shows three liquefied-gas rectifying columns.

Uses. The known commercially practical uses of liquefied petroleum gases can be briefly summarized as follows:[2]

1 Domestic fuel transported in cylinders, thus supplying gas to districts not having that service. This use is a valuable ally to the gas companies as outlying districts can be serviced and held against encroachment by electricity or other fuels, pending gas main extensions. Commercial propane, being self-vaporizing at all temperatures above minus 40 deg. fahr., is the most applicable and most widely used fuel for this "bottled gas" service.

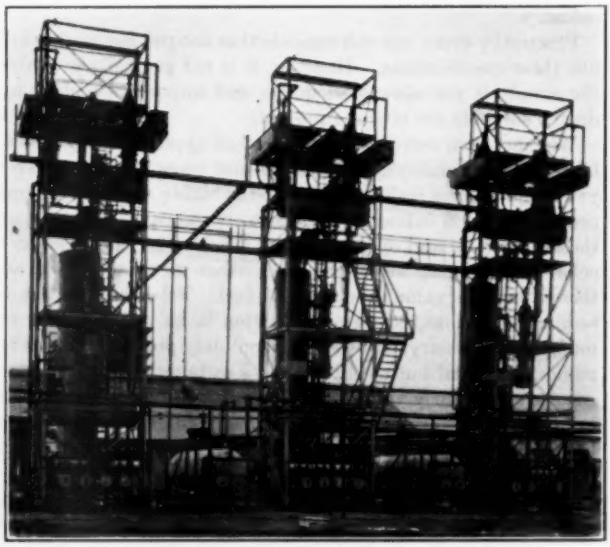


FIG. 2 RECTIFICATION COLUMNS MANUFACTURING LIQUEFIED PETROLEUM GAS

2 Industrial gas for various heating operations when the fuel consumption is large, and where an economical but clean and easily controlled fuel is necessary. This phase of utili-

[2] R. W. Thomas and O. M. Setrum, "Commercial Propane and Butanes in the Manufactured Gas Industry," p. 1284, American Gas Association, 1928 Proceedings.

zation, unless the marketing is controlled by the utilities themselves, is in direct competition with the industrial-fuel sales efforts of the gas companies. Commercial butanes are best adapted to this exacting large-scale use.

3 Enricher for manufactured gas in place of gas oil to carburet producer gas, blue-water gas, or other gases of relatively low thermal value to the required calorific standard. The efficiency of "cold carburetion" is 100 per cent. Commercial butanes are most applicable here, due to their favorable physical characteristics.

4 Base material for manufactured gas.

a Carbureting gases of little or no calorific value, these acting as diluents or carrying mediums. Exhaust or flue gases, or air are the materials commonly used

b Reforming or cracking high-calorific-value liquefied petroleum-gas vapors to produce low-density gaseous fuel of the desired B.t.u. content. Although the use of any of the standard grades is practical for either of the foregoing operations, nevertheless the commercial butanes offer outstanding advantages and economies.

5 Commercial solvents.

6 Raw materials for the manufacture of chemical compounds.

Industrial Fuel Uses

Grade. Consideration of the physical properties of commercial butane indicates that this grade is better adapted for economical transportation, storage, handling, and use than is the case with the more volatile propane fractions. Butane is favored with a low vapor pressure, averaging from 10 to 50 lb. gage within the usual atmospheric temperature range, and likewise has an advantage over propane in a higher concentration of heating value per gallon of liquid or per cubic foot of vapor. Only the butane grade will be considered herein, and the word "butane" will describe the commercial grade rather than the chemically pure compound.

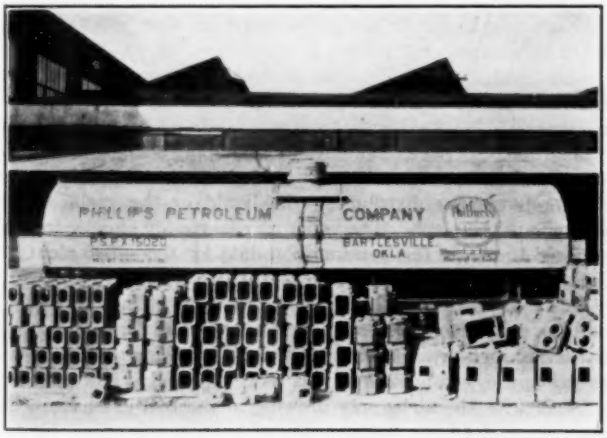


Fig. 3 Class 104-A Tank Car for Butane Transportation

Transportation. Butanes are shipped in class No. 104-A tank cars of special heavy construction to withstand the specified 100-lb. test pressure. The safety valves are set at 75 lb., and the cars are insulated with 4 in. of cork lagging covered by a metal jacket. Fig. 3 shows such a car. It will be noted that no bottom outlet connection, no dome, and no removable dome cover are provided. All fittings are mounted inside a protective housing on a cover plate bolted to a nozzle at the top of the car. Outage or expansion space is left in the tank when the car is loaded. The ordinary shell capacity is 10,500 gal., and about 10,000 gal. or 48,000 lb. are actually shipped. Unloading is provided for by eduction pipes extending from the valves on the top cover plate to sumps in the bottom of the shell. All outlets are equipped with upward-seating check valves which close when the rate of flow exceeds a safe point, thereby insuring against loss of contents in case the dome valves or fittings are damaged or unloading connections accidentally broken. As the contents are at practically all times under pressure, the liquid level in these cars is determined by a graduated tube passing into the tank through a stuffing box, equipped with a needle valve at the upper end, and provided with an accurate vertical adjusting device.

Liquefied petroleum gases are shipped in tank cars under the

TABLE 3 FREIGHT RATES ON REFINED OIL AND LIQUEFIED PETROLEUM GAS

City	Per cwt.	Per gallon
Boston	$0.80	$0.0528
Birmingham	0.555	0.03663
Chicago	0.360	0.02376
Cleveland	0.560	0.03696
Chattanooga	0.570	0.03762
Des Moines	0.330	0.02178
Detroit	0.560	0.03696
Indianapolis	0.435	0.02871
Kansas City	0.270	0.01782
Louisville, Ky	0.50	0.033
Minneapolis	0.42	0.02772
New Orleans	0.455	0.03003
New York	0.78	0.05148
Omaha	0.33	0.02178
Pittsburgh	0.605	0.03993
St. Louis	0.300	0.01980



Fig. 4 Butane Storage Installation

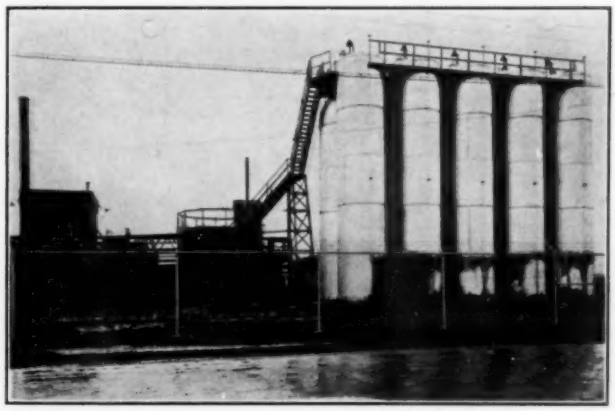


Fig. 5 Recent Installation for Butane Storage

refined-petroleum-products freight rates, which are based on the arbitrary weight of 6.6 lb. per gal. irrespective of the actual weight of the commodity. Table 3 gives the present rates from shipping points in Oklahoma, Group III, to various localities.

Butane is easily and economically unloaded from tank cars to storage tanks by the most convenient of the following methods: (*a*) Temperature differential, (*b*) pressuring with gas or air, and (*c*) pumping. Cars can be unloaded in slightly over one hour, and the cost is negligible, ordinarily being less than $0.0005 per gallon.

Storage. Storage tanks are usually of the horizontal cylindrical type, either riveted and hammer calked, riveted and electrically calked, or fusion welded. It is recommended that these tanks be designed in accordance with section VIII of the A.S.M.E. Boiler Code, and made for a working pressure equivalent to the vapor pressure of commercial butane at 100 deg. fahr. or about 65 lb. Tanks should preferably be installed above ground for convenience, but may be placed underground to satisfy special or local conditions. They are equipped with safety valves, drains, thermometers, gage glasses, or slip-tube gaging devices, and sometimes heating-tube bundles, in addition to the inlet

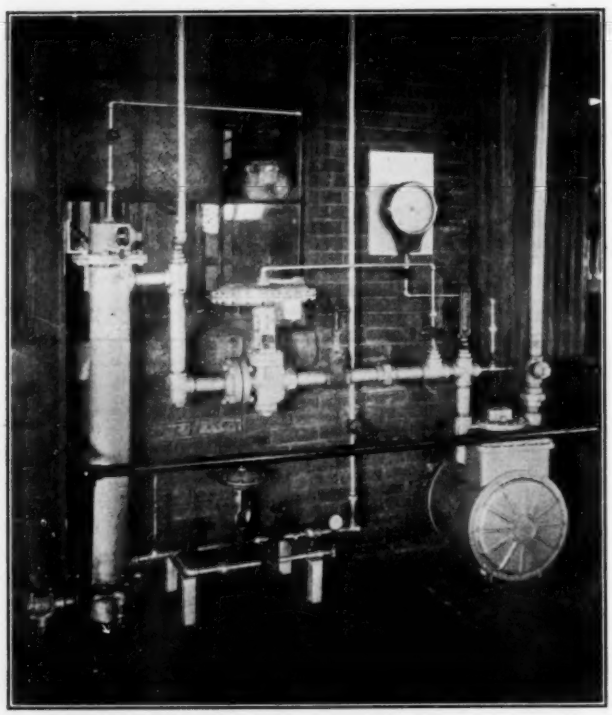


FIG. 6 BUTANE VAPORIZER AND REGULATORS

and outlet liquid and vapor lines, etc. Figs. 4 and 5 depict some recent butane storage installations.

Vaporization. Liquid butane must be provided with its latent heat of vaporization of 170 B.t.u. per lb. before it will evaporate. The level at which this heat is supplied can be as low as 32 deg. fahr. if vaporization takes place at atmospheric pressures. A multitubular vaporizer is ordinarily used, and the heat is supplied by hot or cold water or by exhaust steam. Fig. 6 illustrates a typical vaporizer and regulator installation. It will be noted that the pressure on the liquid from the storage is stepped down to a few pounds by means of a regulator ahead of the vaporizer, and that the uniform butane vapors resulting from the flash vaporization of the liquid are further reduced to gasoline or burner pressure by a final pressure governor. The vapor

TABLE 4 APPROXIMATE LIMITS OF INFLAMMABILITY OF GASES AND VAPORS IN AIR AT ORDINARY TEMPERATURES AND PRESSURES[1]

Gas	Lower limit by volume per cent	Higher limit by volume per cent
Butane	1.9	8.5
Propane	2.4	9.5
Ethane	3.2	12.5
Methane	5.3	14.0
Hydrogen	4.1	74.0
Carbon monoxide	12.5	74.0
Gasoline	1.4	6.0
Water gas	9.0	55.0
Natural gas	4.8	13.5
Illuminating gas	5.3	31.0

[1] U. S. Bureau of Mines Bulletin No. 279, by H. F. Coward and J. W. Jones, "Limits of Inflammability of Gas and Vapors," pp. 87–88.

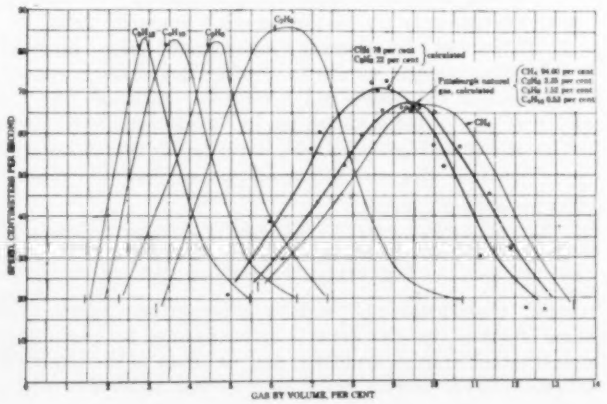


FIG. 7 SPEED OF UNIFORM MOVEMENT OF FLAMES IN A 2.5 CM. TUBE

(Taken from U. S. Bureau of Mines bulletin No. 294, "Carburetion of Combustible Gas With Butane and Propane-Butane Mixtures," by W. W. Odell, page 17.)

(3200 B.t.u. per cu. ft.) may be piped any distance at moderate pressure provided it is not cooled below its dew point (26 deg. fahr. at atmospheric pressure). The cost of vaporization is nominal and obviously depends on the heating medium used, but in any event is much less than the expense of heating and pumping fuel oil. With exhaust steam at $0.0008 per pound, the cost is about $0.0007 per gallon. No fuel-pumping charges are necessary as the vapor pressure of the liquid butane provides the motive power for distributing the liquid or the gasified fuel throughout the plant. The vapor may be readily and accurately metered to obtain fuel-consumption data for the entire plant or for tests on individual furnaces.

Burners. Owing to the high calorific value of butane vapors, and the consequent large amount of air required for combustion, the use of high-pressure burners where the flowing stream of gas entrains the combustion air is not practical. Low-pressure burners with the air supply drawing in the fuel gas have been very successful for a large variety of heating operations. Automatic premix types and hand-controlled premix-burning equipment have likewise shown their adaptability to handle this high calorific value fuel.

The combustion equation for butane is $C_4H_{10} + 6.5\ O_2 = 4\ CO_2 + 5\ H_2O$. It is apparent that 31.1 parts of air (O_2 = 20.9 per cent) are necessary to afford complete combustion of one volume of butane vapor, and that five volumes of water vapor are given off per unit volume of butane.

The inflammability limits of various gases mixed with air at atmospheric temperatures and pressures in large vessels are given in Table 4. The rates of flame propagation of several industrial gases mixed with air are shown in Fig. 7. It will be noted that the inflammability limits of butane in air are somewhat narrower than those of methane or natural gas, and very

much less than the limits on any of the manufactured gases. The maximum rate of flame propagation of butane in air (32 in. per sec.) is slightly greater than that of an average natural gas, but only one-half as much as the rate on typical carbureted water gas (64 in. per sec.). These factors make for safety in the handling and use of butane vapors, and permit combustion conditions and furnace design to be almost identical with natural-gas practice. On burners adapted for manufactured gases butane vapors

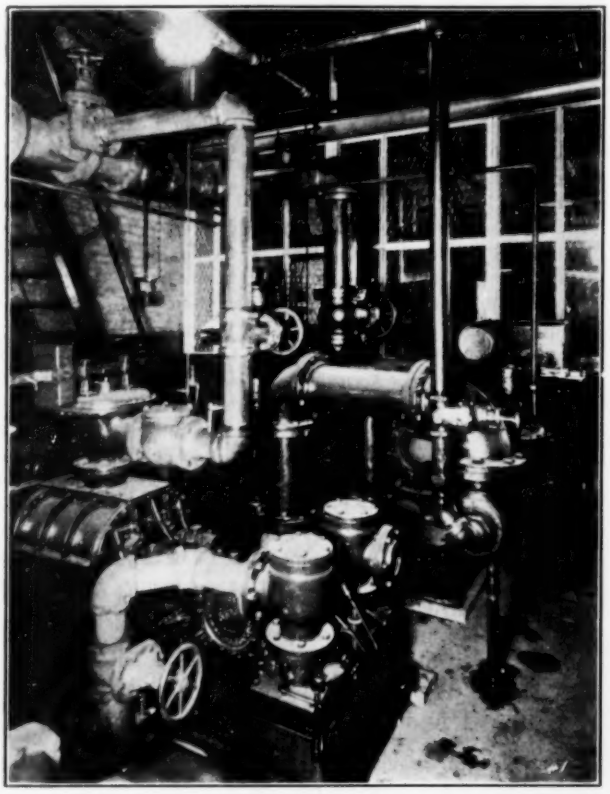


FIG. 8 AUTOMATIC BUTANE-AIR PREMIX MACHINES

will blow off the nozzles or tips owing to the lower rate of flame propagation. However, no difficulty is encountered in purchasing excellent butane-air mixing devices and burners from any of the prominent manufacturers, or in adapting practically any existing low-pressure or premix burners for proper combustion of butane.

Advantages. The unexcelled uniformity of composition of commercial butane liquid, and of the gas resulting from the flash vaporization thereof, coupled with the available gas-air proportioning devices and burners, results in an unparalleled ease of temperature regulation and furnace atmosphere control. These advantages plus the absence of sulphur compounds, dust, water, tar, gums, or other fuel contaminations insure constant furnace conditions, and make possible a hitherto commercially unattainable regularity of production with a minimum of interruptions and a negligible number of rejections or damaged stock.

The fact that each user of butane has in effect a private gas plant, insures against interruptions to the fuel supply and avoids periods of low pressure or insufficient quantity. Peak fuel loads can be carried without the penalty of a demand charge, and without pumping or boosting the gas.

Gas-piping sizes are much smaller when the high calorific value butane is used than is the case with manufactured or even natural gas. Fig. 8 illustrates this point. The 3-in. line carries

TABLE 5 PRICE COMPARISON BETWEEN BUTANE AND CITY GAS ON THERMAL EQUIVALENT BASIS

Selling price of "butanes" f.o.b. Oklahoma group III, cents per gallon	Cost of "butanes" at burner, cents per gallon	Equivalent cost of 530 B.t.u. city gas, cents per M cu. ft.
7.0	10.7	55.3
6.8	10.5	54.2
6.6	10.3	53.2
6.4	10.1	52.1
6.2	9.9	51.1
6.0	9.7	50.0
5.8	9.5	49.0
5.6	9.3	48.0
5.4	9.1	46.9
5.2	8.9	45.9
5.0	8.7	44.8
4.8	8.5	43.9
4.6	8.3	42.9
4.4	8.1	41.9
4.2	7.9	40.8
4.0	7.7	39.8
3.8	7.5	38.8
3.6	7.3	37.7
3.4	7.1	36.7
3.2	6.9	35.7
3.0	6.7	34.6

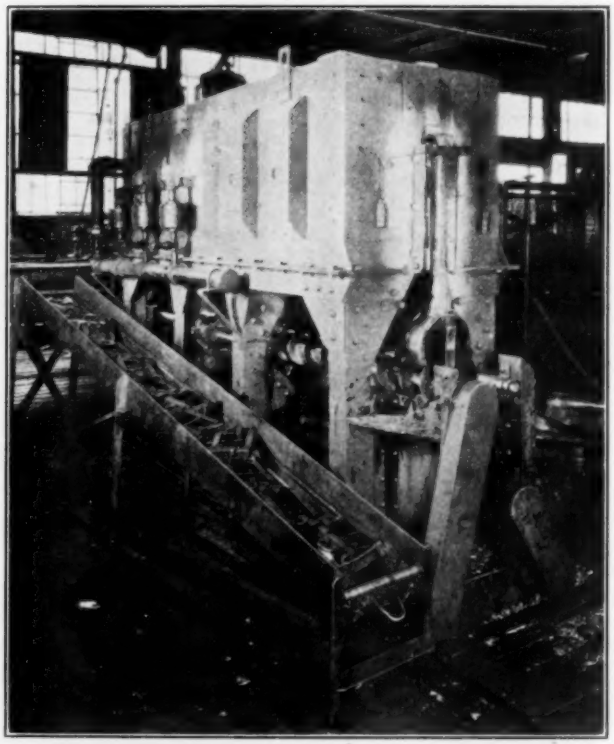


FIG. 9 CONTINUOUS BUTANE-FIRED FORGE FURNACE

butane vapors to a premixing machine formerly used on city gas. The old 8-in. manufactured gas main connected to the same machine can be seen and contrasted.

A tremendous gas reserve can be cheaply maintained in storage. One 15,000-gal. tank costing $2500 installed will ordinarily be filled to 14,500 gal. or 1,484,800,000 B.t.u. This is equivalent to 2,800,000 cu. ft. of 530 B.t.u. gas. The cost of a conventional low-pressure holder for this amount of gas would run in the neighborhood of $200,000. Any gas-fuel user would consider himself fortunate were he formerly able to afford such a fuel reserve.

Industries where refined heating operations are conducted, and where gas fuel is practically a necessity, can now locate their plants to take advantage of low labor costs, better employee living conditions, low real estate values, and other allied advantages, when they were formerly forced to operate in the congested territories already served with gas.

Butane manufacturers are offering long-term flat-price contracts with commitments up to five years. Other agreements are entered into at a fixed differential above gas or fuel-oil markets with minimum price stipulations. Continued execution of such contracts indicates the stability of the liquefied petroleum-gas industry and implies its vigorous growth and long life.

The application of butane to industrial uses sometimes builds up a competitive fuel-buying situation that is often used to advantage by the gas, butane, or oil purchaser, and tends to force each fuel into its natural field of activity where its value in terms

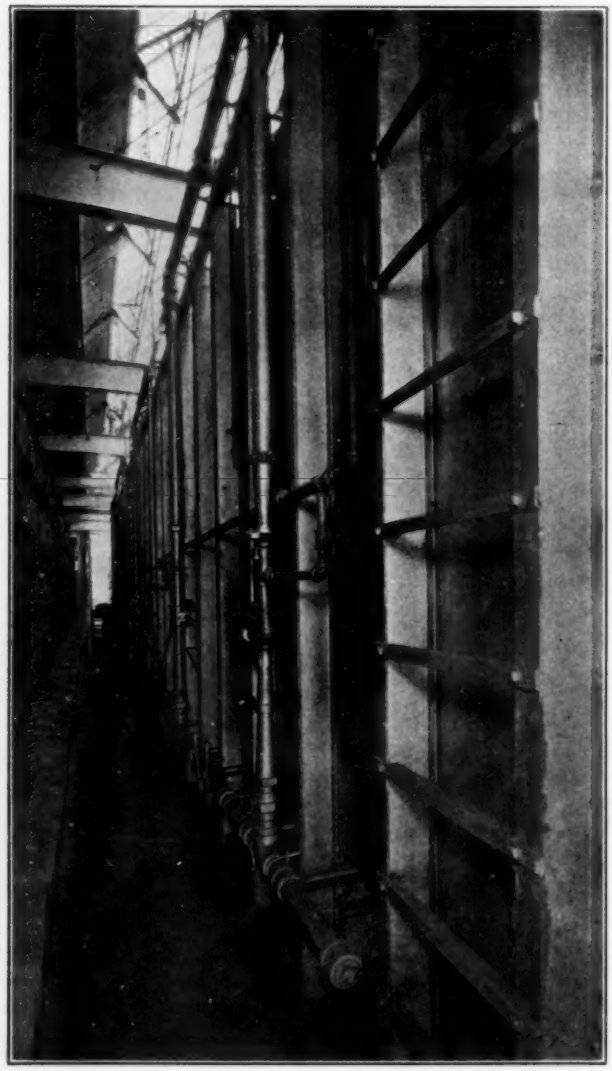


FIG. 10 BUTANE-FIRED TUNNEL KILN MALLEABILIZING CASTINGS

of finished production is greatest, and its economies most pronounced.

Fuel Cost. Butane is ordinarily sold in tank-car lots in the manufacturer's cars f.o.b. the shipping points, which are practically all at present located in Oklahoma and Texas. The price ranges from $0.04 to $0.07 per gallon in cars, or from $0.39 to $0.69 per therm, depending on the quantity contracted for. It is obvious that the freight rates to destination are a controlling factor in the economies of butane use. The freight rates (see Table 3) vary between $0.02 and $0.05 per gallon, with an average of perhaps $0.035 per gallon to points of use in the Mississippi valley and Great Lakes industrial districts. Adding a storage and handling charge of $0.002 per gallon, which is ordinarily found to be ample, a total average addition of $0.037 per gallon may be used. A strictly thermal comparison between butane and 530 B.t.u. manufactured gas results in the cost relations shown in Table 5. Needless to say, the purchase of manufactured gas for industrial use at 30 to 60 cents per thousand is ordinarily considered attractive. The use of butane, with its several advantages over other gaseous fuels and at the indicated delivered costs, presents an even more attractive picture, and will command increasingly greater attention from large-scale users of clean, high-grade, economical, and easily controlled fuels.

Disadvantages. The primary disadvantage encountered in the use of butane lies in the necessity for making a capital expenditure for liquid storage. A minimum 15-day fuel supply is ordinarily recommended. The cost of completed storage will average $0.18 per gal. exclusive of real estate, but including the vaporizer and regulator equipment. If this storage is filled and emptied 18 times per year (each 20 days) the storage cost is $0.0015 per gal. based on a 15 per cent annual capital charge. The lack of available space for storage tanks in a congested manufacturing district is sometimes a factor, but is usually overcome by burying the tanks below the surface. If the tanks are installed properly according to local fire ordinances, state regulations, and insurance company rules, the vaporizers located outside plant building, and the gas piping run as usual, no objections are encountered from the regulatory bodies, and no increase in insurance premiums results. Liquid butane is considered to carry the same fire or explosion hazard as gasoline or motor fuels. The butane vapor in low-pressure piping has the same relative classification as piped natural or manufactured gas.

Butane manufacturers have a large and adequate fleet of tank cars. Fuel deliveries are therefore prompt, and present no problems different from the rail delivery of fuel oils.

Uses. Present successful industrial uses of butane include: Heat treating and hardening, annealing and normalizing, carburizing, forging, malleabilizing, enameling, japanning, core baking and mold drying, metal cutting, galvanizing, glass-machine forehearths, glass annealing, ceramic ware, textile singeing, and internal-combustion engines.

CONCLUSIONS

1 Butane is transported economically and handled as easily as liquid, and also has all the advantages of other gaseous fuels.

2 Butane is uniform in composition and is uncontaminated with deleterious substances. Precise temperature and atmosphere control is easily obtained.

3 Butane can be burned economically with standard combustion appliances and in standard furnaces.

4 Butane can only be used industrially where freight rates permit economical delivery. For this reason it will be utilized primarily in the Middle West, the Mississippi Valley, and Great Lakes industrial regions. It will not invade New England or the far South.

5 Butane is competitive with manufactured city gas for practically all large-scale industrial uses in the territories to which it can be economically transported. It is not competitive in any event with natural gas.

6 Butane is not competitive with fuel oil on a thermal basis, but will win many an industrial load from oil because of its cleanliness, ease of control, and lower cost per unit of finished product.

7 The supply of butane is adequate to justify almost unlimited industrial use without fear of depletion of raw material supply.

8 Butane is swiftly becoming a factor in industrial fuel markets, and its use and importance are destined to increase.

FSP-52-22a

Carbonization-Plant "Lurgi" Process of Lehigh (N. D.) Briquetting Company

By MAX TOLTZ,[1] ST. PAUL, MINN.

The first operating installation of the German "Lurgi" process for treating low-grade fuel is described. The process is particularly adapted to North Dakota lignite; and the carbonized-coal residue, with pitch by-product as a binder, is pressed into merchantable briquets, which burn as a smokeless fuel with high recoverable heat value.

THE carbonization-plant "Lurgi" process of the Lehigh Briquetting Company is located about one mile south of the Northern Pacific Railway's main track at Lehigh, N. D., and about 3½ miles east of Dickinson, N. D. It is the first completed American installation of the German process for treating low-grade fuel now in operation.

The process is adapted to North Dakota lignite only and produces a carbonized-coal residue which, when bound together with one of the by-products, pitch, and pressed into briquets delivers a smokeless, sootless, clinkerless fuel of high recoverable heat value and which can compete successfully in the market with anthracite and all intermediate grades of natural and briquetted raw coal.

This process was invented by Dr. F. A. Oetken and Dr. O. Hubmann, of Germany, and was developed under their patents by the Lurgi Corporation of Frankfort-on-the-Main, Germany.

The lignite tar recovered by this process is of a high grade, from which many by-products of value such as creosote and paraffin are recovered.

The result of the carbonization of the lignite is a residue called carbon char, which when not briquetted can be used with high efficiency, especially in pulverized form for firing boilers and also for smelting purposes. This latter will probably influence the production of iron and steel from the great mines of northern Minnesota, due to the fact that the char not alone can stand the pressure in the cupola, but having no impurities such as sulphur, will produce a better and cleaner pig iron than that made with common coke.

There are in the region of these lignite deposits some of the finest clays of the United States for the manufacture of face-brick, paving brick, firebrick (good for 4200 deg. fahr.), hollow tile, terra cotta, etc., which excel in quality. Some of the clays can be used in the manufacture of pottery and white porcelain, also of white cement. There are also immense deposits of alumina and other minerals, all of which await suitable cheap fuel to become commercially profitable.

The plant is built on top of the coal mine, the lignite of which lies about 70 ft. below the surface and has a thickness of from 12 to 18 ft. The coal is mined according to the room-and-pillar system, the rooms being 20 ft. wide and about 100 ft. long, and the pillars or walls between the rooms are 8 ft. thick. The supporting roof is 4 ft. thick coal.

The buildings are of substantial fireproof construction, consisting of a steel structure (see Fig. 1), with brick curtain walls and steel windows. The floors are of reinforced concrete. The layout of the plant is shown in the diagram, Fig. 2.

The accomplishment at Lehigh has shown that through this new process of chemical change and from the vast lignite resources of North and South Dakota and eastern Montana can now be made available for commercial and domestic use.

The process of carbonization of this lignite is as follows:

The lignite comes from the mine in chunks of from 12 to 20 in., and is deposited in a hopper, from where it is belted to the magnetic separator and then into the crusher. The flow sheet is shown in elevation and in plan in Fig. 3. On the belt, coal which contains clay matter is thrown out. The crusher reduces the size to 4 in. or less. The coal is then raised by a bucket

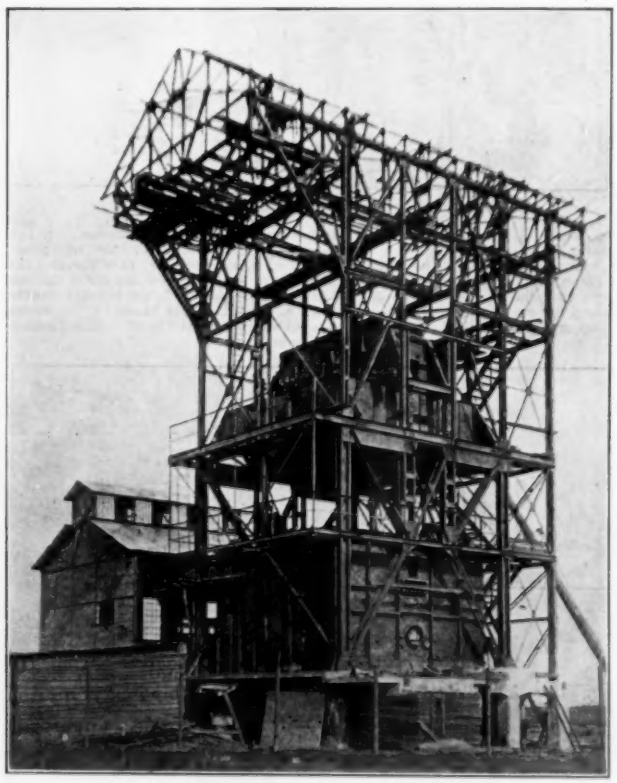


FIG. 1 CARBONIZER BUILDING UNDER CONSTRUCTION

elevator and goes over a vibration screen, which takes out the sizes of ¾ in. and under. This fine coal is conveyed to the power house and is used on the chain grates under the boilers. The coal ¾ in. and over falls into hoppers, from which it is conveyed by a belt to another bucket elevator to the top of the carbonizer building, where an automatic loader delivers it to the two raw-lignite bunkers of the carbonizer. (See Fig. 4.)

From here the coal falls into the dryer, which is heated by gas generated in the drying chamber, and which is circulated by a fan after it is heated by the combustion of gas from the scrubber in a specially built combustion chamber. The temperature

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

in the dryer is kept at about 230 deg. cent. (446 deg. fahr.). The dried coal falls then by gravity into the carbonization chamber, to which heat from the cooling zone as well as from the combustion of the generated gas, similar to that in the dryer, is supplied. (See Fig. 5.)

The temperature of carbonization for this lignite was established after experiments and must not be lower than 600 deg. cent. (1112 deg. fahr.) and not higher than 620 deg. cent. (1148 deg. fahr.). This temperature is checked by a set of pyrometers and is regulated from a board located in the gas-cleaning room. If

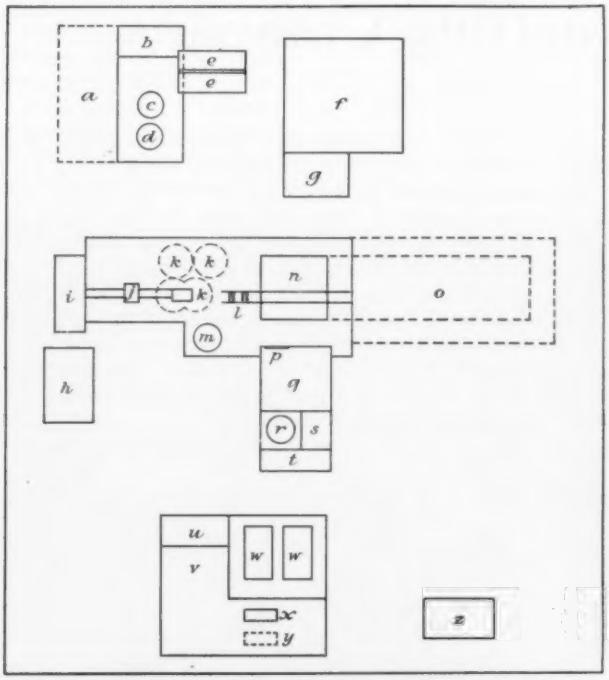


FIG. 2 PLAN OF BRIQUETTING PLANT

(*a*, extension for output of 400 tons of briquets; *b*, compressor; *c*, preheater in distillation plant; *d*, still; *e*, pitch tanks; *f*, briquetting plant; *g*, char hopper; *h*, mine hoist; *i*, tipple; *j*, crusher; *k*, raw-lignite bins; *l*, elevator; *m*, gas producer; *n*, carbonizer; *o*, extension for three carbonizers to balance for output of 400 tons of briquets; *p*, pyrometer control board for carbonizer; *q*, gas cleaning; *r*, cooler; *s*, tar tanks; *t*, tar water; *u*, laboratory; *v*, shop; *w*, boilers; *x*, turbine; *y*, position of future turbine; *z*, spray pond.)

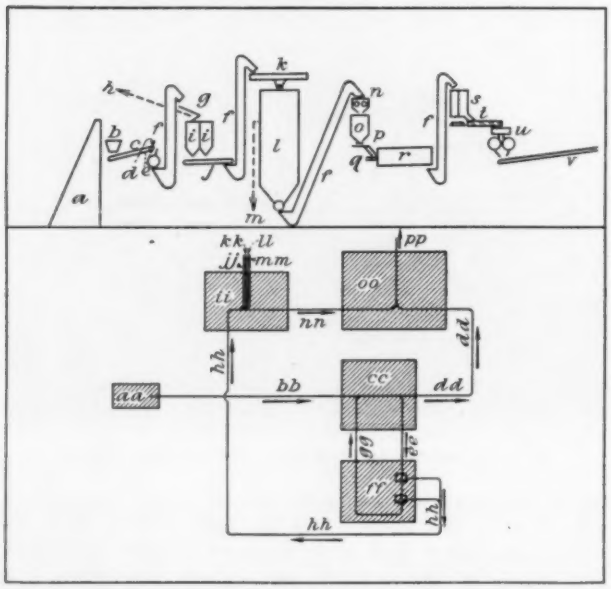


FIG. 3 FLOW SHEET—ELEVATION ABOVE AND PLAN BELOW

(*a*, mine tipple and hoist; *b*, hopper; *c*, belt; *d*, magnetic separator; *e*, crusher; *f*, elevators; *g*, vibration screen; *h*, screenings, to power plant; *i*, hopper for raw lignite; *j*, belt conveyor; *k*, movable loader; *l*, carbonizer; *m*, gas, to tar distillation plant; *n*, char crusher; *o*, hopper; *p*, feed table; *q*, pitch atomizer; *r*, special mixing drum; *s*, fluxer; *t*, screw conveyor; *u*, briquetting press; *v*, cooling belt, to storage; *aa*, mine hoist; *bb*, direction of raw coal; *cc*, carbonizer; *dd*, direction of char; *ee*, gas; *ff*, gas-cleaning plant; *gg*, clean gas; *hh*, tar; *ii*, tar distillation; *jj*, light oil; *kk*, heavy oil; *ll*, paraffin; *mm*, creosote tar; *nn*, pitch; *oo*, briquetting plant; *pp*, briquets.)

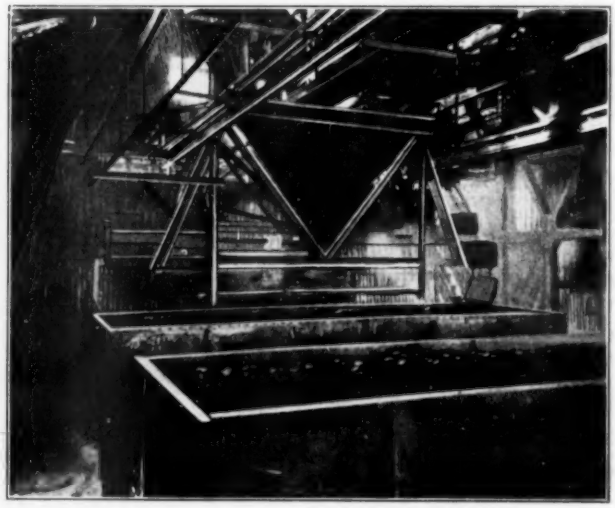


FIG. 4 AUTOMATIC LOADER FOR CARBONIZER

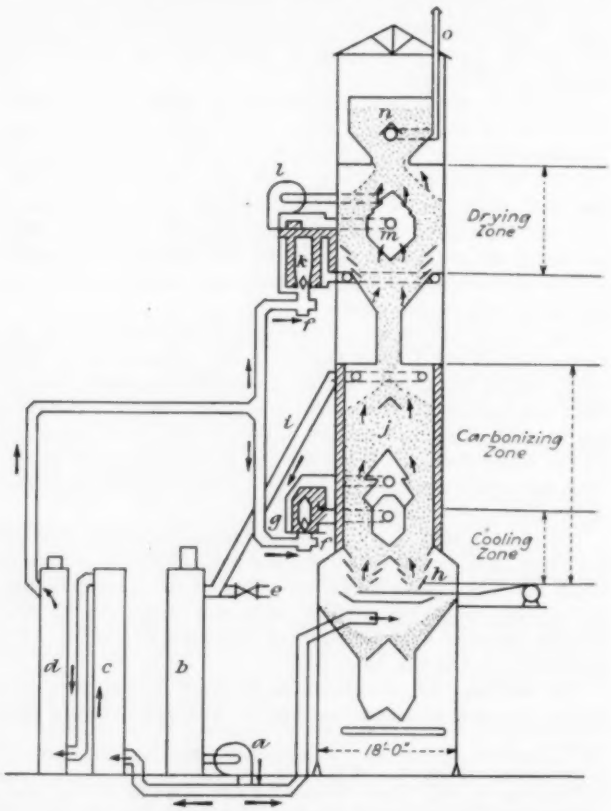


FIG. 5 SECTION OF CARBONIZER AND GAS DISTRIBUTION

(*a*, blower; *b*, Theissen tar disintegrator; *c*, cooler; *d*, Raschig scrubber; *e*, producer gas; *f*, air; *g*, heater for carbonization chamber; *h*, cooling hopper for char; *i*, air; *j*, carbonization chamber; *k*, heater for dryer; *l*, blower; *m*, dryer; *n*, raw-lignite bunker; *o*, ventilator.)

the temperature is too low, the heat created in the heater is increased, but when the temperature is too high, cooling is effected by supplying cooled air and gases from the cooling hopper.

The reason why the "Lurgi" carbonizer was selected was that it had no moving parts except the two gates in bottom of the char hopper.

It has been found by daily operation that if the lignite contains less than 38 per cent of moisture the gas generated in the carbonization zone is sufficient for heating the lignite particles to change them into carbon char. If the moisture exceeds 38 per cent, then producer gas has to be added.

The gas generated in the carbonization zone is drawn off at the top of that zone into a Theissen disintegrator in which the tar is extracted. The gases from here are used partly for cooling the carbon char in the hopper below the carbonization zone, and partly are drawn through a cooler and washer or scrubber and are then conducted to the heaters for the dryer and for the carbonization chambers. This process converts the lignite into carbon char of the size of pea coal. (See Fig. 6.)

After this char has gone through the cooling zone, still having a temperature of 120 deg. cent. (248 deg. fahr.), it falls into the cooling hopper, from where it is elevated by another bucket elevator to the char crusher. This is a special machine imported from Germany. The rolls of high-tempered hard-steel alloy have recently been replaced by carborundum rolls, also made in Germany, and the result of crushing the char to $^1/_8$ in. size has been considerably improved, so much so that the finished briquets have the appearance of anthracite coal. The crushed char is hoppered and goes to a table, which feeds the right amount into the mixing drum, into which at the same time pitch in atomized form is introduced. The mass from the mixer is elevated to a fluxer, where if necessary superheated steam is added to make the mixture pliable, and from thence is taken by a screw conveyor to the briquetting press. (See Fig. 7.)

The finished hot briquets, in the shape and size of an egg (see Fig. 6), are then sent over a cooling belt of wire mesh to the storage bin.

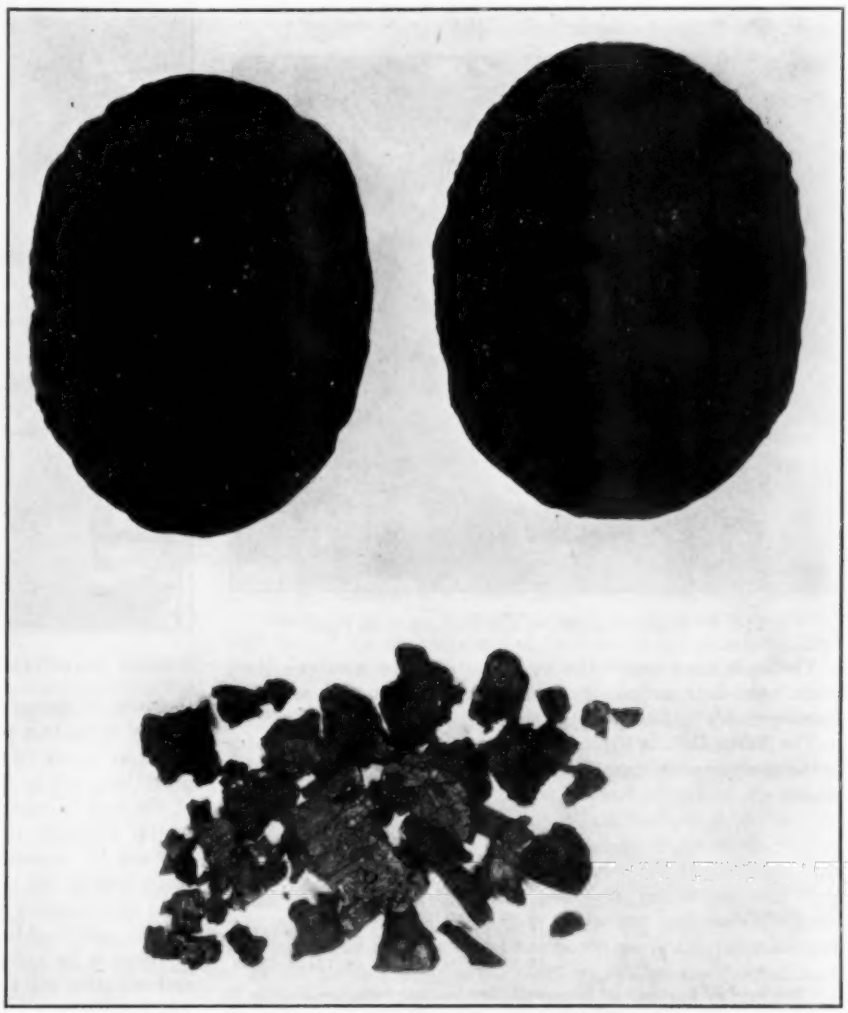


FIG. 6 BRIQUETS, NATURAL SIZE, SHOWN ABOVE; LIGNITE CARBON CHAR, NATURAL SIZE, SHOWN BELOW

From the tar the tar water or ammonia liquor is separated, which latter is wasted at present, although by-products of ammonia sulphates and acetate of lime can be procured from it. (See Fig. 8.)

The tar is then pumped by compressed air in superheated steam-jacketed pipes, a novel installation, to the distillation plant, in which from a still heated to a temperature of about 320 deg. fahr., the pitch is drawn off. The remainder of the tar, on account of being dust free, has therefore a good market value or can be converted into light and heavy oils, creosote, and paraffin.

The apparatus used in this process is of standard type as to rubber belt conveyors, bucket elevators, magnetic separator, coal crusher, char crusher, mixer, fluxer, screw conveyor, and briquetting press. The belt which conveys the hot briquets is of wire mesh. The new features are the carbonizer, the Theissen disintegrator, and the Raschig scrubber, all of which are patented.

The carbonizer is open on top and closed by two gates at the bottom of the char hopper. It has no moving parts and consists of the drying zone, carbonizing zone, and cooling zone. It is built of steel throughout, 18 ft. square at the bottom and about 80 ft. high. The sides of the carbonization chamber are lined with firebrick. (See Fig. 5.) The automatic loader in connection with the carbonizer is almost human, and also can be handled by remote control.

The Theissen disintegrator is really a separator of the centrifugal type to separate the tar and tar water. It is a vertical machine. Of the Raschig scrubber no details could be obtained.

The capacity of the plant at present is not balanced; that is, the carbonizer can produce only 100 tons of char per 24 hours while the briquetting plant is able to make 400 tons in 24 hours, or in other words, three more carbonizers will have to be installed to balance the output of the briquetting plant, and at the same time the distillation plant will have to be doubled to balance with this future output.

The power house contains two 300-hp. longitudinal water boilers and one 375-kva. turbine at 80 per cent power factor, 175-lb. initial pressure, running condensing. Water for condensation and boilers is furnished by a spray pond. The boilers are equipped with chain-grate stokers

The power used in the different parts of the plant is as follows:

Coal crushing	30 hp for 4 hours per day
Carbonization	120 hp. for 24 hours per day
Briquetting	120 hp. for 6 hours per day
Mine operation	100 hp. for 24 hours per day
Pumping tar and pitch with compressed air	20 hp. for 8 hours per day
Lighting, etc	10 hp. for 24 hours per day

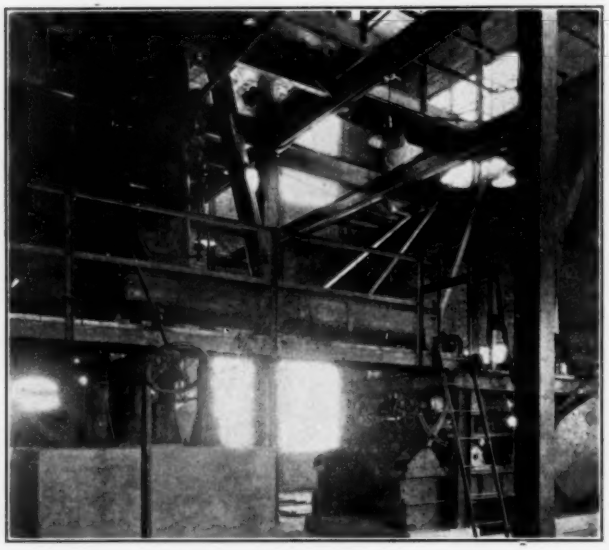


FIG. 7 SCREW CONVEYOR AND BRIQUETTING PRESS

The carbonizer needs the attention of three men per shift, while when four carbonizers are installed, the attendance will be increased only to four men per shift.

The North Dakota lignite is a non-coking coal. The following is the analysis of the raw lignite, char, and briquets of the Lehigh plant:

		Moisture	Volatiles	Fixed carbon	Ash	Sulphur	B.t.u.	B.t.u., combustible[1]
Lignite	wet	40.40	26.35	26.45	6.80	0.86	6380	
	dry		44.35	44.30	11.45	1.44	10704	12088
Char (600° C.)	wet	1.15	10.90	73.80	14.15	2.24	12759	
	dry		11.00	74.50	14.30	2.26	12778	15177
Briquets	wet	1.30	15.95	70.05	12.70	1.98	12942	
	dry		16.65	71.00	12.85	2.04	13112	15048

[1] The sulphur in char and briquets is combustible to approximately 25 per cent; the rest is ash-sulphur and non-combustible.

It takes 2 1/3 tons of lignite of the foregoing quality to make one ton of char. The amount of tar extracted is 2.8 to 3.2 per cent in weight, of which 42 per cent is pitch, and which, as stated, is used for binder in the briquets. The balance of 58 per cent of tar consists of 8 per cent water, 5 per cent gas, 19 per cent creosote, 26 per cent paraffin fractions (8 per cent paraffin).

To briquet the char 6 to 6 1/2 per cent pitch in weight is necessary for a binder. From this it can be seen that the lignite does not produce enough pitch for all the manufactured char to be briquetted. It is therefore necessary to procure pitch from other sources for over 50 per cent of the char, unless this part of the char is marketed, which can be done readily.

The author is not able to give exact expenditures and cost of operation, but the following will give a fair picture of the financial side of this plant:

The total cost of the plant as it stands today, not including

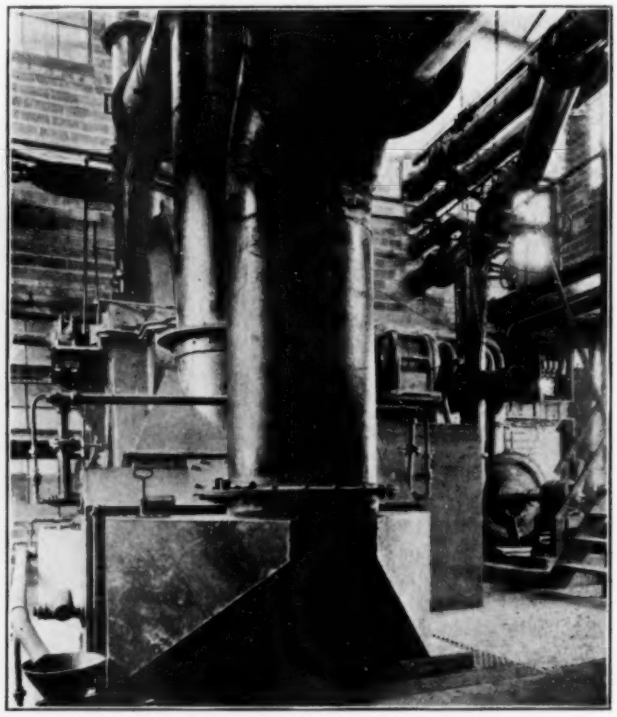


FIG. 8 GAS CLEANING AND TAR AND TAR WATER SEPARATION

the mine, is approximately $550,000. Having a capacity of 100 tons of briquets a day, the cost per ton output would be $5500, which is excessively high. If the plant is extended so that everything will be balanced, which would give a daily capacity of 400 tons of briquets, the total expenditure would be approximately $700,000, or $1750 per ton output. This is also high, but can be accounted for by the reason that this is the first plant built in this country. The author's estimate of cost of an eight-unit plant, or 800 tons daily capacity, is $900,000, or $1125 per ton output of briquets.

There is no question that the processes of low-temperature carbonization will be improved and simplified in the near future, which will reduce initial investment and also manufacturing costs.

So far as could be ascertained, the cost of manufacturing one ton of briquets at the Lehigh plant is a little more than $6, but the author has estimated, and quite conservatively, that with an eight-unit plant and a strip mine in which coal can be produced for less than one-half of the present cost of operation, one ton of briquets can be manufactured for $3.30 or less.

The plant has been in operation since December, 1928, and has given the results promised by the Lurgi Corporation.

In conclusion the author acknowledges with thanks the cooperation of Dr. Hubmann, Mr. K. Loven, manager, and Mr. R. E. Brewer, superintendent, of the Lehigh Briquetting Company.

[Discussion of this paper follows the paper by J. D. McQuade.]

FSP-52-22b

The Hayes Process Low-Temperature Carbonization Plant at Moundsville, W. Va.

By J. D. McQUADE,[1] PITTSBURGH, PA.

Low-temperature carbonization was adopted at Moundsville so that the large percentage of slack and screenings could be converted into a merchantable fuel and sold at a profit, clearing the way for the prepared coal at a price that would enable it to enter the open market and justify the continued operation of the mine. The process adopted had to meet the requirements of being operated over long periods and be free from carbon formation when working high-volatile bituminous coal; the retorts must be of a design to allow of easy repair with a minimum of time and labor; the process must be automatic; it must yield tar and gas as by-products; the investment must be so low as to yield an operating profit. Operating costs and returns have been such that the unit will be expanded for a much larger output.

THE primary reason for undertaking the development of a low-temperature carbonization process at Moundsville was to provide a means whereby the slack coal or screenings, which constitute from 30 to 40 per cent of the output of the mine, could be changed into a fuel that could be sold at a profit.

It is unnecessary to go into detail regarding the troubles which for the last five years have beset the coal industry, as they are generally known. However, the company became convinced that if it had an outlet for the screenings, through which they could be sold at a price above the cost of production, the prepared sizes of coal could be sold on the open market at a price that would justify the continued operation of the mine.

With this object in view, a thorough investigation was begun of the different processes having for their object the utilization of low form value screenings. Many schemes were investigated, which ranged from various methods of cleaning and preparing the coal to the various methods of carbonizing the coal at low temperatures. Some of these processes were installed and operated on a small scale, only to be discarded because of high installation and operating costs or because the finished product was not in a suitable form for domestic use.

As a result of these investigations, the company was thoroughly convinced that low-temperature carbonization offered the most satisfactory and most profitable means of changing the form value of bituminous-coal screenings, provided a process could be found that met with the following requirements:

1 It must be a machine capable of continuous operation over reasonably long periods and without difficulties due to carbon formation and sticking when using high-volatile bituminous coals.

2 The retorts must be of such design that permit necessary replacement and repairs to be effected in the shortest possible time and with a minimum of labor.

3 The process must be automatic or nearly so in its operation so as to reduce the amount of labor and supervision to the lowest possible figure.

4 The process must be of such a type that the entire solid product could be recovered in a form suitable for use as a high-grade domestic fuel and in such form that it would stand the abuse of handling and shipping with the least amount of degradation.

5 It must yield a large amount of tar oils of such characteristics that have the highest market value and at the same time yield a gas of high heating value.

6 The initial investment cost per ton day input of coal must be sufficiently low to allow the fuel product to be sold in competition with existing fuels and at the same time realize a substantial return on the investment.

With these objectives in mind, and after many tests on small scale retorts, the patent rights for the Hayes process were purchased in 1926. In 1927 the first full-scale unit was built, having a capacity of 50 tons of coal input per 24 hours.

With the operating experience and data obtained from this unit as a basis to work on, a second unit of 50 tons capacity was constructed in which many improvements were incorporated. This unit has been in successful commercial operation up to date.

The Hayes process retort consists of an alloy steel tube 17 in. inside diameter by 20 ft. long. This tube is placed in a furnace setting and supported by rollers on each end. Stationary feed and discharge castings are located at either end of the retort and are connected to it by special packing rings. The tube is rotated at a speed of $1^1/_2$ revolutions per minute.

Within the retort is a specially constructed screw conveyor. This screw is driven by a train of gears which gives it a progressive oscillating motion. The motion is such that the screw moves the coal toward the discharge end of the retort a given distance, and on the reverse oscillation it is returned a distance slightly less than the forward motion. Due to this forward and backward motion, the coal has a theoretical travel of 240 ft. in passing through the length of the retort.

The problem of coal carbonization resolves itself into a question of heat transfer. In low-temperature carbonization, due to the relatively low heat head available, it is difficult to raise the temperature of a layer of coal to the point where complete distillation is effected in a reasonable length of time. In this process, due to the motion of the retort screw, every particle of coal is brought into contact with the hot walls of the retort, and the carbonization is completely effected in the short time of 25 minutes.

In the present unit, seven of these retorts are located in a battery type of furnace. Each retort is heated externally by means of gas burners located on the under side of the furnace, and each retort is entirely independent in the operation of the other six. Removable one-piece covers form the top of the furnace, and any retort can be removed for inspection or repairs without interfering with the operation of the other six.

Coal is fed to each of the retorts by a screw feeder located on the feed end casting. The rate of feed is 700 lb. per hour for each retort.

The char is discharged from each retort into a main collecting conveyor, which runs the length of the retort setting, thence being discharged into a quenching conveyor, where it is cooled by water sprays. From this conveyor the char discharges through a rotary seal valve and is conveyed to a crusher, where it is ground to the proper size for briquetting.

The coal gas and tar vapors are removed from the retorts through offtake connections located at either the feed or discharge

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[1] Coal Carbonization Company.

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end of the retorts and are conducted through standard condensing and cleaning apparatus, where the tar is removed and the gas is sent to the holder.

In this installation the gas produced is returned to the furnaces and is used for heating the retorts, and the excess gas is used for steam raising.

The coal processed is the $^3/_4$-in. screenings produced by the Ben Franklin Coal Company and has the following average analysis: Moisture, 2.25 per cent; volatile, 39.40 per cent; fixed carbon, 51.04 per cent; ash, 7.31 per cent. B.t.u. per pound, 13,500.

This coal is rescreened over a $^1/_2$-in. bar screen, and the coarser sizes are crushed to pass a screen of the same size.

The average yield of products derived from one ton of this coal is as follows: Char (12 per cent volatile), 1326 lb.; tar, 30 gal.; gas (800 B.t.u. per cu. ft.), 5000 cu. ft.

The char after being ground is passed through a standard briquetting machine of the Belgian roll type where about 10 per cent of coal tar pitch is used as a binder. The briquets are pillow-shaped and weigh about $2^1/_2$ ounces.

This type of fuel is ideal for domestic use. It is readily controlled in a heating furnace or stove, will hold the fire for long periods, and retains its shape in the fire until entirely consumed. It will withstand the roughest abuse in handling and shipping, is not affected by weather conditions, and is remarkably free from broken pieces or dust when delivered in the consumer's bin.

As is the case with nearly every machine, the success of its operation is mainly dependent on the method of control and its ability to maintain uniform conditions throughout the entire system.

The heating of the retorts is controlled in accordance with temperature readings taken in the furnace chamber at the feed and discharge end of the retort. All the firing is done at the feed end, where a temperature of 1400 to 1600 deg. fahr. is maintained dependent on the nature of the products desired and the nature of the coal being processed. This produces a furnace temperature of 1100 to 1300 deg. fahr. at the discharge end of the retort. This method of firing has the effect of removing the moisture from the coal and raising it to the distillation temperature before it passes the middle of the retort. All that is then required is to maintain this temperature until complete distillation is effected. The heat required for carbonization is approximately 1200 B.t.u. per pound of coal, or about 60 per cent of the total gas made.

Automatic governors regulate the vapor pressure on the retorts, and uniform temperature conditions are maintained by automatic valves at various points through the gas-handling equipment.

This unit has now been operated over a sufficiently long period of time to demonstrate its ability to produce the required type of products and to prove the reliability of its operation.

The products have been marketed in different localities, and their reception has been such as to leave no doubt as to the prospects for selling large quantities of both the tar products and fuel briquets.

There has been much speculation as to the value of low-temperature tars, and the general impression has been that their only immediate use would be as a fuel and that their value could only be rated as such. Recently, however, there has been shown a remarkable interest in the distillates from low-temperature tars by the manufacturers of synthetic-resin compounds. Due to their high tar-acid content and other peculiar characteristics, these distillates are beginning to find favor in this industry, and their value is indicated by the willingness of several manufacturers to contract for large amounts at a very good price.

As is the case with every new industrial undertaking, the question arises if it is commercially profitable. The answer to this question is that plans are now under way for the erection of a plant having a coal input of 400 tons per day. The carbonizing units of this plant will be merely a multiplication of the unit now operating, and there will be no untried schemes incorporated in its design. Provision is being made to distil all the tar produced in order to recover a suitable pitch to be used as a binder for the briquets, and only the distillate oils will be marketed.

The estimated operating cost and returns on this plant are as follows:

Labor

	Cost per day	Cost per ton input
General: 1 superintendent at $12 per day, 1 chemist at $6, 6 general laborers at $4	$42.00	$0.105
Carbonizing plant: 3 foremen at $6 per day, 12 men at $4	66.00	0.165
Steam plant: 3 firemen at $4 per day	12.00	0.030
Briquetting plant: 1 foreman at $6 per day, 2 men at $4	14.00	0.035
Tar distillation: 9 men at $4 per day	36.00	0.090
Total	$170.00	$0.425

Operating Cost per Day

	Daily cost	Cost per ton input
400 tons slack coal at $1 per ton	$400.00	$1.000
Labor	170.00	0.425
Interest on $600,000 at 7% per annum	116.66	0.290
Depreciation at 10% per annum	166.66	0.411
Royalty at $0.30 per ton	120.00	0.300
Repairs and supplies	75.00	0.187
Power and water	50.00	0.125
Office and laboratory	25.00	0.062
Total cost	$1123.32	$2.800

Income per Day

	Daily income	Income per ton input
300 tons briquets at $5	$1500.00	$3.75
4000 gal. distillate oils at $0.13	520.00	1.30
Total income	$2020.00	$5.05
Operating costs	1123.32	2.80
Profit	$ 896.68	$2.25

From the foregoing, it would seem reasonable to conclude that within a comparatively short time the larger producers of bituminous coal will, by the low-temperature carbonization of the coal they produce, place their business, which is now unprofitable, on a profitable basis.

It would also seem probable that the time cannot be far distant when bituminous coal will not be in general use in its raw state as a fuel, but its place will be taken by a smokeless fuel, the product or result of low-temperature carbonization of bituminous coal.

From the present trend in the value of low-temperature tar products, it would appear that by the use of a low-temperature distillation process the coal producer will be able to sell the smokeless fuel resulting from such a process at a price to bring it within the reach of the average domestic consumer.

In addition, when such smokeless fuel can be thus had by domestic consumers, it is reasonable to suppose that the municipal authorities in the cities and larger towns, in order to eliminate smoke, will prohibit by law the use of raw bituminous coal as a fuel.

From all these considerations, it may be possible that low-temperature distillation of bituminous coal will prove to be the remedy for the "smoke nuisance" about which so much is heard in the cities today.

Discussion

HORACE C. PORTER.[2] Low-temperature carbonization of coal has, in most cases where it has been tried commercially, proved uneconomical due to high costs of plant and operation. The author presents figures of estimated costs and returns on a plant that has been tried out on a smaller scale. These figures, if they work out in practice as indicated, show a substantial profit.

[2] Consulting Chemical Engineer, Philadelphia, Pa.

The coal is charged at $1 per ton, which, with a profit of $2.25 per ton from the carbonizing process, totals $3.25 net return to the mine owners on their slack coal.

There are two sets of conditions under which low-temperature carbonization is likely to find its best prospect of success. One is as outlined in this paper—the conversion of low-value screenings at the mine into an improved fuel product commanding an enhanced value. The other is pretreatment of sticky and smoky coals as an auxiliary in steam-power-plant practice.

In both of these cases the profit will depend on the market for the oils and gas produced. It is a question whether the demand for low-temperature tar oils at the figure of 13 cents per gallon, as given in the paper, can be maintained indefinitely as production is increased.

It is a waste of a valuable by-product to burn rich gas as is done at this plant for heating the retorts. This gas (800 B.t.u. per cu. ft.) would have a very much higher value if applied to enrichment of blue water-gas for distribution as a public supply. When large installations of this type of plant are built, the possibilities should be well considered of placing them where they can contribute to a public gas supply, perhaps through long-distance transmission, and thereby add to the profits obtained through by-products.

The success of low-temperature carbonization will depend to a large degree on the relative future market returns from tar oils and gas. High-temperature carbonization gives high gas yields; low-temperature carbonization gives high tar yields. Future economic conditions affecting the demand for these products will have a large influence on the progress of low-temperature carbonization.

ALBERT L. KLEES.[3] In a study of the possibilities and economics of low-temperature carbonization processes, the one factor which will perhaps be of the greatest importance in determining the economies of this method of carbonization over standard methods will be the relative values of low-temperature tar and high-temperature tar. While this may not be particularly true in connection with the manufacture of prepared domestic fuel from bituminous coals, it is undoubtedly of prime importance in connection with the low-temperature carbonization of coal for gas-making purposes or for power development.

While considerable time and money have been spent during the last eight years in this country in investigating the possibilities of low-temperature tar, it is only during the last 18 months that one has been able to visualize a tangible relative value of this material in comparison with high-temperature tar. A few years ago, in setting up a budget analysis of a low-temperature carbonization process, it was even difficult to substantiate a price for low-temperature tar comparable to that of high-temperature tar. Today, because of the great progress that has been made during the past 18 months, one can conservatively estimate a considerably higher price for low-temperature tar.

Of course there is considerable discussion as to the present and potential value of low-temperature tar products, but the writer feels confident that any one who is thoroughly familiar with the laboratory and semi-commercial and commercial tests that are being made with products of low-temperature tar and familiar with the viewpoints of the manufacturers that are using this material and the success that they are having, can no longer be in any doubt or have any question as to the basic value of low-temperature tar in comparison with high-temperature tar.

While it is always difficult and dangerous to mention specific prices, it may be helpful to have some tangible indication of the relative prices of low-temperature tar and high-temperature tar in accordance with the way this situation looks to men who have been studying this problem for several years.

There is a sound basis for believing that low-temperature tar in comparison with 6-cent high-temperature tar is worth and will be worth a minimum of 10 cents per gallon. This figure is based not only on the present value of low-temperature tar in small quantities, but takes into consideration the tremendous production and consumption of low-temperature tar and the probable reduction in price that will have to be made when this material is used for large-quantity purposes.

The picture today of low-temperature carbonization is rather clean-cut. We can definitely see a price of 10 cents a gallon or better for low-temperature tar, and on this basis the processing of coal by low-temperature carbonization methods can be economically projected for the three major divisions—namely, domestic fuel, gasification of coal, and the production of char in the power plant.

At this time the picture is economically sound for the application of low-temperature carbonization to the manufacturer of domestic fuel. In the near future it should be applicable to the processing of coal prior to gasification, and in the not-distant future it should become economically feasible in connection with the processing of coal before such fuel is burned under the power-plant boiler.

[3] Assistant General Manager, Combustion Utilities Corporation, New York, N. Y. Mem. A.S.M.E.

FSP-52-23

Smoke-Abatement Methods in Knoxville

By F. L. WILKINSON, JR.,[1] KNOXVILLE, TENN.

Classed as one of the smokiest cities in the Southeast, Knoxville has been making a determined effort to clear its smoke-laden atmosphere. Methods used to enlist the cooperation and support of the public, to establish a bureau for smoke regulation, and to enact necessary measures to carry out recommendations are discussed in this paper.

KNOXVILLE has for many years been classed as one of the smokiest cities in the Southeast. Its rapid growth from a city of 36,000 people in 1910 to approximately 135,000 in 1928, with little or no control over the direction or class of its expansion, has augmented the atmospheric pollution due to smoke.

The city is located on the banks of the Tennessee River, is surrounded on all sides by the foothills of two ranges of the Southern Appalachian Mountains, and is within easy freight haul of the rich Southern Appalachian bituminous-coal fields. The city is built on a series of ridges and valleys, and its streets are narrow and crooked, as is common among old cities that have expanded rapidly with little or no attention to city planning. Its proximity to the Eastern Kentucky and Eastern Tennessee coal fields has made bituminous coal almost the only fuel for the generation of heat and power. The cheapness of coal has constituted a decided incentive to almost a total disregard of efficiency in its use, resulting in a marked fuel wastage and a decidedly smoky atmosphere. The locality has an average yearly wind velocity of only six miles per hour, making it difficult to dissipate the river fogs so prevalent during the winter months.

The agitation for smoke abatement in Knoxville began in 1911, when pressure was brought to bear on the city council to pass an ordinance prohibiting "the emission of dense-gray or dense-black smoke" within the city limits. After a futile attempt to enforce this ordinance on the citizens by the building inspector, who was also the smoke inspector by virtue of the ordinance passed, the city council remitted all fines that had been adjudged under the act and let it lie dormant, but still on the books, hoping that some day it would gain new life under different auspices.

At the time of the passage of this early ordinance, there were many good citizens who were anxious to comply with its provisions. These fell dupes to the salesmanship of a group of men who found Knoxville a "happy hunting ground." They found eager purchasers for everything from water tanks for washing the smoke, at several thousand dollars, to patent salts that when thrown upon the fire were guaranteed to cleanse the smoke of all evil. The results of this early and undirected effort at smoke abatement could be expected to leave a very bad taste in the mouths of many people, and smoke abatement would seemingly have a very hard path ahead of it. This short history of the early efforts of Knoxville to clean its atmosphere is only given here as a contrast to the situation as it appears at present.

The agitation for the passage of the present ordinance covered a period of approximately five years. A Smoke Abatement League was formed by a group of earnest and hard-working women a short time prior to the advent of the city manager-council form of government in 1923. These women, drawn largely from the women's clubs of the city, finally enlisted the aid of every representative group of citizens in the city. The civic clubs, Technical Society, Manufacturers' Association, Retail Merchants' Association, local section of the A.S.M.E., local chapter of the N.A.P.E., and other organizations gave active and financial support to their efforts.

The chemistry department of the University of Tennessee conducted a soot-fall determination that awakened the public to the immense loss attributed to a smoke-laden atmosphere. The monetary loss due to smoke was variously estimated at from $15 to $17.50 per person annually.

The Smoke Abatement League had its "ups and downs" for several years until, with perseverance, the entire public was aroused to the necessity of doing something in an attempt to abate the smoke nuisance. Mass meetings were held at which doctors outlined the effect of smoke on health, merchants gave data concerning the yearly loss due to damaged stock, housewives testified to the drudgery of housekeeping within the city limits, representatives of modern industrial plants assured other industrial managers of the benefits to be derived from modernizing power-plant equipment, and engineers advised the public at mass meetings and through the press of the best way in which smoke reduction could be accomplished.

As a result of the concerted efforts of all agencies lasting over a period of approximately five years, the city council was moved to action and appointed a committee composed of councilmen, laymen, and engineers to study the problem, and to suggest legislation and a course of action for the municipality to follow. This committee recommended the adoption of the standard ordinance of the A.S.M.E. with slight changes to meet local needs. This ordinance was passed in March, 1928, and active work started by the Bureau of Smoke Regulation in May of the same year.

Some of the distinctive features of the ordinance are as follows:

1 It provides for a Bureau of Smoke Regulation under the Department of Public Welfare with a technically trained man at its head.

2 It provides for an advisory board to serve without pay, of not less than five members, three of whom shall be mechanical engineers, who shall act as advisors to the chief engineer.

3 It provides that the ordinance shall apply, in substance, to all fuel-burning equipment except in a private residence.

4 It prohibits the emission of smoke equal to or greater than No. 3 Ringelmann chart for a period in excess of two minutes in any fifteen, except for a sufficient period of time necessary for starting and cleaning fires. The time limit is reduced to one minute in any eight for locomotives, steamboats, steam rollers, etc.

5 It provides that a permit be obtained for the installation of new fuel-burning equipment or for alterations and repairs to existing equipment.

6 It provides penalties for owners or contractors installing equipment without obtaining a permit from the chief engineer of the Bureau of Smoke Regulation. No charge is made for these permits.

7 It provides that owners shall make changes recommended by the chief engineer, looking toward the reduction or elimination of smoke, within thirty days of the date the recommendations are made.

8 It provides that the owner or contractor shall have the right to appeal to the advisory board from the decisions of the

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[1] Chief Engineer, Bureau of Smoke Regulation. Mem. A.S.M.E.

Presented at the Third National Meeting of the A.S.M.E. Fuels Division, Philadelphia, Pa., October 7 to 10, 1929.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

chief engineer, and further provides that the ruling of the advisory board shall be binding on the chief engineer.

9 One long section of the ordinance contains recommendations as to new installations concerning setting heights of boilers, etc.; in fact, it establishes a code that shall serve as a guide to the purchasers and manufacturers of equipment. This code was first proposed as a requirement, but was wisely changed by the council to act only as a guide. The wisdom of this change has been brought out even in the short time of the ordinance's existence due to the constant advance in the design of heating boilers and furnaces.

10 The ordinance further provides that no prosecutions can be made unless instituted by the chief engineer of the Bureau of Smoke Regulation.

The outline of the preliminary campaign for smoke abatement in the city of Knoxville is given merely as an illustration of the necessity of arousing public sentiment for the undertaking. Unless the residents of a community want smoke abatement and are made to be willing to do their part, the engineers and the municipality can do very little, no matter how many ordinances are passed nor how many enforcement officers are appointed. The public of Knoxville were ready to lend their aid, and were awaiting the word from some authoritative source when Mr. John Hunter arrived in June, 1928, to act as consulting engineer for a period of three months, and to train an engineer to fill the office of chief engineer of the bureau.

A survey of industrial plants had been made prior to the arrival of the consulting engineer, and work was commenced immediately with a view to eliminating smoke. No expensive surveys were made to determine how much smoke was being emitted. Every chimney and smoke stack was considered a potential offender, and investigated. Recommendations were made based on sound engineering knowledge and practice. The work of educating the public was continued, and still continues through newspaper articles, talks before clubs and group meetings, firing schools, and radio.

The municipality was made to realize that the first move would be that of "cleaning its own front porch," with the result that today no soft coal is burned in the city's boilers except in underfeed stokers and on downdraft grates. Coke is used in all other heating and power boilers, with the result that the city itself does not own a single smoking stack. This municipal cooperation is considered very essential in obtaining the cooperation of the general public.

The following methods have been used largely in the operation of the campaign in the city of Knoxville. The first year has been devoted to recommending changes in equipment or to existing equipment and to changes in fuel, with little attention paid to ordinance violations. The different classes of offenders will be taken up separately and suggested methods of insuring smoke reduction in each class discussed.

Industrial and Commercial High-Pressure Plants

The first summer was devoted to the industrial and commercial plants operating boilers at a steam pressure in excess of 50 lb. This group included all types and sizes of boilers, from the small vertical boiler so common in the small dry-cleaning plants, to the largest water-tube boiler. The majority of the industrial-plant boilers, however, were of the horizontal-return tubular type of approximately 150 hp. Of these, it is safe to say that fully 75 per cent had been installed with little regard for furnace efficiency, their settings being characterized by a seeming attempt to bring the boiler shell as close to the firebox as possible. It seems to have been the practice in industrial plants, where little or no engineering supervision was had in the design and installation of the boilers, to require a setting height that allowed only 30 in. from the dead plate to the boiler shell, and in many cases older installations have been found as low as 24 in.

The author feels that this state of affairs has been brought about by some of the boiler manufacturers themselves, as they have been entirely willing to set a return tubular boiler at any height that the customer desired. This practice might be condoned where boilers were installed about twenty years ago, but many of the low settings found in Knoxville were of recent construction and their heights seemed inexcusable. It was found in many cases that the small-plant owner had trusted the boiler manufacturer to give him the setting height he needed. The need for sufficient furnace volume for the combustion of high-volatile coal has been very difficult to impress upon many of the managers of small plants. The answer has been invariably, "If we needed more furnace volume, why didn't the boiler manufacturer give it to us?"

Space limitations prevented increasing furnace volume in many cases, but a simple form of steam-air jet was used to bring over-fire air into the boiler furnace at high velocities to accelerate mixing at times of stoking. The jets consist simply of $2^1/_2$-in. cast-iron or steel pipes carried through the front wall of the boiler, and inclined at an angle to make the stream of air strike the bridgewall at about eight inches above the rear of the grate. Into each pipe was inserted a $^3/_8$-in. nipple connected to a $^3/_8$-in. header to which was admitted steam at full boiler pressure from a $^1/_2$-in. connection to the main steam line. The nipples were capped with a simple pipe cap into which a $^1/_8$- or $^3/_{16}$-in. hole had been drilled. The position of the nipple in the air pipe depends upon the amount of air necessary to insure combustion of volatiles distilled off at the period of firing. Steam is used only as an agent for carrying the air into the furnace and giving it necessary velocity and turbulence. No claims for great economies have been made in their use, but in several cases slight reductions in coal consumption have resulted.

The main difficulty met with in air-jet installations has been the difficulty in getting the firemen to use them, as there is no denying the fact that, unless muffled, the noise is objectionable. There is no license law for operators in the state of Tennessee, and as a result, the average small-plant manager has considered that the cheapest man that he can obtain as an operator is the most economical. This idea is rapidly being overcome, and managers are beginning to realize the economies that can be obtained in the boiler room from efficient personnel.

Many of the larger plants are installing underfeed stokers, and others are planning to do so as time and money permit. Smoke abatement in Knoxville has resulted in fifteen underfeed stokers being installed in industrial plants during the past year, and two of the largest plants have completely scrapped their boiler equipment, and are now installing new boilers and stokers at a cost of approximately $90,000. The installation of jets is considered merely as a temporary expedient to abate smoke immediately, and they are proving to be the forerunners of more modern fuel-burning equipment.

In the smaller vertical and firebox boilers, coke is resorted to as a fuel, and in some cases where the steam pressure is above 50 lb., a jet is introduced to the firebox through the furnace door, with holes cut in the door to admit over-fire air. Where coke is used, the engineers of the Bureau of Smoke Regulation have given advice and assistance in its use, and it is becoming a popular fuel for the vertical and firebox boiler, especially where the fireman has other duties to perform around the plant.

In making the recommendations to the plant owners, the bureau engineers try to drop useful hints to the management with a view to effecting fuel economies beyond those incident to the elimination of smoke. This tends to make the bureau popular and to increase the desire of the management to cooperate.

At the present time, it is safe to say that virtually every industrial plant in the city is capable of operating within the ordinance requirements. The author does not mean by this statement to convey the impression that there is no smoke from industrial plants. The fact that there is smoke means that the operating personnel is not using the necessary precaution. At the present time, the bureau does not have sufficient personnel to insure proper operation at all times, but it is hoped that this condition will be remedied during the coming year, and more pressure can be brought to bear on the operating personnel.

Large Heating Plants

The heating plants of Knoxville's business sections have been the dumping grounds for any and all boilers, the straight-draft cast-iron boiler predominating. In comparison with other cities of its size and larger, coal is the cheapest commodity by comparison on the local market. As a result of its very low price per ton, the cheapest coal has been the rule for use in the downtown business houses and office buildings. The result of smoke emission can be realized easily when run-of-mine bituminous coal, running as high as 40 per cent volatile, is burned in anthracite heating boilers. That this class of fuel user has been quite a problem during the past heating season is evident.

Many experiments were tried with the use of better grades of coal, repairs to boilers, changes in grate area, improved firing conditions, and better draft control, and it has been found that while in many cases conditions have been improved and operation can be maintained within ordinance limits, satisfactory results from the standpoint of smoke elimination cannot be had in the straight-draft low-pressure heating boiler unless a smokeless fuel is used or an oil burner or small stoker installed.

It was found that the majority of the owners of heating plants were willing and anxious to do their part. They knew better than any one else the cost to them of a smoke-laden atmosphere filtering through their doors and windows, spoiling stocks, and adding to building cleaning costs. Many changes of fuel were made during the winter, coke being used extensively. Particular attention was paid to instructing janitors in the use of coke, and with careful firing under the supervision of the bureau's engineers, it was shown that coke could be used as economically as the better grades of bituminous coal. Fortunately for smoke abatement there is only a price differential of approximately 20 per cent between coke and the better grades of bituminous coal on the local market.

Sixteen motor-driven unit stokers have been installed, or are being installed in the larger store and office buildings, and the saving resulting from their use this past winter is expected to result in many future installations. One department store alone reports a saving of 45 per cent in fuel costs during the past season with an underfeed unit stoker installed under a cast-iron boiler.

Many of the older buildings in the downtown section are small two- or three-story structures, overtopped by large office buildings. Even should these buildings operate within the limits prescribed by city ordinance, their low stacks, augmented by fog and low wind velocities, would make them quite objectionable in the congested areas. As a remedy for this condition, a block heating system has been proposed and designed by the bureau, several units of which are now under way.

Six of the worst offending city blocks are commanded by large office buildings, department stores, or hotels. Each of these buildings is equipped with duplicate boilers, each of sufficient capacity to carry the load of the entire building. It has been proposed to connect them with the smaller buildings in their blocks, and have them supply steam for heating purposes. This system when completed, and if entirely carried out as planned, will eliminate 34 chimneys within the congested downtown section. The owners of the supplying agencies have been shown that even if their equipment is operated at a loss, the returns from reduced cleaning costs and stock losses will more than pay for the trouble and expenditure. The cooperation of the supplying agencies has been freely given, and it is believed at the present time (July) that three of these units will be in operation by November, and the others will fall in line as soon as enough customers can be assured to make the plan operative.

Financial organizations are being interested in the establishment of a central heating plant for the entire congested area, and the lines for the block systems are being laid so that they can be connected to the low-pressure mains of a large central-heating system. It is hoped that the block-system units will be a nucleus for the larger system.

A record has been made of all the larger and offending heating plants. Letters were sent in the early spring to all owners, making recommendations covering fuel, furnace changes, and repairs that should be made during the summer months in order to bring their plants within the ordinance limits during the next heating season. Although these recommendations had been made previously, by getting them on record in the early spring and before the winter's fuel was purchased, the bureau will be able to bring concrete evidence before the court, should occasion arise, that the owners had been sufficiently warned well in advance of the heating season. Answers to many of these letters have been received assuring the bureau of the owners' desire and entire willingness to cooperate by carrying out its recommendations.

Procedure With Railroads

Knoxville has a very decided smoke menace from the two railroads serving the community. Both lines enter the heart of the city, traversing its full length and cutting it almost in half. As industrial plants line the railroads, switching operations are carried on day and night within the heart of the city. Most of the through freight has been shunted around the city within recent years, relieving the smoke nuisance somewhat from that source. Nevertheless, the highest soot-fall records have been recorded within the business districts adjacent to the railroad lines.

Little was done in the early months by the Bureau of Smoke Regulation to combat the smoke nuisance from this source. The cooperation of the railroad officials was requested, with the result that each road appointed a supervisor to look after smoke abatement in its own organization. Air jets and refractory arches were installed by both railroads in all switching locomotives. These have been effective in reducing smoke emission, but again the operating personnel has been careless in the use of the equipment supplied them. The bureau attempted during the past winter to send each week to the different roads records of smoke-density readings taken as time permitted. Each case has been handled by the road, and penalties invoked on the crews where carelessness and failure to use the jets have been shown.

With the coming of spring and the release of an assistant engineer from heating-plant work, a new system has been adopted. The assistant engineer spends several days with the railroads observing firing methods and working conditions of the locomotives and recording smoke densities. The records obtained, both good and bad, are sent to the officials as before, but a monthly grade sheet is made of the records taken during the month, and each locomotive is given a standing in smoke emission for that month. These grade sheets are presented at a monthly meeting between representatives of the bureau and the railroads. Problems relating to smoke abatement and fuel economy are discussed at these meetings. The locomotive crews are given the opportunity to attend, and a spirit of rivalry among the crews is being fostered. Although this system has been in operation for only a short time, it is believed that it will bring beneficial results to the railroads

and to the bureau. Nothing is said about ordinance violations, and locomotive crews are endeavoring to keep their grades low, which means, in effect, a minimum of smoke emission.

Residence Educational Work

Although the smoke-abatement ordinance exercises no control over the smoke from residences, as much if not more educational work has been done among this class of smoke maker than any other. It is, in the author's opinion, useless to say that the smoke from domestic furnaces and boilers can be eliminated unless a smokeless fuel is burned or soft coal burned in one of the successful automatic underfeed stokers now appearing on the market.

Of course, any reduction of smoke is thankfully received and strived for, but it is almost impossible to fire bituminous coal by hand in the average domestic boiler or furnace without objectionable smoke emission. The author understands that furnaces and boilers are now appearing on the market for domestic use, designed to accomplish this purpose, but as yet they have not been found in any of the homes under his jurisdiction, nor has he ever seen such a unit in operation.

In the effort to obtain as much smoke reduction as possible, and smoke elimination when possible, an intensive educational campaign was carried on during the past heating season. Talks were given for a few minutes over the radio each week for several months. Articles were printed in the local papers urging the individual responsibility of each citizen to effect smoke abatement and recommending effective means to bring this about. Articles have been continued in the press during the past summer, keeping smoke abatement alive when fires were dead. Instruction cards were placed in the homes of the city by Boy Scouts, illustrating most successful methods of firing coal and coke to bring about smokeless or near smokeless combustion. Pamphlets entitled "The Smoke Nuisance and Its Abatement," prepared by John Hunter, were placed in all public places with placards asking that one be taken. Large cards were placed on the front and rear of all street cars impressing on each individual his responsibility in the work for cleaner air. Home owners were urged through the press to call on the Bureau of Smoke Regulation for individual assistance in helping with their fuel and heating problems.

The above methods are believed to have brought results. Coke which had previously been a drug on the local market has been used extensively. Coke sales increased over 300 per cent and the local gas company has reported that a very large amount of their coke output has been contracted for the coming winter. By-product coke has been shipped in from Birmingham and Chattanooga, and sold at a price in competition with the local product. The gas company has installed many gas burners in the homes of the wealthier class, and oil-burning equipment has gained a strong foothold.

The retail coal merchants have been called upon to assist in the domestic campaign, and while a few of the less prominent dealers have not been won over to smoke abatement, the larger and more important ones are assisting admirably by recommending, and making an earnest effort to see the right kind and size of fuel go into the home. One large company has employed a fuel advisor to give free service to the home owner, and two others have assumed the agencies for approved types of domestic stokers, and are waging an effective campaign for their use.

The campaign in the home has brought encouraging results, but there is still a menace to combat within the residential area. Plans have been laid to intensify efforts in this direction during the next winter. A school for home owners has been projected for the early fall to be conducted through the afternoon paper. Citizens will be asked to enroll in this school. Lessons or lectures will be published daily and questions submitted will be answered each day. Following this educational week, there will be a city-wide "Clean Your Chimney Week" sponsored by the Citizens' Smoke Abatement League. Prizes will be offered school children for essays on smoke abatement, and for posters calling attention to the evils of a smoke-laden atmosphere. Smoke-abatement programs will be requested of all organizations, and radio talks made by leading citizens. The effort will be made to intensify public opinion, and make them smoke abatement conscious.

The cooperation of the press has been splendid during the entire campaign and requests for copy sometimes have been more urgent than it has been possible to comply with.

New Construction Improved

New construction and alterations to old installations have been handled by the permit system. All heating units, large and small, except in private residences, must pass the approval of the Bureau of Smoke Regulation before installation is started. No charge is made for this permit, but the ordinance provision requiring that it be obtained is enforced rigidly. The only case brought to court during the past year was due to a violation of this provision. A contractor was brought before the municipal court for failure to obtain the necessary permit, after having been warned. He was given a fine and appealed his case to circuit court, where the fine was upheld. Some difficulty was met with at first in getting the contractors to obtain the necessary permits, but since the above court case, their cooperation has been improved greatly.

The bureau makes no attempt to specify the make of boiler or furnace that can be installed, but it does assume the authority of specifying types. The bureau has taken the stand that no boiler or furnace will be permitted until it has proved its ability to function within the limits prescribed by city ordinance. Where a permit is requested for a boiler that is not known to the engineers of the bureau, and the manufacturer claims smokeless operation, its installation is permitted subject to test. If it cannot be fired under average operation within ordinance limits, it must be removed and permission for its future installation will not be granted.

The architects have been informed of the bureau's requirements, and their cooperation in specifying accepted types has been evidenced in all new construction. During the past summer the construction of three large buildings has been begun, the plans and specifications having been made by three different local architects. It is significant that the heating boilers of each of these buildings will be equipped with motor-driven underfeed stokers.

Experience in Knoxville with high-volatile bituminous coal has led the bureau to disregard the claims of manufacturers that their equipment has met the approval of smoke-abatement organizations in other cities. This is not meant as a reflection on either the manufacturers making such claims or on the engineers engaged in smoke-abatement work elsewhere. The author simply believes that the conditions in Knoxville are not enough like those in other cities to take the approval of smokeless operation in other cities as a criterion for smokeless operation in Knoxville. For that reason the Bureau of Smoke Regulation has its own list of approved types and adheres to it strictly, permitting no others unless the installation is made subject to test or the owner makes a written agreement to burn a smokeless fuel.

Types of Smokeless Boilers and Combustion Equipment

The experiences of the Knoxville bureau with different types of boilers and combustion equipment are given here as a matter of information.

1 *Straight-Draft Boilers.* (*Cast Iron and Steel.*) This type of boiler is not permitted in the sizes where approved boilers will apply. When permitted, the owner must burn a smokeless fuel.

2 *Smokeless Boilers With Refractory Arch.* It has been found that it is possible for this type of boiler to be operated within ordinance limits only with the most careful of fireman and the very best grade of coal. It is certain to smoke with local coals when operated at low ratings and low firebox temperatures. For these reasons permission for its installation is not granted.

3 *Smokeless Boilers With Air Admission Through Arch.* The later designs of this type of boiler are being operated within ordinance limits with a clean grade of coal and careful firing. Permits are issued for its installation, but particular attention is always given to instructing the fireman in the proper methods of operation to produce smokeless results.

4 *Downdraft Boilers.* The downdraft boiler has proved itself the best type of hand-fired boiler for smokeless operation. It is essential, however, to insure sufficient draft and a clean coal, even if the boiler is fired continually on the upper grates. The author knows of a number of cases where run-of-mine coal is used with considerable success in this type of boiler, but in each case the fireman is superior to the average, and works his upper grate in a manner to insure freedom of holes through which the fines can fall to the lower grate.

5 *Underfeed Stokers.* By far the greatest boon to the smoke-abatement engineer is the motor-driven underfeed stoker for low-pressure work with bituminous coals. The cheaper fuel and the smaller sizes used make them popular with the owner and the smoke-abatement authorities. They are gradually being reduced in size until their application is bringing splendid results in all sizes of boiler furnaces from the smallest to the largest. Some very great claims were made for these appliances in the early days of their appearance, and dissatisfaction was experienced due to salesmen's claims. The author was called by a new stoker owner, and requested to tell him what was the matter with his unit as it was giving considerable trouble. It was found that the stoker had been sold with the assurance that coal would be burned with "no smoke and no ash." As a result the ash and clinker had accumulated all the way from the grate surface to the boiler shell, and the owner was very much disturbed because his ashes had not burned up completely as he had been led to believe.

With a reasonable amount of care, proper setting height and average intelligent operation, the unit underfeed stoker will prove itself quite useful in smoke abatement.

Fines and Penalties

The provisions of the ordinance invoking fines and penalties on offenders have not been enforced during the first year except in the one case noted.

It has been thought that the public needed educating more than it did penalizing, and all efforts were made to that end. Some harsh warnings have of course been necessary to bring unwilling owners to make necessary changes in equipment, but with the exception noted, no court cases have resulted.

Up until the present time the bureau has not had sufficient personnel to exercise a constant supervision over smoking stacks. During the past summer a closer supervision has been kept over industrial stacks. When a violation is noted an immediate inspection is made of the boiler room, and the owner is given a notice containing an exact record of the smoke density and period of violation with a recommendation for improvement. Three notices will be given before court action is taken. Each notice seems to bring a decided improvement in conditions, and it is believed that the necessity of court action will be infrequent. The Bureau of Smoke Regulation is completely divorced from the police department and no prosecutions can be made except by the chief engineer.

Conclusion

The author has tried to refrain from discussing the question of smoke abatement from a technical standpoint. It has been his object to outline the work of the Bureau of Smoke Regulation in Knoxville, and submit their methods as a suggested means of approach in other cities confronted with a similar problem.

It is realized that Knoxville is not a very large city, and that a closer contact can be maintained with its citizens than in a city of much larger size. He does believe, however, that as close a contact as possible with the general public should be maintained in order to bring about worth-while results.

The small budget requested for the first year curtailed the activities of the bureau somewhat. An expansion is planned for the coming budget year, commencing in October, with a decided increase in personnel and a corresponding budget increase. Whether this will be allowed is not yet known. It is known, however, that in the face of a 20 per cent budget reduction ordered for the next year, smoke abatement will not be touched, unless the incoming council upsets the plans of the city manager. The chances of a decided increase are excellent.

While Knoxville is still classed as a smoky city, it cannot be denied that progress has been made. There are people that can see no improvement, and there always will be, but it is believed that the general public has been won for smoke abatement, and it is daily growing in favour.

A new soot-fall survey was started by the University of Tennessee in February, but due to the summer vacation data for only three months are at hand. While this survey has not been conducted for a long enough period to give an exact comparison, the total soot fall measured in tons per square mile per year shows an appreciable reduction over the 1927 survey.

The author, in conclusion, wishes to submit the following essentials for a successful smoke-abatement campaign:

1 A well-directed preliminary campaign designed to make the public want smoke abatement.

2 The passage of a complete ordinance that can be enforced without necessary hardships on those concerned.

3 The complete removal of smoke abatement from the police department.

4 The necessity of appointing an experienced engineer, not only to carry out the provisions of the ordinance, but to gain the respect and confidence of the public.

5 Reduce prosecutions to the minimum and prosecute only when other methods fail.

6 Start the work with a small budget and personnel, increasing it only after the work of smoke abatement has proved itself in the minds of the public.

7 Enlist the aid and support of the press. Without its aid little can be accomplished.

8 Most essential of all, keep smoke abatement free from political influences.

FSP-52-24

Coal Pretreatment

By CLARENCE B. WISNER,[1] NEW YORK, N. Y.

PRETREATMENT of coal must offer financial profits or economic advantages, or both, in order to function under present-day conditions. Assuming a sound business basis, important economic considerations include health, sunlight, and cleanliness. The health commissioner of Chicago is quoted as stating that 16 per cent of the deaths in his city are caused by smoke, and that its elimination would reduce the death rate from the present figure of 12 per thousand to 10 per thousand. It has been estimated that smoke cuts off one-third of the sunlight of New York City, and the annual damage to buildings, house furnishings, and decorations, added to the cost of cleaning, is said to total one hundred million dollars in New York City alone.

Loss of Fuel

The combustion of bituminous coal includes three separate and distinct actions, each centered about a different fuel. The fuels are tar, gas, and carbonized residue. Tar eduction is a distillation process with a time element in burning much longer than the gas or carbonized residue. When the tar molecule is released it is covered with a gas envelope requiring turbulence to bring it into intimate contact with oxygen. This often does not take place until the gases are in the chimney or until they have passed out of it. Thus smoke is caused by the breaking up of the gas unit at a reduced temperature releasing the carbon.

The gas contains a large portion of methane burning under about the same conditions as natural gas. Carbonized residue carrying volatile is free burning with rapid combustion. One of the products obtained from carbonizing coal is oil. Many large bodies of bituminous coal contain three-fourths of a barrel per ton. Next to petroleum this oil is believed to offer the cheapest raw stock for motor fuel. Petroleum derivatives occupy such an important place among the necessities and luxuries of life that the maintenance of a future supply is a serious matter.

Increased demand for petroleum products is such that the present consumption will be doubled in less than four years. Therefore, a pretreatment of coal, which will provide oil as one of the products, will unquestionably be found essential. This is on the assumption that the production of petroleum will not double in the next four years. In some quarters it is felt that the cleaning of coal meets present requirements. A study of the matter, however, indicates that this is not the case.

Coal Cleaning

In keeping with modern industrial progress, the operation of mining is performed with mechanical undercutters and loaders. Mining machines have no means of differentiating between coal and rock handled, and although the coal in place may be low in ash, more or less wall rock will be included with the loaded coal. The rock usually breaks finer than the coal when sized, thus throwing the highest ash content into the slack, which ordinarily means all that will go through a $^3/_8$-in. screen.

Slack coal, as burned in the big steam plants, on an average carries from 11 to 16 per cent ash. It is worthless as domestic fuel. If, as is claimed, 40 per cent of all coal mined breaks to slack, the need of processes for turning this enormous tonnage into an improved fuel is obvious.

The geologically older coals have been the first to be cleaned; the volatile content is quite stable, and spontaneous combustion is not a troublesome factor when stored. This is not true of the younger and higher-oxygen coals mined in the greater part of Ohio and the states west.

Operators complain that cleaning does not go far enough; that coal pretreatment must give a product strong and shipable to be attractive. With this understanding it will be seen

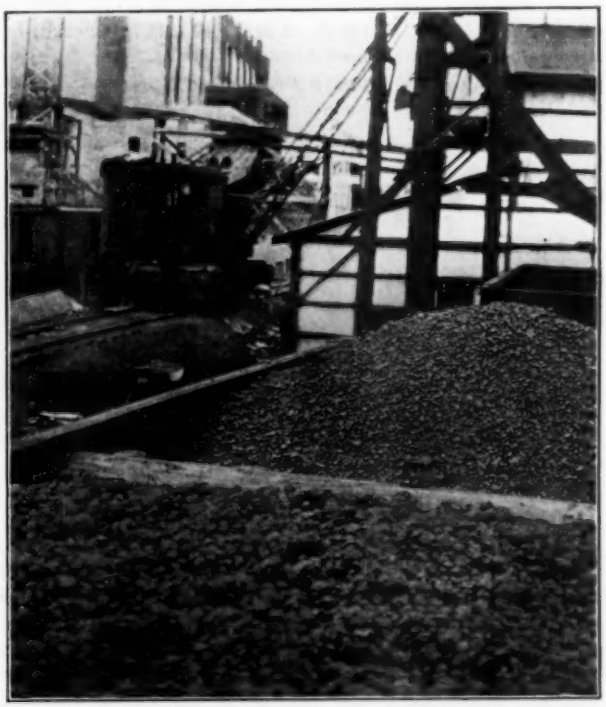


Fig. 1 A Carload of Coal Balls; Half Are Stove Size, Half Nut

that although coal cleaning is essential to the best use of coal for any pretreatment program, it does not play a major part, because it only reduces high-ash coals to a par with low-ash coals, of which there is not enough to go around. It has accomplished nothing toward reducing smoke or conserving by-products.

Low-Temperature Carbonization

It is not the purpose of this paper to discuss a process that has already been carefully covered,[2] further than to say that the coal-ball process is a two-stage continuous method of low-temperature carbonization founded upon two basically new principles.

---

[1] President, The Carbocite Co. Mem. A.S.M.E.

Presented at the Third National Fuels Meeting, Philadelphia, Pa., October 7 to 10, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

1 *Thermodizing*, modifying, and conditioning the coal in a thermal pretreatment in which oxygen may be brought into intimate contact

[2] Proceedings of the First International Coal Conference on Bituminous Coal, Carnegie Institute of Technology, 1926. These contain a description of the Carbocite process by which coal balls are made.

with heated coal while it is tumbling through an indirectly heated closed rotating container.

It prevents sticking to the retort during the plastic stage.

It controls the swelling, the density, and weight of the fuel.

It makes possible an ideal domestic fuel, coal balls.

It controls the maximum size of coal balls.

It carries into the coal two-thirds of the carbonizing heat required in the next step of carbonization, utilizing the waste heat from the carbonizer.

2 *High velocity of heating medium*, a double-shell retort, the space between the coal container and the outer wall forming the heating flue, limited in depth to draw the heating medium around the coal container at a velocity of 75–100 ft. per sec.

The demonstration plant at Philo, Ohio, has convincingly taught that capacity is in direct proportion to the velocity of the heating medium drawn over the wall of the container.

In these two methods, steps exclusively parts of the coal-ball process, lies the key to successful low-cost processing of all solid fuels giving off hydrocarbons at critical temperatures.

The gross fuel used in the process operating at 100 tons per day has been 700 B.t.u. per lb. of raw coal. It is believed that the fuel used will be reduced to under 500 B.t.u. per lb., when operating a 250-unit, bringing the efficiency up from 55.5 per cent to 78 per cent.

TABLE 1 RELATIVE SUPERIORITY OF SMOKELESS FUELS

Requirements	Anthracite, per cent	By-product coke, per cent	Pocahontas, per cent	Coal balls, per cent
Smokeless	100	100	75	100
Cleanliness	100	100	25	100
Low ash	25	50	75	50
Clinker	75	25	75	100
Fire control	75	25	100	75
Firing periods	100	50	75	75
Value to consumer, total points.	475	350	425	500

TABLE 2 SALES OF SMOKELESS DOMESTIC FUEL IN CLEVELAND, OHIO

	Anthracite	By-product coke	Pocahontas	Coal balls
Annual sales, 256,500 tons	60,000	20,000	175,000	1,500
Price in buyers' bin	$14.50	$11.25	$10.85	$12.00
Dealers' gross margin	3.25	4.00	4.50	4.00
F. o. b. dealers' siding	11.25	7.25	6.35	8.00



FIG. 2 STOVE-SIZE COAL BALLS MADE FROM PITTSBURGH VEIN COAL



FIG. 3 NUT-SIZE PRODUCT MADE FROM PITTSBURGH VEIN COAL



FIG. 4 A SUPERPOWER COAL-HANDLING UNIT (This plant cost in excess of $500,000. It has an unloading capacity of 500 tons or more per hour and distributes to storage or to the boilers. A car of coal is over the unloading hopper at the left. There is a duplicate unit to the right.)

It is now proposed to take up the practical use of such a product, and outline the experiences encountered. Sufficient sale and use has taken place so that definite trends have been established.

The prepared fuel known as coal balls is a smokeless product of bituminous coal. It has been produced in considerable quantity, and distributed to the consumer in competition with other smokeless fuels. The selling price was next to anthracite, but in spite of that it was very popular and the demand exceeded the amount which it was possible to supply.

COAL BALLS

In character the fuel is strong and dense. It has the same weight as by-product coke of equal size and made from the same coal. It has stood the test of being loaded with a clamshell bucket, of being dumped into the coal pockets of dealers, and of being delivered by truck, without complaint of degradation loss.

Coal balls are formed during the plastic stage of carbonization, while tumbling through a rotating container, and because of their oval shape they do not offer rough edges to crumble and form breeze. Of the product, 85 to 100 per cent may be shipped in perfect form as domestic fuel.

Table 1 was made by a mine operator and representatives of wholesale and retail distributors.

A preliminary test of this fuel for steam raising on chain-grate stokers indicates economies that may make it as useful for this purpose as for domestic fuel.

With 85 per cent to 100 per cent of the solid product in suitable state for sale as domestic fuel in competition with other smokeless fuels, the sale value of the by-products is not the impelling factor which it becomes when the coal is to be carbonized for the power house, because of the wide margin between the cost of slack and the sale price of smokeless domestic fuels.

When considering the latter the value of low-temperature oil becomes of major importance, because there must be delivered to the power house the same number of heat units in the carbonized fuel and surplus gas as were contained in the raw coal, to make it attractive. Whatever saving is made in decreased upkeep cost, extending to firing and degradation loss in stored reserves, will be accepted as a profit and reason for adopting processed fuel.

Following this basis of reasoning there is thrown upon the by-products the duty of producing a revenue equal to or exceeding (1) loss of B.t.u., extracted in the volatile hydrocarbons, (2) cost of processing, and (3) interest and amortization of capital.

Power-station fuel in Ohio, Indiana, and Illinois is generally noncoking, high in volatile hydrocarbons, and quite liable to catch fire in storage. Although these coals do not make a good carbonized domestic fuel without mixing with some coking coal, they do produce an excellent steam fuel when carbonized alone.

These coals are greatly improved as steam fuel by carbonizing, and as now mined with mechanical undercutters they produce a high-ash slack with an average heat unit content as burned of, approximately, 25,000,000 B.t.u. per ton. There will remain in the carbonized fuel 17,500,000 B.t.u.; in surplus gas, 1,125,000 B.t.u., and charged to process 25 per cent of the cost of the coal, 6,375,000 B.t.u. per ton. The cost of raw coal delivered to boilers in Ohio is $2.25, and in Indiana $3.35.

It is assumed that at the power plant no market will exist for the gas and that the surplus over processing requirements will be burned as fuel. There will be no need of duplicating the coal-handling and storage plants, so that it is believed that the cost of the carbonizing plant can be kept within $2 per ton of annual capacity.

TABLE 3 CHARGES THAT MUST BE BORNE BY THE RECOVERED OIL PRODUCT

25 per cent of the raw cost		$0.57	$0.84
Operating labor	$0.18		
All other operating expense	0.27		
Capital charges, 15 per cent on $2 ton	0.30	0.75	0.75
Cost of oil per ton of coal processed		$1.32	$1.59

The oil content will approximate 30 gallons per ton, and 2½ to 3 gallons of light oil from the gas.

The place for the power-house processing plant is at the unloading station. The coal passes through it with as little difficulty as over a belt conveyor. The carbonized fuel, positively cooled, is put back on the belt for delivery to boilers or storage.

It is believed that the true future market for the oil is as a cracking and hydrogenation stock for motor fuel, and that it will follow the price of petroleum for crude stocks of similar gravities. Additional research is of great importance at this point as indicated by the increased recovery of motor fuel from petroleum from 30 per cent to 45 per cent by means of cracking.

Low-temperature oil, by cracking, has produced 22 per cent water-white motor fuel of a superior anti-knock quality, and 13 per cent of tar acids. This is exclusive of the light oil recovered from the gas. The development of hydrogenation at atmospheric pressure applicable to this oil is well on toward



FIG. 5 DISTRIBUTING BELT WITH TRIPPER DISCHARGING STREAM OF COAL TO THE STORAGE WHARF

FIG. 6 END OF DISTRIBUTING BELT AWAY FROM THE POWER HOUSE WITH TRIPPER DISCHARGING TO STORAGE

FIG. 7 METHOD USED TO KEEP STORED COAL SAFE FROM SPONTANEOUS FIRING

For superpower steam fuel it has been shown that there must be assured at least $1.32 in Ohio and $1.59 in Indiana from the oil by-products before the cost of processing is justified. If 30 cents be taken as the value of the light oil recovered from the gas, then the 30 gallons of heavy oil must sell for $1.03 for Ohio coal and $1.29 for Indiana coal; in other words, for from 4 cents to 5 cents per gallon.

Time contracts have been made at a higher price than this, but the supply has been small, and this may not represent a permanent price for unlimited supply.

This low cost of operation is made possible by operating as a department of a going business with an engineer-superintendent in charge and includes:

Working shift bosses
One extra chemist in laboratory
One control room with one operator for each 1000 tons capacity
Eight men on shift
Ordinary repairs handled by shift crews
Major repairs by power station machine-shop crew on hourly basis
No hand labor in the flow sheet from raw coal to finished products

This cost estimate is based on actual operation.

commercial use. It may open a highly profitable outlet for these oils.

Another point to be borne in mind is that the preferable location for carbonizing plants will be in communities of considerable size, which require large quantities of motor fuel. To make use of motor fuel produced from low-temperature distillation and refined at the plant would therefore save freight on the crude out and the refined in; the price approaching closely filling station prices.

As regards the domestic market there is little reason to doubt the permanency of demand. Experience has shown that the domestic uses will absorb a large volume of carbonized fuel, and that this is possible with relatively little comeback or dissatisfaction.

The margin between the price for slack coal and the selling price of smokeless fuels is great enough to warrant large production. The fact that about 40 per cent of all bituminous coal breaks to slack size, provides material for large production at reasonable figures. Much of this slack is not utilized at the present time, but it is beyond question that as the tonnage of processed fuel increases, the price to the consumer will decrease, and this in turn will expand the demand for the product, provided these manufacturing advantages result in greater quantity production.

FSP-52-25

A Graphic Method for Determining Expansion Stresses in Pipe Lines

By C. T. MITCHELL,[1] NEW YORK, N. Y.

The modern engineer recognizes an expansion problem in every pipe that carries steam. Increasing temperatures and pressures may compel the labeling of every important joint in the piping of a power plant with the bending stress allotted to it, the same as the stress diagram of a truss. The paper describes a serviceable method for determining the stresses. The claimed advantages are that it is graphic and thus free of tables and formulas; it is concerned only with the entire pipe structure from one anchor point to another; it is a standardized method applicable to any pipe structure in one or more planes; it involves less work with greater accuracy.



AN expanding pipe line gives no visible evidence of the terrific bending strain that it may be exerting on a certain joint—no evidence but the sudden escape of steam. The blame for failure of a joint is frequently laid to the gasket or to the bolts or to the pipe fitter.

If a pipe joint could be furnished with a guarantee as to the amount of bending strain it would stand before beginning to leak, due care would be exercised to see that it was not overstressed. But since no such guarantee is available is it not logical to treat the joint with all the more consideration?

The practice of installing large loops on the steam headers and allowing the rest of the piping to take care of itself is nearly obsolete. The modern engineer recognizes an expansion problem in every pipe that carries steam, whether he takes the time to analyze it or not. Increasing temperatures and pressures may soon force him to label every important joint in the power plant with the bending stress allotted to it, in similar fashion to the stress diagram of a truss. There is a tendency in this direction now, but very little system in carrying it out, partly because of the lack of a serviceable method for determining the stresses.

In this paper an attempt has been made to supply this deficiency. The method to be described is recommended as having the following advantages:

(1) It is almost entirely graphic and is thus independent of tables and complicated formulas.

(2) It is concerned only with the entire pipe structure from anchor point to anchor point, and thus avoids the uncertainty of trying to piece together the supposed deflections of individual bends.

(3) It is a standardized method, applicable to any independent pipe structure, whether in one or more planes.

(4) It involves less work than any other known method for the same degree of accuracy.

(5) It has been in active use for two years by the author and thus has been proved out as a practical working method.

Two broad assumptions are made in every problem covered by this method: first, the ends or anchor points of a structure are assumed to be 100 per cent fixed—that is to say, the direction of the axis of the pipe where it passes through the anchor does not change; second, the moment of inertia of the pipe is assumed to be constant, whether the section be taken in a bend or elsewhere.

It has been found that any assumptions other than these involve an amount of labor that is entirely out of proportion to its value. Furthermore, there is no means for determining the degree to which a pipe end is fixed nor the amount of flattening of a pipe section in bending. The very uncertainty regarding these matters makes it superfluous to aim at high accuracy. It is better to maintain a good average in all problems than to hit the bullseye in one problem and be wide of the mark in the next. A structure proportioned on the foregoing assumptions will be conservatively designed. The cost of such conservatism is believed to be negligible.

Two other features of this graphic method deserve comment, since they are at variance with the usual practice: first, no attempt is made to determine the horizontal and vertical reactions at the anchor points, for that only complicates the analysis, the only force dealt with being the one which passes through both anchors, whether the anchors are on the same elevation or otherwise; second, all bends are eliminated from the structure during the analysis, and square corners are substituted in their place, a correction being subsequently made to allow for the influence of the bends. This procedure does away with the most laborious element of these problems. Its justification will be demonstrated at the conclusion of the paper.

For the benefit of those who are willing to accept the method on its face value a few typical problems will first be worked out in detail. These will be followed by the theoretical derivation. There is nothing new in the theory. It is as old as the equation of the elastic curve. It should be accepted as a necessary adjunct of this paper, however, since those who are conversant with the theory will make the most intelligent use of the graphic method.

Notation

- P = thrust between anchors, lb.
- Δ = expansion of pipe between anchors, in.
- E = modulus of elasticity, lb. per sq. in.
- I = moment of inertia of pipe, in.[4]
- M = sum of moment areas times distance of center of gravity of each to neutral axis, cu. ft.
- S = stress in outside fiber of pipe, lb. per sq. in.
- P_c = thrust corrected for bends on pipe, lb.
- R = shortest distance from neutral axis to point of greatest bending moment, ft.
- D = outside diameter of pipe, in.
- d = inside diameter of pipe, in.
- T = total length of pipe with square corner bends, ft.
- t = total length of pipe with full radius bends, ft.

[1] Allied Engineers, Inc. Mr. Mitchell was born in August, 1878. He was graduated from Sheffield Scientific School, Yale, 1901, in mechanical engineering, and then was associated for several years with Western copper mines and smelters in general engineering work. Since 1916 he has specialized on power-plant engineering with J. G. White Engineering Corporation, Electric Bond and Share, and Allied Engineers, Inc.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

A, B, C, D, E, F, G = moment areas, sq. ft.
$a, b, c, d, e, f, g, l, m, n, o$ = distances on sketches, ft.

ASSUMPTIONS

All pipe ends 100 per cent fixed.
Moment of inertia of pipe constant at all sections.

FORMULAS

$$I = \frac{\pi}{64}(D^4 - d^4)$$

$$P = \frac{\Delta EI}{1728\,M}$$

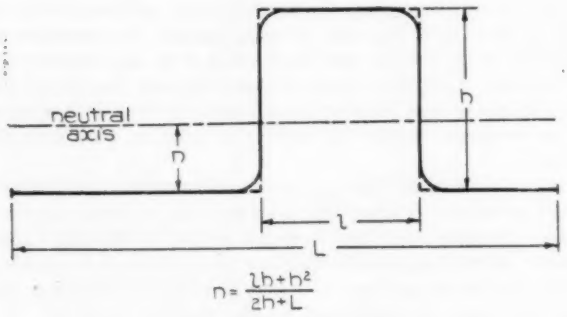


FIG. 1

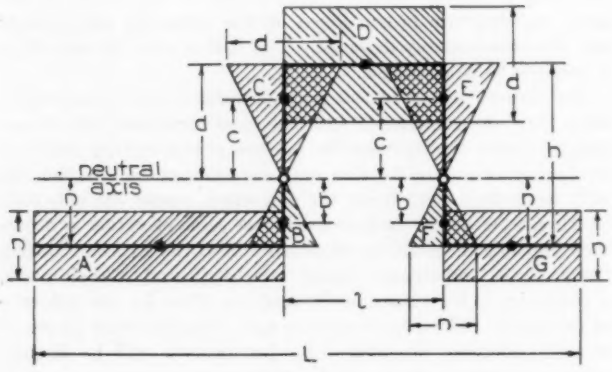


FIG. 2

For determination of M, see cases below.

$$P_c = \frac{PT}{t}$$

$$S = \frac{6P_cRD}{I}$$

DETERMINATION OF M

Case I, Figs. 1 and 2.

(A) Locate neutral axis. The neutral axis passes through center of gravity of shape and is always parallel to the line connecting the end or anchor points, as shown in Fig. 1.

(B) Draw moment areas A, B, C, D, E, F, and G, as shown in Fig. 2. These areas should be drawn about the member to which they apply, the member bisecting the area.

(C) Compute square feet in moment areas.

(D) Compute M by multiplying each moment area by its respective moment arm. The moment arm is the perpendicular distance from the center of gravity of the area to the neutral axis.

$$M = An + Bb + Cc + Dd + Ec + Fb + Gn$$

Case II, Figs. 3, 4, and 5.

(A) Locate neutral axis parallel to line connecting pipe ends, as shown in Fig. 3.

(B) Draw moment areas A, B, C, D, and E, as shown in Fig. 4.

(C) Compute square feet in areas A, B, C, D, and E.

(D) Locate center of gravity of each area.

(E) Compute M.

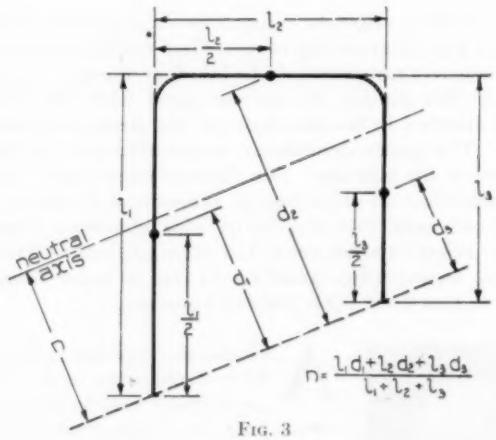


FIG. 3

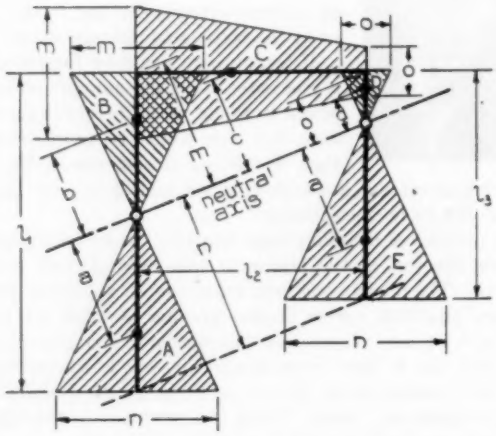


FIG. 4

$$M = Aa + Bb + Cc + Dd + Ea$$

Note: For convenience in locating center of gravity of a trapezoid, the graphic method suggested by Fig. 5 may be used.

Case III, Figs. 6 and 7.

(A) Locate neutral axis, as shown in Fig. 6, through the mid-points of lengths l_1 and l_2.

(B) Draw moment areas A, B, C, and D, as shown in Fig. 7.

(C) Compute square feet in areas A, B, C, and D.

(D) Locate center of gravity of each area.

(E) Compute M.

$$M = Aa + Ba + Ca + Da$$

To Plot the Moment Areas Correctly.

(1) Draw a line at the end of each pipe member perpendicular to the pipe.

(2) The length of this line is equal to the perpendicular dis-

tance from the pipe end to the neutral axis. The pipe member bisects this line.

(3) Complete the figure to obtain closed area.

TYPICAL SINGLE-PLANE PROBLEM (FIGS. 8, 9, AND 10)

Conditions.

12-in. pipe-wall thickness = 1.219 in.

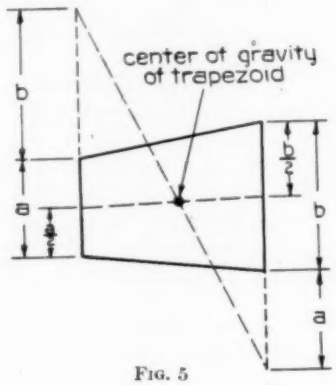


FIG. 5

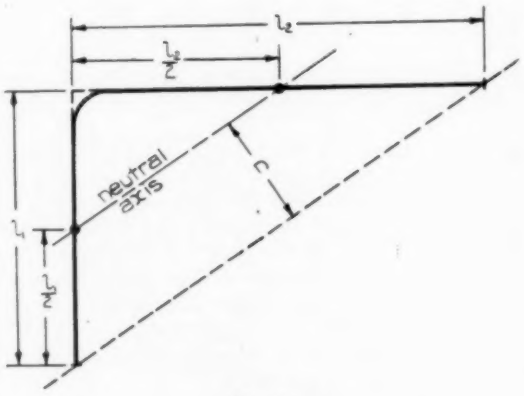


FIG. 6

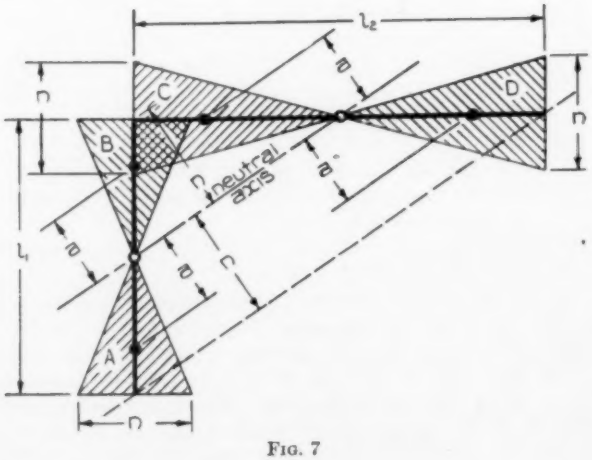


FIG. 7

I = 740.0 in.^{4}
Temperature of steam = 750 deg. fahr.
Room temperature = 70 deg. fahr.
Expansion of steel pipe at 680 deg. fahr. = 7.20 in. per 100 ft.
Fig. 8 shows the piping arrangement.

Procedure.

(A) Redraw arrangement at a convenient scale and make all bends in pipe square corners.

(B) Locate neutral axis, as shown in Fig. 9.

(C) Draw moment areas A, B, C, D, E, and F, as shown in Fig. 10.

(D) Compute square feet in areas A, B, C, D, E, and F.

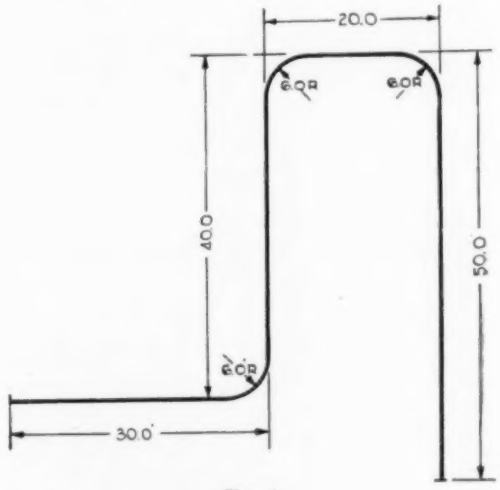
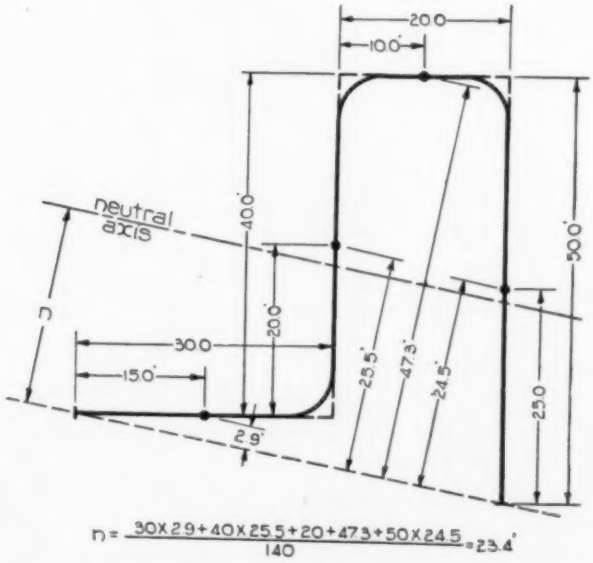


FIG. 8



$$n = \frac{30 \times 2.9 + 40 \times 25.5 + 20 + 47.3 + 50 \times 24.5}{140} = 23.4'$$

FIG. 9

$A = \frac{1}{2}(23.4 + 17.6)\,(30) = 615$
$B = \frac{1}{2}(17.6)\,(17.9) = 158$
$C = \frac{1}{2}(21.9)\,(22.1) = 242$
$D = \frac{1}{2}(21.9 + 25.6)\,(20) = 475$
$E = \frac{1}{2}(25.6)\,(26.2) = 335$
$F = \frac{1}{2}(23.4)\,(23.8) = 279$

Note: A check on the work may be made at this point. The sum of the areas on each side of the neutral axis should be equal.

(E) Compute M.

$M = Aa + Bb + Cc + Dd + Ee + Ff$

$= (615)(20.8) + (158)(11.7) + (242)(14.4) + (475)(24.0) + (335)(16.8) + (279)(15.7)$
$= 39{,}545$ cu. ft.

(F) Compute P.

Distance between anchors = 51.0 ft.

consequently, $\Delta = \dfrac{7.20 \times 51.00}{100} = 3.68$ in.

therefore, $P = \dfrac{\Delta EI}{1728M} = \dfrac{3.68 \times 29{,}000{,}000 \times 740.0}{1728 \times 39{,}545} = 1160$ lb.

and, $P_c = \dfrac{PT}{t} = \dfrac{1160 \times 140}{132.2} = 1230$ lb.

(G) Compute S.

The greatest stress in the pipe line is located at a point farthest from the neutral axis. From Fig. 10 this distance R is found to be equal to 25.6 ft.

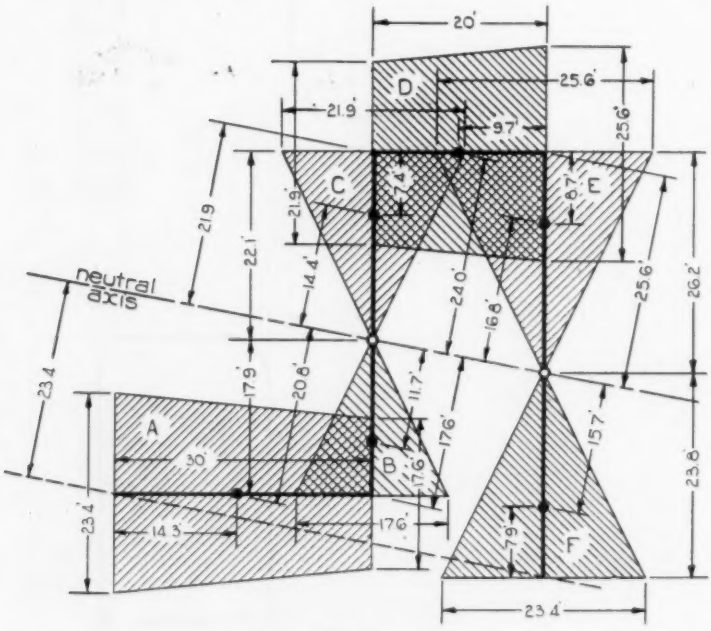


Fig. 10

Then, substituting in formula $S = \dfrac{6P_cRD}{I}$

$S = \dfrac{6 \times 1230 \times 25.6 \times 12.750}{740.0} = 3250$ lb. per sq. in.

The stress may be calculated at any point along the pipe line by scaling the perpendicular distance from this point to the neutral axis and substituting it for R in the foregoing stress formula.

Typical Two-Plane Problem (Figs. 11, 12, 13, 14, and 15)

Conditions.

6-in. pipe-wall thickness = 0.5 in.
$I = 45.4$ in.^{4}
Temperature of steam = 750 deg. fahr.
Room temperature = 70 deg. fahr.
Expansion of steel pipe at 680 deg. fahr. = 7.20 in. per 100 ft.
Fig. 11 shows the piping arrangement.

Procedure.

(A) Preliminary drawing.

(1) Lay out plan view to scale, with line X–X connecting anchors parallel to top edge of paper and all bends changed to square corners, as in Fig. 12.

(2) Draw elevation, as in Fig. 13, which shows the line X–X in its true length.

(3) Project another plan, as in Fig. 14. This view also shows line X–X in its true length.

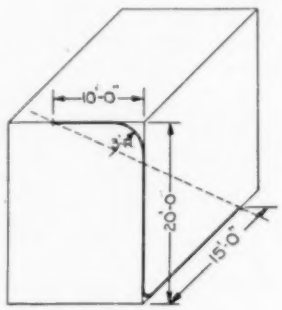


Fig. 11

(B) Locate neutral axis.

(1) Draw neutral axis through center of gravity of each view parallel to line X–X, taking care to use the true and not the projected length of each pipe member in the calculations, as follows:

$$n_1 = \frac{(10 \times 2.05) - (20 \times 2.60) - (15 \times 4.60)}{45} = -2.23 \text{ ft.}$$

$$n_2 = \frac{(10 \times 4.15) + (20 \times 8.35) + (15 \times 4.15)}{45} = 6.02 \text{ ft.}$$

(C) Draw moment areas A, B, C, D, E, F, and G, as shown in Fig. 15.

(1) To determine the moment areas it is necessary to know the perpendicular distance from the neutral axis to all parts of the piping arrangement. This distance for any point is the hypotenuse of a triangle, the legs of which are the ordinates to the neutral axis from the point in question. These ordinates may be scaled from Figs. 13 and 14.

(2) Draw the moment areas about the line Y–Y shown in Fig. 15. This line represents the total length of pipe with each member marked out to scale.

(*a*) Above line Y–Y and perpendicular to it, mark off the ordinates for the end of each member, as scaled from Fig. 13.

(*b*) Where the neutral axis crosses the middle member in Fig. 13, the ordinate with respect to line Y–Y is zero. Mark this point on line Y–Y in its relative location.

(*c*) Draw all ordinates *above* the line Y–Y irrespective of their position in Fig. 13.

(*d*) Connect the plotted points with straight lines.

(*e*) Draw the irregular line below line Y–Y in such a way that the vertical distance between the lines above and below Y–Y at any point is the bending moment at that point. This is conveniently done by laying off the required ordinate, scaled from Fig. 14, horizontally along line Y–Y, and by swinging the hypotenuse or true bending moment into position, as illustrated in Fig. 15.

(3) Divide the total moment area into a convenient number of trapezoids.

(D) Compute square feet in areas A, B, C, D, E, F, and G.

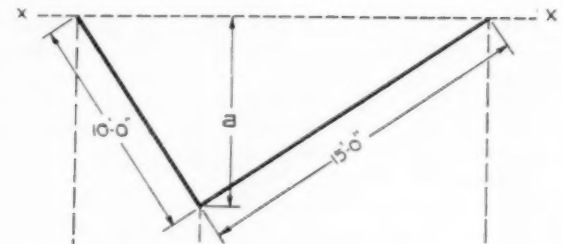


Fig. 12

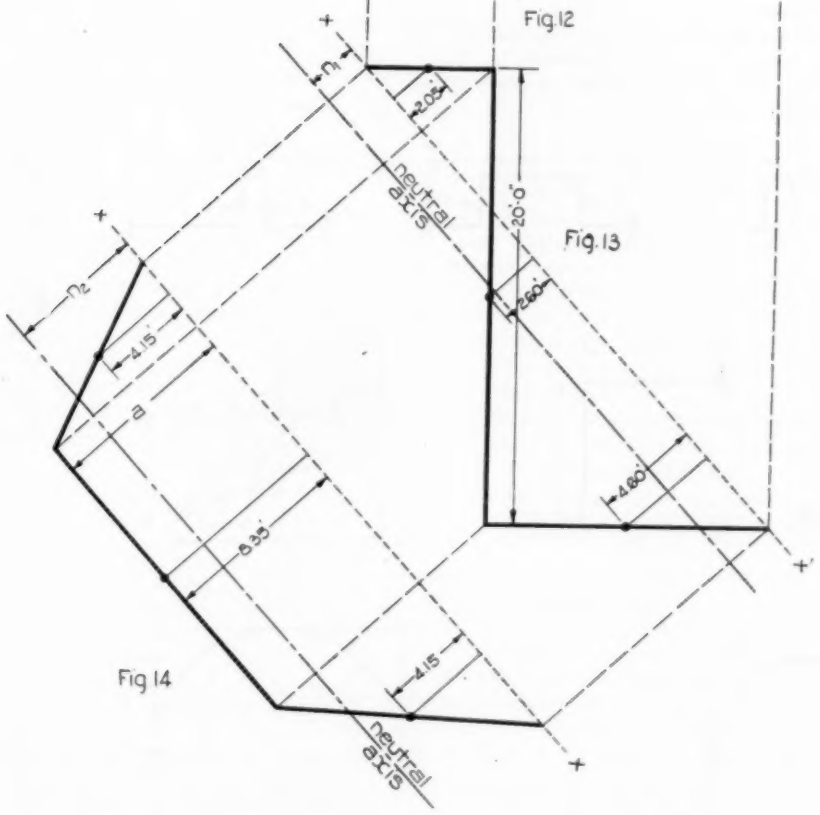


Fig. 13

Fig. 14

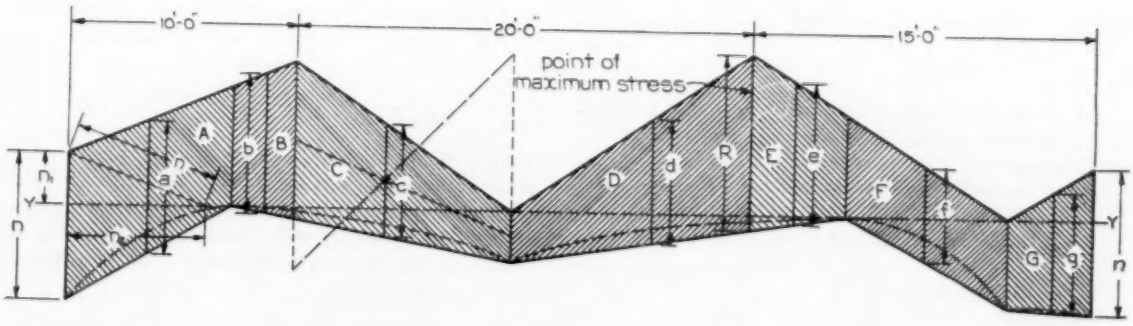


Fig. 15

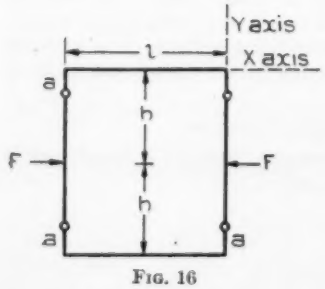


Fig. 16

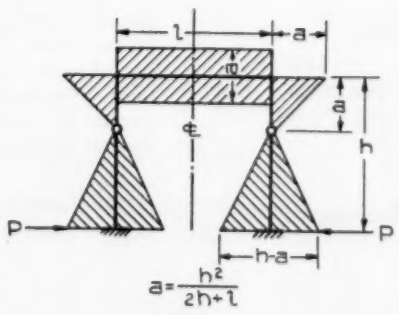


Fig. 17

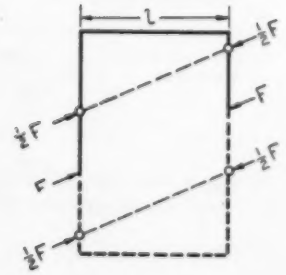


Fig. 18

$A = \frac{1}{2}(6.4 + 5.2) \times 7.2 = 41.7$
$B = \frac{1}{2}(5.2 + 6.9) \times 2.8 = 17.0$
$C = \frac{1}{2}(6.9 + 2.3) \times 9.5 = 43.7$
$D = \frac{1}{2}(2.3 + 7.7) \times 10.5 = 52.5$
$E = \frac{1}{2}(7.7 + 4.5) \times 4.2 = 25.6$
$F = \frac{1}{2}(4.5 + 4.0) \times 7.1 = 30.2$
$G = \frac{1}{2}(4.0 + 6.4) \times 3.7 = 19.3$

(E) Compute M.

(1) Determine distances a, b, c, d, e, f, and g.

(a) Locate center of gravity of each trapezoid by the graphical method previously described.

(b) Draw lines through each center of gravity and perpendicular to Y–Y.

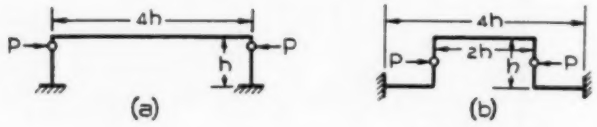


Fig. 19

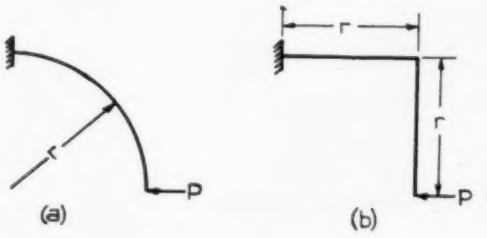


Fig. 21

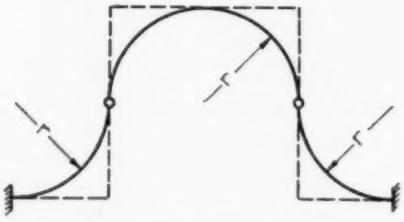


Fig. 22

(c) Scale the length of each line a, b, c, d, e, f, and g, as shown in Fig. 15.

(2) Compute M.

$$M = Aa + Bb + Cc + Dd + Ee + Ff + Gg$$
$$= (41.7)(5.8) + (17.0)(6.1) + (43.7)(5.0) + (52.5)(5.4) + (25.6)(6.2) + (30.2)(4.1) + (19.3)(5.2)$$
$$= 1237 \text{ cu. ft.}$$

(F) Compute P.

Distance between anchors = 26.9 ft.

$$\text{consequently, } \Delta = \frac{7.20 \times 26.9}{100} = 1.94 \text{ in.}$$

$$\text{therefore, } P = \frac{\Delta EI}{1728\,M} = \frac{1.94 \times 29{,}000{,}000 \times 45.4}{1728 \times 1237} = 1195 \text{ lb.}$$

$$\text{and } P_e = \frac{PT}{t} = \frac{1195 \times 45}{42.6} = 1260 \text{ lb.}$$

(G) Compute S.

The maximum stress occurs at the point of greatest bending moment. In the two-plane piping arrangement this stress may be due to a combination of torsion and bending. An explanation will be given later as to why it may be treated as though it were a simple bending stress.

The point of maximum stress is shown in Fig. 15, and the distance R is found to be equal to 7.7 ft.

Then, substituting in formula $S = \frac{6\,P_e\,RD}{I}$

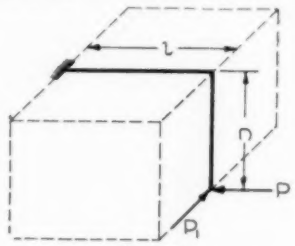
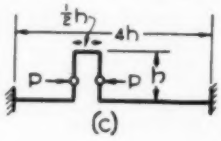
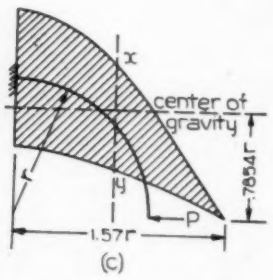


Fig. 20





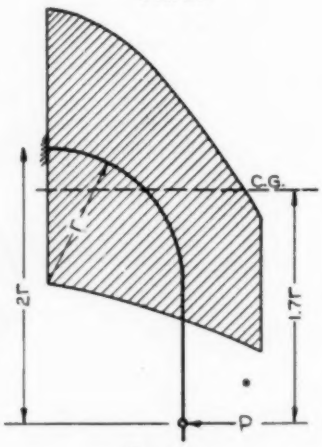
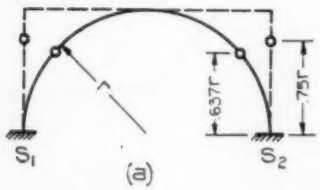


Fig. 23



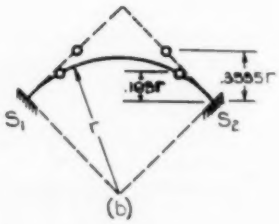


Fig. 24

$$S = \frac{6 \times 1260 \times 7.7 \times 6.625}{45.4} = 8500 \text{ lb. per sq. in.}$$

The stress may be calculated at any point along the pipe line by scaling the ordinate at this point on line Y–Y in Fig. 15, and substituting it for R in the foregoing stress formula.

Explanation of Dotted Curves in Moment Diagram

If all points on each member are plotted for the bending moment, it will be found that the lower line of the diagram, Fig. 15, is a series of curves instead of straight lines. The labor involved in plotting these curves is not considered worth while in view of the small amount of accuracy sacrificed. It is more important to avoid errors in plotting than to be too precise about the last decimal.

Theoretical Analysis

The theory upon which the graphic method is based may be demonstrated as indicated on page 7.

The first step is to eliminate all bends and to replace them with square rigid corners. Square corners are physically impossible, of course, but for the purpose of theoretical analysis they are excellent. Having computed the stresses in a square-cornered structure, a method will be indicated for correcting the results and applying them to an equivalent round-cornered structure.

Fig. 16 represents four lengths of pipe joined at angles of 90 deg. at the corners, each corner having the same rigidity as the pipe itself. At the middle of the vertical length a force F is acting, and is opposed by an equal force on the opposite side. It is evident that when flexure occurs four points of neutral bending moment are formed in the vertical members at a distance a from the corners. To determine the length a, follow the procedure used in mechanics in the analysis of continuous beams.

TABLE 1 MOMENTS OF INERTIA OF PIPE SECTIONS

Nominal size of pipe, in.	O.D.	250 Lb. Std. I.D.	250 Lb. Std. I	400 Lb. Std. I.D.	400 Lb. Std. I	600 Lb. Std. I.D.	600 Lb. Std. I	900 Lb. Std. I.D.	900 Lb. Std. I	1350 Lb. Std. I.D.	1350 Lb. Std. I
4	4.500	4.026	7.2	3.826	9.6	3.938	8.3	3.750	10.4	3.500	12.8
6	6.625	6.065	28.5	5.761	40.2	5.855	36.8	5.625	45.4	5.250	57.2
8	8.625	7.981	72.5	7.625	106.0	7.688	100.6	7.375	126.5	6.875	162.2
10	10.750	10.020	160.5	9.750	212.0	9.688	223.0	9.250	296.0	8.625	383.0
12	12.750	12.000	280.0	11.750	363.0	11.500	439.0	11.000	580.0	10.313	740.0
14	14.000	13.125	428.0	12.750	589.0	12.688	616.0	12.125	822.0	11.313	1081.0
16	16.000	15.000	733.0	14.625	969.0	14.500	1045.0	13.938	1365.0	13.000	1812.0

Take an origin at the upper right-hand corner, with Y axis vertical and X axis horizontal. Clockwise moments will be considered positive. The moment of inertia of the pipe is I and the modulus of elasticity E. The bending moment at any point in the length immediately below the origin $= M =$

$$EI\frac{d^2x}{dy^2} = F(h-y) - \frac{F}{2}(2h-y-a) \quad \ldots\ldots [1]$$

integrating

$$EI\frac{dx}{dy} = Fhy - \frac{Fy^2}{2} - Fhy + \frac{Fy^2}{4} + \frac{Fay}{2} + C_1 \quad \ldots [2]$$

and

$$EIx = \frac{Fhy^2}{2} - \frac{Fy^3}{6} - \frac{Fhy^2}{2} + \frac{Fy^3}{12} + \frac{Fay^2}{4} + C_1y + C_2 \ldots [3]$$

As x, or the deflection, equals zero when y equals O, C_2 equals O. If $y = h$, $EI\frac{dx}{dy} = O$, since a tangent to the deflection curve at F is vertical. This gives the value of C_1. Substituting it in Equation [2], the tangent to the curve at the origin

$$EI\frac{dx}{dy} = \frac{Fh^2}{4} - \frac{Fah}{2}$$

To solve for a, an expression must be obtained for the tangent to the elastic curve at the right of the horizontal member. It is evident that there will be a reaction $P = F/2$, acting at distance a from the horizontal member, and since this force P is exerted parallel to this member, the bending moment at any point of it will be Pa, or $M = EI\frac{d^2y}{dx^2} = -Pa$, the minus sign being necessary since the moment is counter-clockwise with respect to the origin.

The tangent to the curve caused by this moment is

$$EI\frac{dx}{dy} = -Pax + C_3$$

But the tangent is O when x equals $\frac{l}{2}$, and consequently $C_3 = \frac{Pal}{2}$. Therefore, when x equals O, $EI\frac{dx}{dy}$ equals $\frac{Pal}{2}$.

Since the corner is rigid, $\frac{dy}{dx}$ may be considered equal to $\frac{dx}{dy}$, and consequently $\frac{Pal}{2} = \frac{Fh^2}{4} - \frac{Fah}{2}$.

whence a equals $\frac{h^2}{2h+l}$.

Now remove the lower half of the rectangle in Fig. 16, substituting fixed supports at the point of application of the force F. The result is a familiar expansion loop with the vertical members anchored at the bottom. Expansion of the length l causes horizontal reaction P at the base and also at the points of neutral bending moment. The tangents to the deflection curves at the base of h are zero. Having the length a, the bending moment at any point in the structure can be readily computed provided the value of the force P is known.

To determine P an expression must be developed for the total deflection of the structure at the point of application of P and equated to the actual expansion in the length l. It can be assumed, of course, that P causes the bases of the vertical members to approach each other an amount equal to the expansion in l. The deflection of the structure will evidently equal the sum of the deflections of each separate member. Each a and each $(h-a)$ is a cantilever with the force P acting at the free end. The deflections of a and $(h-a)$, for instance, are expressed by the familiar formulas $\Delta = \frac{Pa^3}{3EI}$ and $\Delta = \frac{P(h-a)^3}{3EI}$. If the top member l were inflexible, the deflection of the structure would equal the four deflections of the vertical members a and $(h-a)$. But the member l is not inflexible, and when it bows up slightly, due to the constant bending moment Pa it permits additional lateral deflection of the members a. To determine the total laterial deflection of the member a at its lower end, refer to Equation [2], which is an expression for the tangent to the deflection curve at any point of h below the origin. If $y = O$ in this equation, $EI\frac{dy}{dx} = C_1$.

But it has already been proved that $EI\frac{dy}{dx}$ and $EI\frac{dx}{dy}$ at the origin equal $\frac{Pal}{2}$. Therefore C_1 equals $\frac{Pal}{2}$.

It has also been proved that C_2 in Equation [3] equals zero. If $y = a$ in Equation [3], an expression is obtained for the total deflection of a at its lower end. For convenience convert F into its equivalent $2P$. Then

$$EIx = Pha^2 - \frac{Pa^3}{3} - Pha^2 + \frac{Pa^3}{6} + \frac{Pa^3}{2} + \frac{Pa^2l}{2}$$

or

$$x = \frac{Pa^3}{3EI} + \frac{Pa^2l}{2EI}$$

The value of P is now obtained from the following equation, in which Δ is the known expression of the horizontal member:

$$\Delta = \frac{2P}{EI}\left[\frac{a^3}{3} + \frac{(h-a)^3}{3} + \frac{a^2l}{2}\right]$$

Two very interesting discoveries manifest themselves at this point. The first has to do with the three items in the foregoing

brackets. In order to visualize them relatively, the moment diagrams have been sketched as shaded areas in Fig. 17.

The first two items are concerned with the triangular areas $a^2/2$ and $(h - a)^2/2$. The third item is concerned with the rectangle $al/2$ (only one-half of the structure being considered). The rectangular area is quite important, as it represents the sum of the bending moments along the length l. The relation of these areas to the deflection of the members on which they are superimposed is very simple. For example, the area $a^2/2$ only has to be multiplied by $2/3a$ to make it equal to the first item in the brackets, or $a^3/3$. The area $al/2$ becomes $a^2l/2$ by simply multiplying it by a. But $2/3a$ is the center of gravity of the upper triangle with respect to the line of force which passes through the point of neutral bending moment in h, while a is the distance of the center of gravity of the rectangle from the same point. From this is deduced the following important law: If the moment diagram be superimposed on a member, with the center of gravity of the diagram coinciding with the axis of the member, the deflection of the member will be proportional to the area of the diagram times the shortest distance from its center of gravity to the line of the force which causes the deflection. To avoid overlapping of areas in Fig. 17, the upper triangular diagrams have been drawn on either side of h instead of axially, which is permissible as long as it does not disturb the horizontal center of gravity of the areas.

This device of using moment diagrams to compute deflection is quite familiar to structural engineers. Fortunately it is peculiarly adapted to the problem of pipe expansion.

Now comes the second discovery. Equating the sum of the areas above the line of neutral bending moment to the areas below results in the following:

$$a^2 + al = (h - a)^2$$

Whence $a = \dfrac{h^2}{2h + l}$, the formula which has already been deduced. The sum of the moment diagram areas above the neutral line is therefore equal to the sum of the areas below the line. In other words, the neutral axis or "secondary force" passes through the center of gravity of the structure.

This necessarily holds true for any structure between two rigidly anchored supports, provided it is in one plane, consists of intersecting straight lines, and has no force acting on it other than the internal ones due to expansion. It is also true, for any similar structure, that the neutral axis is parallel to the line passing through the anchor points.

That these lines of force are parallel may be demonstrated as follows: Fig. 18 is a closed rectangle with rigid corners. The force F acts diagonally through the center of the rectangle. Since the structure is in equilibrium, the horizontal reaction along the members l must be equal and opposite. This, however, is an apparent anomaly, since the force F does not act at the center of either vertical member. The only way in which the force F can cause equal reaction at the corners is through the development of neutral axes on each side of the force at equal distances from its point of application. The vertical member then becomes a cantilever bridge, the suspended span supporting the load in the middle and transmitting half of it to each unequal cantilever arm. Since the suspended spans are of equal length on either side of the rectangle, it follows that the force $F/2$ which passes through the points of suspension must be parallel to F.

If the dotted half of the rectangle is removed, there remains a three-sided expansion bend with unequal legs, a common enough case in practice. Each leg is so anchored tha' tangents to the deflection curve at the base are vertical. Conditions are identical to those which prevail in the closed rectangle.

The graphic method of solving for M and the location of neutral axis for this shape has already been illustrated in Fig. 4. A similar method is applicable to any straight-sided plane shape whatever.

A further study of the method reveals the fact that expansion bends of the same overall dimensions will establish quite dissimilar values for P according to the arrangement of the horizontal and vertical members.

In Fig. 19 three expansion bends are shown, each having a length between anchors of $4h$ and a height of h. In order to illustrate clearly the effect of arrangement on the internal stresses, relative value for P and the maximum bending stresses have been worked out for each shape and assembled in the following table:

	Fig. 19(*a*)	Fig. 19(*b*)	Fig. 19(*c*)
Relative values of P	100	43	63
Relative bending stress, top	20	26	56
Relative bending stress, bottom	100	26	19

The discrepancy between these figures is quite surprising. A comparison indicates that, in general, the arrangement shown in Fig. 19 (*b*) is the best, especially as regards the value of the direct thrust P. It also indicates that the place to locate a valve or a flanged joint is certainly not at the top of arrangement Fig. 19 (*c*) or the bottom of arrangement Fig. 19 (*a*), provided they can be located elsewhere. The figures in this table must not be accepted too literally, as they are exact only for arrangements whose length is four times the height. They may be accepted as applying to round-cornered shapes as well as to the square-cornered ones shown in the illustration.

A word of caution should be inserted regarding the shape (*c*) in Fig. 19 in case the expansion loop is placed near the end of a very long horizontal line. Theoretically it does not matter how long the line is or at what part of it the expansion loop is located. This line is in effect a horizontal strut with an eccentric load which causes it to deflect slightly upward toward the expansion loop. If this upward deflection is prevented by external means, as is very apt to be the case in a long line (the weight of the pipe itself will resist it somewhat), a new arrangement of forces results. The expansion will be the same as before, but the length must now be taken from the point of application of this external force. The neutral axes will assume a higher position on the vertical members, and the structure will be less flexible. There are formulas in use which give the deflection of a similar expansion loop when anchored on one side and free or unattached on the other, the assumption being, apparently, that the free end is really part of a long length of pipe. There is actually no such thing as a "free end" in a steam line. Such an assumption makes it impossible to arrive at the correct stresses in the structure.

Pipe Structure in Two Planes

Before approaching the subject of bends, a brief discussion may be inserted of the method as applied to the two-plane structure. In general, it may be said that a pipe line which departs from a single plane does so with the primary purpose of getting from one point to another, rather than with the purpose of insuring flexibility. Such structures, as a rule, are more flexible than is really necessary, due to the introduction of torsion. Occasionally, however, this is not the case, and there are times when space requirements do not permit of sufficient flexibility in the plane structure.

In such cases the method already described in detail may be of assistance. In principle it is identical to the method used for single-plane structure. The development of projections in plan and elevation is merely a device to assist in locating the line passing through the center of gravity of the structure and in measuring the perpendicular distances from this line to the various members. It will be found that the "secondary force" usually does not intersect the structure at any point. In these

cases torsion is present in every member and is inextricably associated with direct bending. It is unnecessary to separate them, since the deflection in either case varies as the length of member acted upon times the sum of the squares of the bending moments.

This relationship is illustrated in Fig. 20, which represents a pipe of length l, fixed at one end and acted upon at the other by a force P or P_1 at the end of an inflexible lever arm of length n. Internal stresses in the pipe cause a deflection at the point of application of the force. The deflection due to P equals Pln^2/EI, while that due to P_1, considering the effect of torsion only, equals P_1ln^2/EJ.

Since the polar moment of inertia J for pipe sections equals $2I$ and since Es is about 11,200,000 lb. per sq. in. as against 29,000,000 lb. per sq. in. for E, the actual deflection due to P_1 is about 30 per cent greater than that due to P. This relative differential persists for all variations of l and n. It is considered permissible to disregard it in practical problems for the following reasons: First, since the torsional moments in a structure are usually less than half of the direct bending moments, the inaccuracy is thereby reduced in proportion. Second, disregarding the differential adds a slight increment to the factor of safety. This is offset to a certain extent by plotting the sides of the moment diagrams with straight lines. These lines are found to be slightly curved in some cases if plotted for all points along the length of the member. The straight line thus adds a small amount of area to the diagrams, while another small amount is subtracted by ignoring the constant differential between torsional and bending deflection.

In general, the graphic method, as described, is slightly less accurate for two-plane than for single-plane problems. Nevertheless, it makes possible the solution of problems which by any other known method are practically insoluble.

A word of explanation may be added regarding the determination of maximum pipe-wall stress in the two-plane problem. In the problem illustrated, the procedure is exactly the same as with the the single plane. It is obviously an unnecessary refinement to distinguish betwen torsional and bending stress. The polar moment of inertia equals $2I$; consequently, if the maximum moment happens to be all torsion, the resulting shear is $S/2$. If it happens to be all direct bending, the resulting stress is S. The allowable stresses for shear are roughly one-half those for bending. Where there is combined bending and torsion, the method of measuring the moment arm from the neutral axis to the point in question and computing the stress as though it were a simple bending stress is again on the side of conservatism. The net resultant stress is theoretically somewhat lower. To compute it accurately would be outside the field of practical engineering.

Pipe Bends

It is not the intention to develop any new ideas or formulas on the subject of expansion bends, but merely to apply the graphic methods outlined in the foregoing paragraphs with a view toward establishing a relationship between the bend and the theoretical square corner. Square corners are much easier to work with. A correction factor for the bends appears both feasible and desirable and will tend to clarify the whole subject.

Only one general assumption is made, which is that the moment of inertia of a section taken in the bend is the same as that for the straight pipe.

That many writers object to this assumption has no particular bearing on the following analysis: Fig. 21 shows a 90-deg. bend with radius r, fixed at one end and with a force P acting at the other. Alongside it is drawn an equivalent square corner with legs r, and the same force P acting at the free end. The horizontal deflection of the latter shape has been found to be:

$$\Delta = \frac{P}{EI}\left(r^3 + \frac{r^3}{3}\right) = \frac{4Pr^3}{3EI}$$

The deflection of the bend can also be obtained in the same manner by the exercise of a little patience. Divide the bend into any number of equal parts. Using intervals of 5 deg., for instance, will give 18 sections of the curve, each of which, for all practical purposes, is a straight line. From the center of each section drop an ordinate perpendicular to the line of force P. The length of these ordinates may be obtained from a table of sines. The sum of all the ordinates divided by 18 will be the average height of the moment diagram. The area of the diagram is its average height times the length of the member, or 1.57 r. Multiplying the two quantities results in r^2 as the area of the diagram. Now the deflection of the bend is proportional to the sum of the squares of the ordinate times the length of a single section of the curve. In the case of a straight member the deflection is more readily computed by the use of the moment diagram and its center of gravity. But the moment diagram of the curved member is irregular in shape, as illustrated in Fig. 21 (*c*). To determine its center of gravity would be only an added labor. The result by either method is the same, and is expressed as follows in terms of deflection:

$$\Delta = \frac{\pi}{4}\frac{Pr^3}{EI}$$

The deflection of the bend is therefore 59 per cent of that of the square corner when both are acted upon by the same force. Now in an expansion loop similar to that shown in Fig. 22, this ratio evidently applies to the structure as a whole. The neutral axes coincide with the change in direction of the curves, and the effect is that of four 90-deg. bends with a force acting at the free end of each. The neutral axes of the curved structure also coincide with those of the square-cornered one. In this one case, then, the deflection can be first determined for the square-cornered structure and corrected with the factor 0.59.

But the neutral axes do not always occur at the end of the curve, and there is usually a quantity of straight pipe to be accounted for. In Fig. 23 a case is shown where the neutral axis is below the end of the curve an amount equal to the radius of the curve. The distance r must be added to each ordinate in obtaining the area of the moment diagram. There is also a moment diagram to be considered for the straight length below the curve. Neglecting this (since it is the same for both bent and square corners), the deflection, for the bend only, is found to be proportional to the quantity 4.37 r^3. Similarly the deflection of a square corner with legs of r and r in the same relative position to the neutral axis proves to be 6.33 r^3. The ratio between the two equals 4.37/6.33 = 69 per cent.

Lowering the neutral axis has consequently affected somewhat the deflection ratio between the square and ·ound corners. Now move the neutral axis down to a point $5r$ below the end of the curve and repeat the calculation. The results for the two cases and the ratio between them is 50/66.3 = 75 per cent.

The accuracy of an analysis based on a square corner in place of a bend is therefore 59 per cent, 69 per cent, and 75 per cent for the three cases. But in the second and third cases straight pipe has been introduced whose deflection is obtained with a com parative accuracy of 100 per cent. Giving credit to the straight pipe according to its proportion of the total length of the shape, the foregoing percentages of accuracy become 59 per cent, 79 per cent, and 93 per cent, respectively.

These percentages can evidently be further increased. The simplest factor for this purpose is the length of the curve itself in

relation to the length of the square corner in which it is inscribed, or 0.7854. Experimenting with this factor, it is found that the percentages of accuracy increase to 75 per cent, 92 per cent, and 99 per cent, respectively.

Two more cases will bear investigation before the use of this factor can be thoroughly justified. They are those in which the neutral axis is located on the curve itself, as illustrated in Fig. 24 (*a*) and (*b*). The location of the neutral axis is shown for the square and round shape in each case. The deflections calculated in a straight line between supports S_1 and S_2 for a given force P are as follows:

$$(a) \quad \Delta = \frac{0.295\,Pr^3}{EI} \text{ for round shape}$$

$$= \frac{0.4167\,Pr^3}{EI} \text{ for square shape}$$

$$(b) \quad \Delta = \frac{0.0244\,Pr^3}{EI} \text{ for round shape}$$

$$= \frac{0.0833\,Pr^3}{EI} \text{ for square shape}$$

Multiplying the second equation in (*a*) by the ratio of round to straight lengths reduces the constant to 0.328. The accuracy is therefore 0.295/0.328, or 90 per cent.

By the same operation the constant in the second equation of (*b*) becomes 0.0655. The accuracy in this case is therefore 0.0244/0.0655, or only 37 per cent. This is due to the projection of the square corner above the top of the curve, with correspondingly longer lever arms about the neutral axes. But the curve in (*b*) is not a legitimate expansion bend. In practice the radius r would be reduced as much as possible with straight lengths inserted between the bend and the supports. If the length of this curve is not greater than one-third the total length, the accuracy by the method suggested will be over 90 per cent.

Summarizing, therefore, the square corner may safely be used in the analysis of practically all pipe-expansion problems, correcting the results for Δ with the factor:

$$\frac{\text{total length of actual structure}}{\text{total length of square-cornered structure}}$$

Results will be from 90 to 100 per cent accurate in theory for all structures having the usual amount of straight pipe between the bends. Greater accuracy than this is hardly possible in any event, in view of the various uncertainties always present in problems of this character.

If the designer wishes to make allowance for the flattening of the pipe cross-section in bending, he may of course use any correction factor he chooses or none at all, bearing in mind that in doing so he lowers the factor of safety of the structure. The point to be emphasized is the small influence of the bends in the average design. In a certain large and probably typical station there is hardly a single instance where the length of straight pipe does not greatly exceed the length of bent pipe between any two fixed points.

Allowable Stresses

In general it is recommended that for very high pressures and temperatures the maximum stress in the pipe wall due to expansion be limited to 5000 lb. per sq. in. at any point. In some cases it may be desirable to limit the direct thrust at the anchors to 1000 lb., and even less for small pipe. These conditions, though conservative, are usually not difficult nor expensive to meet.

Stresses are reduced in proportion to the amount of cold-springing employed. It is recommended, however, that no regard be paid to cold-springing in designing the structure, except in special cases where close check of the work is possible, to insure the pipe being installed as planned.

FSP-52-26

Heat Reaction in the Boiler Furnace

By W. J. WOHLENBERG,[1] NEW HAVEN, CONN.

This paper attempts to point out only a few of the angles from which the modern art of boiler design may be viewed. Each important case must be specially investigated, for there are no hard-and-fast rules that may be applied, and many empirical data must still be relied upon in determining proportions that will result in satisfactory operation. The future will require that students become acquainted with certain broad underlying principles reached through such methods of analysis as are hinted at in this paper.

THE boiler furnace may be defined as consisting of a cavity provided with attachments for introducing fuel and discharging ash and in which the progress of the combustion of the fuel should continue practically to completion. This definition is concerned with the first considerations in the art of furnace designing. If this were the only set of conditions to be met, we might say that there has been but small progress in the development of such furnaces since the time of Watt. Even in those days these conditions could be met if the fuel was the right one and the rate of combustion was maintained within certain limits.

Modern developments have been forced in the direction they have taken in order to enable operation under conditions of far greater rates of steam output per unit of floor space which is occupied by the steam-generating units. The end in view has been, of course, that reduction in the total cost per unit weight of steam which results from the attendant reduced requirements in labor, overhead, and maintenance when concentrating the total capacity for generating steam into fewer units. It should be noted, however, that an increase in the rate of energy release per steam-generating unit does not necessarily require a greater release of energy per cubic foot of furnace volume per hour. For large furnaces it may mean the employment of arrangements and methods which make it possible to utilize to greater advantage the increase in furnace volume which is provided by increasing the height of the furnace cavity. As an example, some of the largest modern pulverized-coal-fired units produce a large amount of steam per square foot of floor space, even though the rate of energy release per cubic foot of furnace volume may be low. In such furnaces a long flame path is provided along the vertical axis, and hence the fuel and air streams are carried through the furnace at a high velocity. The time element during which any particle is en route through the furnace cavity may actually be greater than it would be in a shorter furnace and one in which the velocity of gas travel is lower.

Difficulties Encountered in Operation

The difficulties which are encountered in the operation of either large or small furnaces, but in which the total amount of energy released per unit of time is large at full load, may usually be traced back to the existence, either in localized spots or throughout the furnace, of temperatures which are too low or too high. When the temperatures are too low, the combustion reaction may not progress as it should; or if a furnace is designed to operate slagging (that is, at temperatures sufficiently high to cause the ash particles to melt and run down the side walls), then wall temperatures below the slagging point may cause building up on the side walls of layers of freezing slag, assuming possibly a honeycomb type of structure. If the process continues in this way for some time, such slag formations may later have to be run down in a molten form by raising the furnace temperature, or they may have to be broken down. In the latter parts of the wall are usually broken down with the frozen slag.

If the temperature is too high at any part of the furnace, a wall may fail due to the inability, from some cause or other, to conduct energy as heat through it at a rate sufficient to keep the inner-wall surface temperatures below the softening point of the material composing this surface. In pulverized-coal-fired furnaces slag penetration of the refractory surfaces will occur under these conditions. This is often followed by a chemical action between certain ash and refractory constituents, and the new compounds then often have lower melting points than either of the combining constituents. The inner refractory surfaces now rapidly disintegrate, and the process will continue to destruction unless special provisions, such as indirect water cooling of the refractory, are resorted to in order to arrest the action while the wall is still thick enough to remain intact.

Some attempts have been made to solve the furnace-refractory problem by doing away with all of the refractory materials on the inner faces of the furnace cavity and replacing them with bare iron water-cooled surfaces in the form of tubes through which water is circulated. However, such arrangements have in many instances led to serious losses from incomplete combustion because at low rates of driving, particularly with pulverized-coal firing, maintenance of the proper progress of the reaction during the heating-up and ignition periods has been retarded. Highly preheated air and turbulent firing improve the conditions in this respect at the higher rates of driving the furnace. At the lower rates, however, both the air temperature and turbulence fall off, with the result just stated. On the other hand, at exceedingly high rates of driving with uneven flame distribution, which is bound to occur at times, the localized heating of the unprotected tube elements has led to tube failures.

Thus it appears as though in most cases the design of the structure of the furnace cavity must be for a compromise between the conditions which provide for good combustion at all rates of driving and those which provide for lowest cost of maintenance of this structure under the conditions of operation that will be met. This compromise may resolve itself into a structure in which there are some bare iron water-cooled walls and others of the uncooled refractory type. Or, in another case, a part or all of the inner faces of such walls may be of refractory backed by water-cooling means. Only for large furnaces, for small furnaces, or when the extra expense of extremely fine grinding of the coal may be incurred, should consideration be given to the structure composed almost entirely of bare iron water-cooled surfaces if pulverized coal is the fuel. Even with these special provisions it is in such cases desirable to carry in the air for combustion in a highly preheated state.

All of this leads up to the point of this discussion. The heat reaction in the boiler furnace, as distinguished from combustion or chemical reaction, plays an important part in the process, at least in so far as its practical operation is concerned. Under the term "heat reaction" may be included the effects on the several parts of the structure of the temperature heads, and of their differentials, in discharging energy between the vulnerable parts

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Lecture delivered at the S.P.E.E. Summer School for Teachers of Mechanical Engineering, held at Purdue University, Lafayette, Ind., June 27 to July 18, 1929.

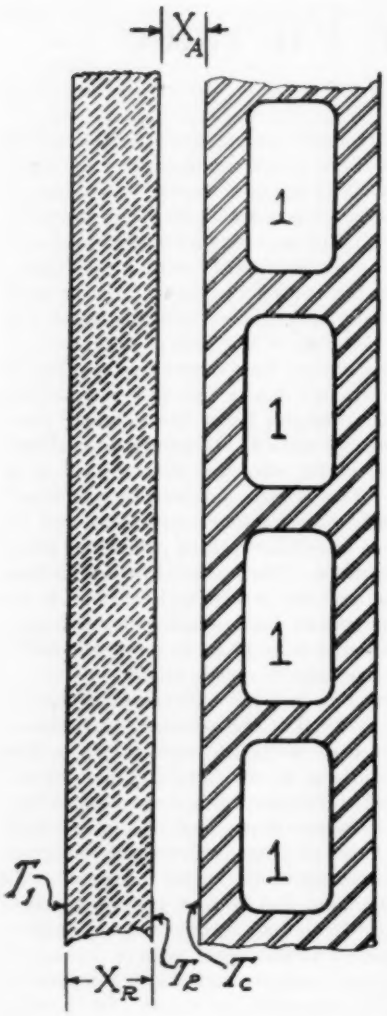


FIG. 1
(1, water circulating through channels in iron or steel water back.)

of the furnace cavity. A solution of this phase of the process thus involves a study of the energy distribution as heat within this cavity. A complete solution of the furnace reaction could, of course, be accomplished only by evaluating step by step the influence of the combustion reaction on the heat reaction and vice versa. Such a method of treating the problem is not within the limits of the possibilities of any analysis based on our rather meager knowledge of the details of the process. In dealing with this problem we are thus forced to separate the heat reaction from the combustion reaction. We assume a certain progress of the combustion reaction to determine the energy which has been released, and then make use of the principle of conservation of energy together with the available data and laws according to which energy as heat is distributed throughout cavity by radiation, conduction, convection, etc.

HEAT TRANSFER AND EQUILIBRIUM AT THE FURNACE WALLS

In discussing the heat reaction one should begin by a short analysis of the process which occurs at the walls of the furnace cavity. As an example, consider a wall such as is represented by Fig. 1.

Let K_R = conductivity of refractory
K_A = conductivity of air gap
T_1 = mean absolute temperature of furnace or hot face of refractory
T_2 = mean absolute temperature of cold face of refractory
T_c = mean absolute temperature of the iron face of structure forming channels for circulation. T_c will be practically equal to water temperature.
X_R = mean thickness of refractory in direction of heat flow by conduction
X_A = mean width of air gap.

The transfer of energy from the furnace face of the refractory through to the water occurs in several stages. First, by conduction through the refractory, and then by a combination of radiation, conduction, and convection from the cold face of the refractory across the air gap to the metal surface bounding the water passage. The quantity transferred by convection is, for obvious reasons, exceedingly small, and hence may be neglected. The equation of heat transfer thus assumes the form:

Heat transferred by conduction through the refractory must be equal to the total transferred across the air gap, or, for a square foot of wall front,

Conduction through refractory		Air-gap conduction		Air-gap radiation
$\frac{K_R (T_1 - T_2)}{X_R}$	=	$\frac{K_A}{X_A} (T_1 - T_c)$	+	$\alpha_o (T_2^4 - T_c^4)$. [1]

in which α_o is the net coefficient of radiation between the two surfaces bounding the air gap.

The particular structure shown in Fig. 1 was chosen for an illustration so that it should be at once obvious what the mean dimensions involved in the heat transfer are. For another type of structure these might have to be evaluated separately before substitution in Equation [1], or perhaps the process as a whole might have to be submitted to analyis by means of the differential calculus. This, however, is beside the point for this discussion.

Equation [1] may be applied to evaluate for given conditions as to T_1 and T_c the heat flow through any commonly used furnace-wall structure. In Fig. 2 it is shown how equilibrium is established for a given hot-surface temperature T_1. It is to be

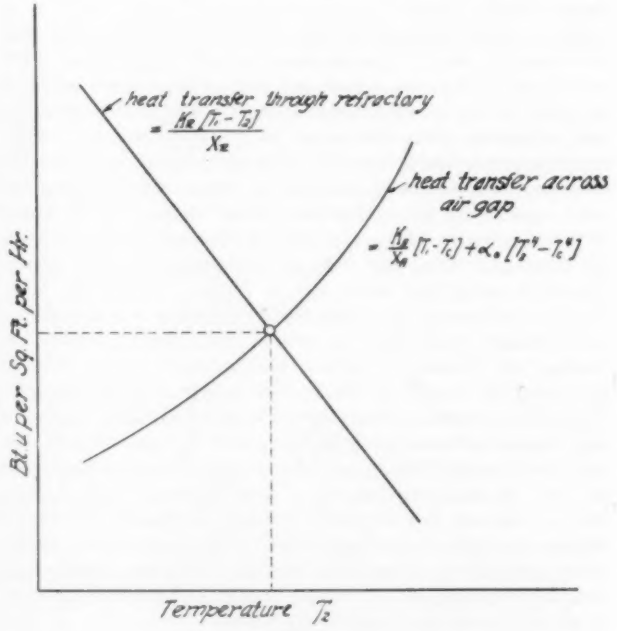


FIG. 2
(a, heat transfer through refractory $= K_R[T_1 - T_2/X_R]$; b, heat transfer across air gap $= K_A/X_A [T_1 - T_c] + \alpha_o[T_2^4 - T_c^4]$.)

noted that the energy conducted through the refractory falls off with an increase in the temperature T_2, whereas that transferred across the air gap increases. The point of intersection of the two curves shows the value of the temperature T_2 and rate of heat transfer at equilibrium.

Obviously now, the thinner the refractory the greater its conductivity, and the smaller the air gap the greater will be the rate of heat transfer through the wall to the water. At one limit the refractory has zero thickness, the rate of heat transfer is for practical purposes infinite, and the wall temperature T_2 remains constant at the value T_c of the water in circulation. At the other extreme the refractory is perfectly insulated, so that zero heat

flows through the wall by conduction and the temperature T_1 of the hot surface assumes an equilibrium value such that the energy received by radiation and convection from the flame mass is equal to the energy discharged by radiation to other parts of the furnace cavity from this surface.

The wall shown in Fig. 1 thus lies between the two limiting conditions described, from the point of view of its ability to receive radiant energy for a given surface temperature. The equilibrium magnitude of the temperature T_1 for this wall is the same time reducing the surface temperature, because now more energy as heat is conducted through the wall. This results in an increase in the total energy discharged to the wall from the flame, and in a comparatively small reduction in the mean flame temperature. The wall, under such conditions, will thus thin down until such an equilibrium state is reached. Let us suppose, for example, that this equilibrium thickness of refractory material is 2 in. A refractory of these dimensions may be maintained in the water-cooled structures, but for solid

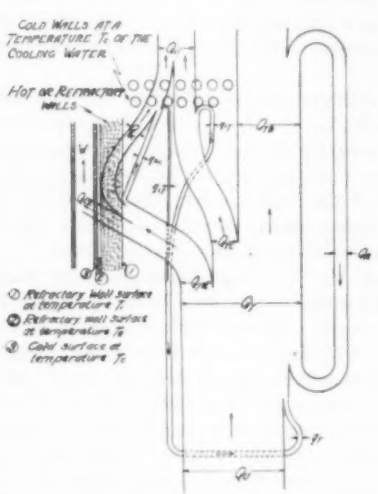


FIG. 3 ENERGY-FLOW DIAGRAM, FURNACE CAVITY

(1, refractory wall surface at temperature T_1; 2, refractory wall surface at temperature T_2; 3, cold surface at tempeature T_c.)

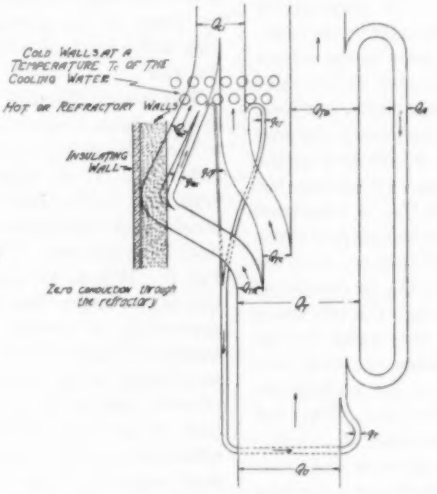


FIG. 4 ENERGY-FLOW DIAGRAM, FURNACE CAVITY

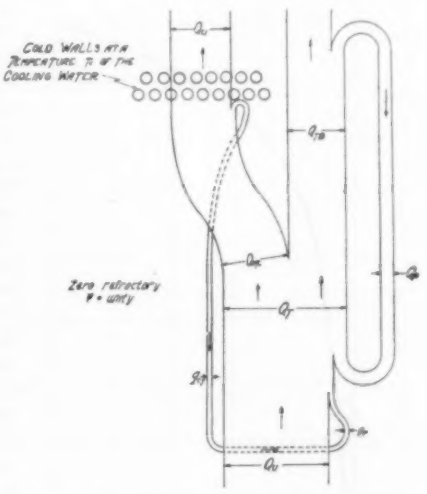


FIG. 5 ENERGY-FLOW DIAGRAM, FURNACE CAVITY

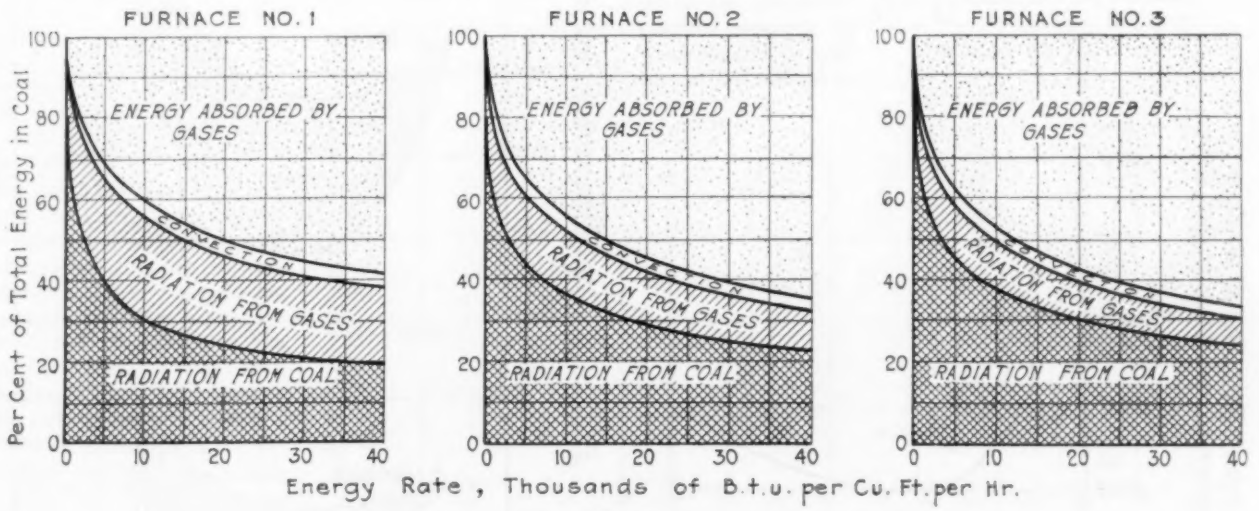


FIG. 6

that at which the energy received by radiation and convection at the wall surface from the flame mass is just equal to the rate at which energy is discharged by radiation from this surface to other parts of the furnace cavity plus the rate at which energy is conducted through the wall. The latter factor is large for this type of wall compared to its value for solid refractory wall, but less than would be the case for bare iron water-cooled walls.

If the equilibrium value of the wall temperature is so high that slag penetration occurs, the wall will obviously disintegrate at the surface and run down, thereby reducing its thickness and at refractory walls destruction is complete long before any such thinning down has occurred. Thus water cooling of refractory walls provides the conditions under which not only a thinner wall may be maintained but it also provides for the conditions for maintaining, as compared to solid refractory walls, a higher rate of energy release in the furnace for a given limiting wall temperature, which is of course an end in view. In addition to this it provides for a hot surface of such walls under all conditions. These are conducive to good combustion, which is another end in view.

Energy Distribution in the Furnace Cavity

Figs. 3, 4, and 5 are conventional diagrams representing the energy distribution throughout the furnace cavity for three different cases. In Fig. 3 refractory walls are represented at R as backed by some water-cooling means W. The bare iron cold walls are represented by circles to represent tubes. The total energy input Q_T is composed of that released by the fuel (Q_U), plus that recirculated by the preheated air (Q_A), plus that recirculated by reflection (q_T). The quantities of energy Q_{TR} and Q_{TC} are respectively radiated from the burning mass to the refractory and relatively colder iron surfaces. Of the former, the part q_{RC} is reflected at the refractory surface, the part Q_{RC} is reradiated from this surface to the cold surface, and the part Q_{c_2}, is conducted through the refractory to the water in the passage or tubes provided for cooling the refractory. The net absorption ($Q_{c_1} + Q_{c_2}$) by the water and steam in the boiler is less than the total discharge to it by the amounts q_{CT} of the reflection at the exposed cold surface C. The energy carried out by the gases escaping from the furnace cavity is represented by the width Q_{TB} of the stream passing straight through the furnace.

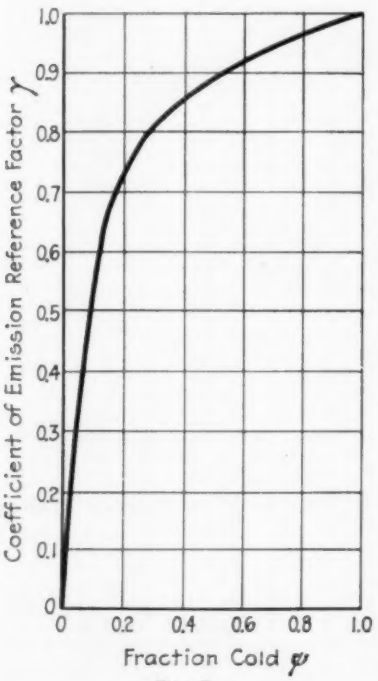


Fig. 7

It now appears as though the energy Q_{c_2} conducted through the refractory walls in effect enables an increase of the energy release rate by an amount just equal to this conduction factor. This, however, is not the case, because any reduction of the quantity Q_{c_2} will, for the same conditions otherwise, influence the width of the three streams Q_{TR}, Q_{TC}, and Q_{TB}, as shown in Fig. 4. The stream Q_{TB} will increase in width and the stream Q_{TR} will decrease, because the flame-temperature increase is now the result of the decreased radiation discharge to the walls by virtue of the increased wall surface temperature, which in turn is the result of a decrease of the energy conducted through the wall.

The stream Q_{TC} will increase somewhat, due to the increase in mean flame temperature, but it will not increase in proportion to the decrease in Q_{TR} caused by the reduction in the conduction factor Q_{c_2}. If all walls of the furnace are of the bare-cooled type, the conductivity of the wall material (iron or steel) is hundreds of times greater than for any refractory structure which may be employed. The resulting energy diagram is shown in Fig. 5. It is to be noted that the total discharge to the walls Q_{TC} is now larger than for either of the other cases, but there is apparently no simple relation between the conduction factor and the amount of energy Q_{TB} that leaves with the gases. Thus, any change in the conduction factor Q_{c_2} disturbs the equilibrium of every part of the heat reaction, or, in general, a change of any part of the heat reaction disturbs the equilibrium of every other part of this reaction. For the present, all that may be said about the influence of conduction on permissible rate of

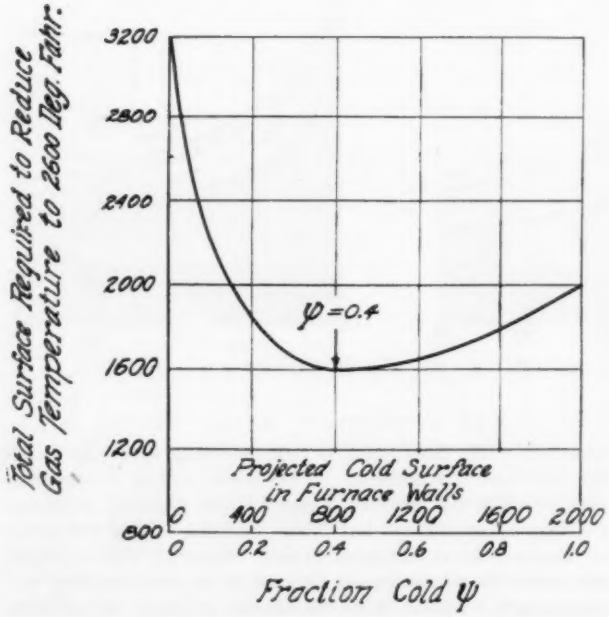


Fig. 8-A Influence of Fraction Cold on Total Surface Requirement

(Furnace No. 2, 8000 cu. ft.; 80,000 B.t.u. per cu. ft. per hr.; Illinois coal; excess air, 40 per cent; stoker fired.)

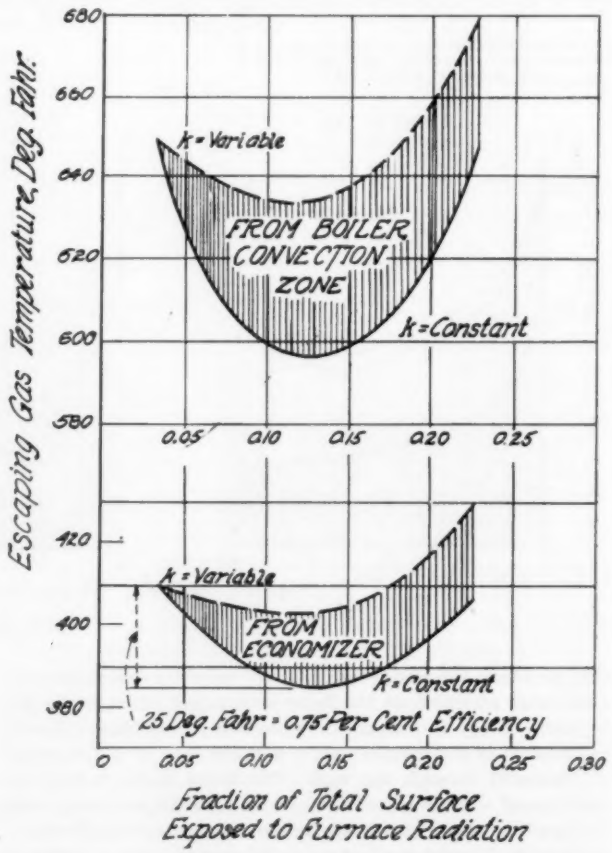


Fig. 8-B Influence of Surface Exposed to Furnace Radiation on Escaping-Gas Temperatures

(k = coefficient of heat transfer.)

energy release is that an increase of the former will result in an increase of the latter, but the two changes are not proportional, and an evaluation of their relative magnitudes involves an evaluation of the resulting equilibrium variation for every part of the process.

The general influence of the wall structure itself on equilibrium has now been discussed to some extent. Let us next consider the influence of furnace-radiation symmetry on the heat reaction. That is, what is the influence on the heat reaction of changes in the proportional parts of the envelope of the cavity which present refractory and exposed bare iron water-cooled surfaces?

Equilibrium and Radiation Symmetry of the Furnace Cavity

To reduce this picture to its simplest terms, consider the energy diagram of Fig. 4, in which all of the refractory walls are shown as perfectly insulated. The energy Q_{ca} conducted through the walls R is now reduced to zero. A solution for the equilibrium conditions of this process involves the following things: for the determination of the energy released Q_U, the fuel weight and its calorific value; for a determination of the escaping energy Q_{TB}, a curve showing the relation between thermal capacity of the resulting gaseous mixture and the temperature of this mixture; and for the energy Q_A carried in by the preheated air, a similar curve. The radiation discharges Q_{TB} and Q_{TC} result from the sum total of radiation from the solid incandescent carbon surfaces and those from the radiating gaseous constituents like CO_2 and H_2O. The former require for their evaluation the solid angles subtended by the receiving surfaces from the radiating surfaces, and of course the extent of the radiating surfaces themselves. The latter depend on what are known as the "thickness concentration functions" of the gaseous constituents, and the radiation is evaluated per unit area of gas front facing the receiving walls. For the solid carbon surfaces the Stefan-Böltzmann law may be applied, but not so for the radiating gaseous constituents. A mathematical expression for these introduces the thickness concentration functions as exponentials. The reflection from the surfaces may, of course, be treated in the usual manner.

In Fig. 6[1] is shown the computed energy-discharge fraction as it appears for certain cases in furnace cavities having zero refractory surface in the envelope. All walls have been considered as bare exposed iron surfaces at the temperature of the cooling water, and therefore the discharge to the walls is the maximum possible for cavities of the size indicated and for the fuel and combustion conditions shown.

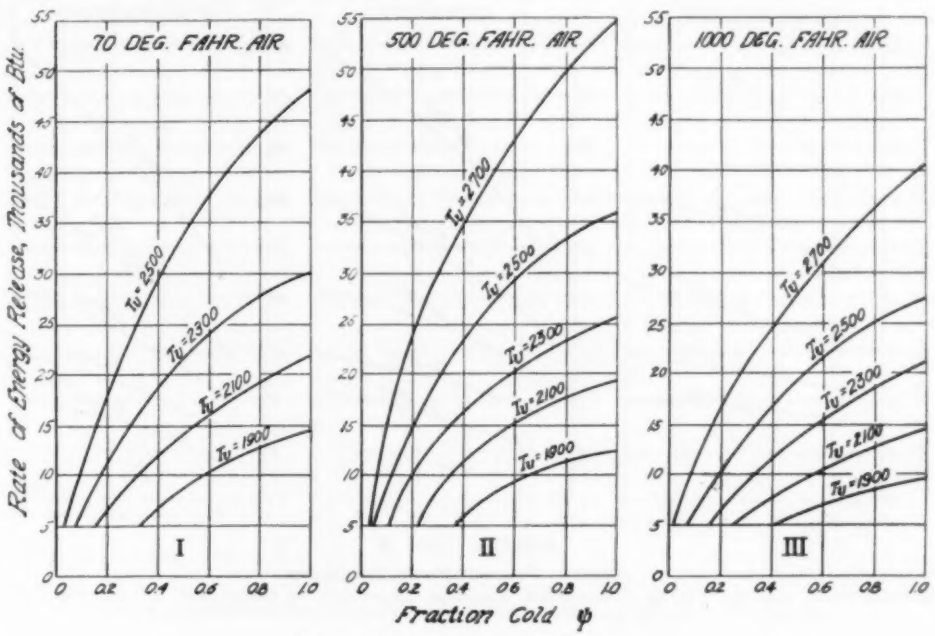


Fig. 9 Furnace-Flame Isotherms
(Furnace No. 2; coal, No. 1 or No. 2, pulverized; excess air, 20 per cent.)

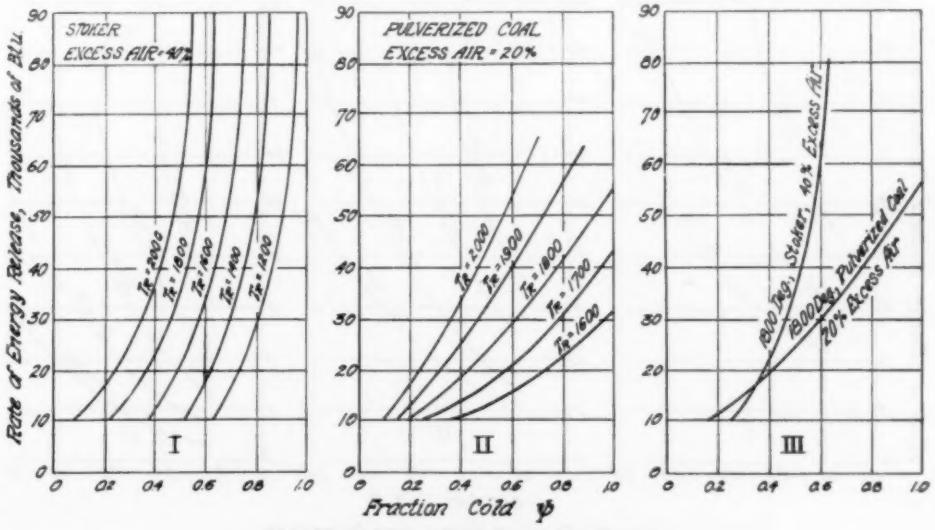


Fig. 10 Furnace-Refractory Isotherms
(Furnace No. 2; coal, No. 1 or No. 2; air temperature, 70 deg. fahr.)

The first solution for the energy distribution is, as seen, the result of a very involved procedure, and it might seem as though no simple way of viewing the sum total of the energy process in terms of furnace proportions were possible. Fortunately this is not the case. The result of a large amount of painstaking analysis shows that, for all practical purposes, the relativity of things, so far as the influence of variation of radiation symmetry on the net energy absorption is concerned, may be satisfactorily

[1] From Wohlenberg and Lindseth.

expressed in terms of a rather simple factor, called the "fraction cold."

The fraction cold is simply the fractional part of the furnace envelope which is occupied by bare iron water-cooled surfaces, the other portion of the envelope being assumed in such a case

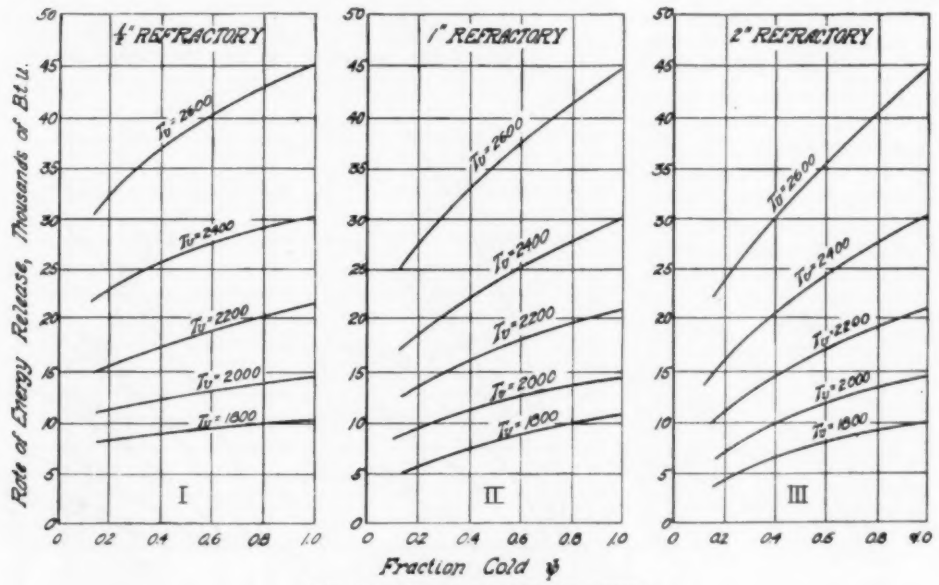


FIG. 11 FURNACE-FLAME ISOTHERMS

(Furnace No. 2; coal, No. 1 or No. 2, pulverized; excess air, 20 per cent; air temperature, 500 deg. fahr.; conductivity, $K = 10$; for any refractory conductivity multiply thickness shown by $K/10$.)

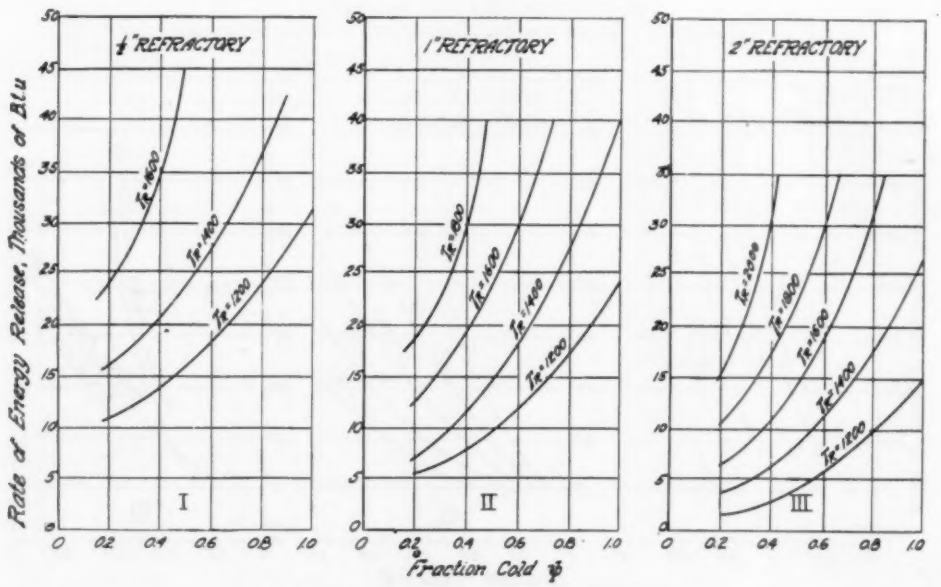


FIG. 12 FURNACE-REFRACTORY ISOTHERMS

(Furnace No. 2; coal, No. 1 or No. 2, pulverized; excess air, 20 per cent; air temperature, 500 deg. fahr.; conductivity, $K = 10$; for any refractory conductivity multiply thickness shown by $K/10$.)

to be occupied by solid refractory walls. In other cases, as will be shown later, such a factor may be used also if the part of the furnace envelope which presents refractory surfaces is backed by water or air cooling. The remaining part of this discussion will be occupied with an attempt to show to what extent a variation in the radiation symmetry of a furnace cavity influences the energy distribution, furnace performance, and steam-generator performance.

THE FRACTION COLD AND ITS INFLUENCE

The questions in the mind of the boiler-furnace designer with respect to fraction cold are these: How much does an increase in the extent of cold surface in a furnace envelope influence the temperature of the gases escaping from the furnace? And how much does it affect the rate at which energy as heat is absorbed by these cold walls? Also, how much is the final temperature of the gases escaping from the steam generator influenced by these conditions? Finally, how much does an increase in fraction cold relieve the conditions at the hotter refractory surfaces, or how much does an increase in the fraction cold permit of increasing the energy-release rate for the same wall maintenance costs? These questions, as stated, are not concerned with the extent to which the progress of combustion may be interfered with, which condition will, of course, have to be investigated separately for each case.

To indicate what the answer may be to the first two questions the curve of Fig. 7 is shown. The ordinate γ is the relation of the energy absorbed by the walls for any fraction cold ψ to the absorption at the walls when the fraction cold ψ is unity. Inspection of this curve shows that beyond a fraction cold of two-tenths the increase in absorption by increase in fraction cold is nowhere near in proportion to the increase in fraction cold. This means that as more and more surface is added, less and less energy is absorbed by each square foot of surface. Therefore, the mean temperature of the gases is influenced far less by an increase in fraction cold, from, say, 0.3 to 1.0, than it would be by an increase from 0 to 0.3.

To answer the second question the curves of Figs. 8-A and 8-B are shown. Fig. 8-A shows how the total surface requirements of the steam generator for a given escaping-gas temperature are influenced by an increase in fraction cold. Inspection shows that for the conditions illustrated there is no advantage from this point of view in going beyond a fraction cold of about four-tenths. The curves of Fig. 8-B show in another way how Münzinger has attacked this problem. The abscissa here represents the fractional part of the total boiler surface which is exposed to furnace radiation, and it is to be noted that the lowest final temperature results when approximately 12 per cent is exposed to radiation.

Both Figs. 7 and 8 indicate that a large fraction cold may never be the means of greatly improving the overall absorption efficiency of a steam generator. Its purpose, then, is, as before stated, either to reduce the cost of maintaining a wall at a given rate of energy release or permit a higher rate of energy release for a given maintenance.

Mean temperatures of wall and flame are of course, as before mentioned, good indicators of what the destructive effect of the combustion process will be for a given fuel. With this in view, the isotherms of Figs. 9 and 10 are shown. They are flame and refractory isotherms plotted between axes of energy-release rate and fraction cold. It is to be noted that an increase of fraction cold from two to four-tenths permits of doubling the energy-release rate for a given flame temperature. For refractory surface temperatures of 1800 to 2000 deg., roughly the same proportions hold as shown in Fig. 10.

The curves of Fig. 11 and 12 represent the same conditions for a furnace in which the refractory portions of the wall are backed by water cooling. The air gap between the refractory and water-cooled iron back is considered as zero. It is to be noted that for such cases fraction-cold variation has less and less influence as the refractory thickness is reduced. This should, of course, be the case, because as the refractory is approaching zero thickness the furnace envelope is approaching one in which the fraction cold is unity.

Conclusion

In concluding this discussion of heat reaction it may be said that it has been attempted to point out only several of the angles from which the modern art of boiler-furnace design may be viewed. No hard-and-fast rules may be applied. Each important case must be the subject of special investigation, and as yet many empirical data must still be relied on in arriving at proportions which it is hoped will prove satisfactory in operation.

It is very important, however, that those of the coming generation of engineers who may enter this field should begin to think of this process in other than purely empirical terms. Methods of analysis such as are hinted at in this discussion will acquaint the student with the broad underlying principles which are involved, and in addition may help to give them some sense of the relativity of things. This is of greater permanent value than, for instance, a table of values which shows what has been accomplished in certain furnace cavities.

FSP-52-27

Some Economic Considerations of Water-Wall Installations

By OLLISON CRAIG,[1] WORCESTER, MASS.

THE walls of boiler furnaces, on an increasing number of power plants, are being covered, in whole or in part, by water-cooled surfaces. These surfaces may be bare tubes or they may be tubes covered by metal blocks or by refractory material. In some cases metal is arranged to present a continuous surface by interposition of metal strips between bare tubes. The purpose of installing such surface depends on individual conditions. If the furnace is fired by powdered coal, it may be for protection of the walls, prevention of slag, increased capacity, or increased boiler efficiency. If the furnace is stoker fired, it may be for protection of the walls, prevention of slag formation on side walls or tubes, prevention of clinker in the fuel bed, protection of metal parts of the stoker, increased capacity, or increased efficiency. Usually, however, such an installation is made, not on account of one of these factors alone, but because of some combination of them, since no one is independent of all the others.

In most boiler-furnace installations a limiting factor is temperature, and this factor influences the others referred to. First the effect of temperature in furnaces fired by powdered coal will be discussed. In this case refractory walls are affected by high temperature in a number of ways. Softening of the wall material may occur. Undue expansion may be experienced, demanding special construction. High furnaces may cause undue compression loads that can result in failure of the softened brick. Slag formation may erode the walls by the action of fluid ash flowing over the softened surface. Certain combinations of ash and fireclay result in chemical reactions at high temperature, which cause deterioration of the brick. In any of these cases furnace-wall maintenance is affected.

Slag formation in the bottom of a furnace is expensive to remove, unless the ash is at all times kept so molten as to permit removal by tapping. Tapping in general is not practical except in cases of special design, and even then is not desirable except with low-fusion ash. Under certain conditions molten-ash particles may slag over the openings between tubes in the first boiler pass.

Refractory walls of stoker-fired furnaces are affected in the same way as walls in powdered-coal furnaces, except that in addition erosion may occur through the rubbing of the fuel bed adjacent to the wall. This form of erosion is increased with increased furnace temperature and increased softening of the refractory-wall surfaces. Slagging of the boiler tubes by molten-ash particles also occurs in stoker operation. This may be even more pronounced than in the case of powdered-coal firing, as the ash particles rising from the fuel bed will be larger and may not be chilled below the softening point before coming in contact with the tubes. High furnace temperatures tend to the formation of clinker in the fuel bed, reducing fuel-burning area, causing holes which admit excess air, and interfering with the mechanical action of the stoker parts. Stoker metal parts are affected by high furnace temperature, particularly if liquid slag comes in contact with the metal. Molten slag may act as a flux, and serious burning of metal parts occurs.

Attempts to secure higher capacity or higher efficiency in either a powdered-coal fired furnace or in a stoker-fired furnace usually result in higher furnace temperatures. Higher capacity demands a higher fuel-burning rate and a higher rate of heat release, resulting in higher temperature in a given furnace. Increase in efficiency may be obtained by less excess air being used or by returning to the furnace heat recovered from the flue gases. Either results in higher furnace temperature.

Furnace temperature then limits both capacity and efficiency. The effect of water tubes in furnace walls is to absorb heat by radiation, leaving less heat in the gases, and to produce a lower temperature in the furnace than would be the case with refractory walls entirely exposed, assuming all other conditions, particularly as to rate of fuel-burning ratio of coal to air, and furnace volume, the same in each case.

Since the action of slag is the cause of most of the furnace difficulties, and since the other difficulties become quite minimized if furnace temperatures are such that slag does not occur, it may be said that the temperature at which softening of ash occurs is the desired limit of furnace temperature, and that the combination of fuel-burning rate, rate of air supply, and water-tube surface exposed should be such that this temperature should not be reached.

At the beginning a number of reasons for the use of water tubes in a boiler furnace were stated. In reality there is only one reason, and that is decreased cost of steam production Decreased cost of steam production may be realized through greater thermal efficiency, increased capacity, or use of cheaper coal, but certain things that can produce greater efficiency, such as reduced excess air, will result in higher furnace temperature. Greater capacity produced by greater fuel-burning rate can also result in higher furnace temperatures. Cheaper coal may mean lower grade coal, having ash which fuses at a lower temperature. Spending money for water tubes in furnace walls may then result in reduced steam cost by preventing such effects of high furnace temperature as have been discussed and by permitting operation either at increased efficiency or at higher capacity or with cheaper coal. The decision in a particular case as to whether water tubes should be installed in the walls of a furnace, and the area of water tube surface to be installed, depends on the load it is desired to develop, the economical limit of excess air, and the fusion temperature of the ash in the coal. The increase in cost of such tube area must be set up against reduction in cost of boiler and furnace, against the reduction in cost of coal consumed, and against reduced price of coal per ton.

The presence of water tubes in a furnace reduces furnace temperature, but does not directly affect overall boiler efficiency to any considerable extent through this reduction in temperature alone. Water tubes in furnace walls absorb heat radiated from the flame and furnace gases, which heat is transmitted to the water contained in the tubes. This heat finally appears in the steam leaving the boiler. The amount of heat that the flame and gases give up to the furnace walls is greater when water tubes are present than when the walls are entirely refractory. As a result, gases will then enter the first bank of boiler tubes at a lower temperature. This is on the assumption that other conditions are

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[1] Consulting Engineer, Riley Stoker Corporation. Mem. A.S.M.E.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the same; that is, that the same weight of coal is burned per hour and that the coal is burned with the same amount of air, or that the CO_2 in the gases is the same. The gases enter the boiler tubes at a lower temperature as a result of this heat absorption in the furnace, but leave the boiler tubes at only a slightly lower temperature. Under these assumed conditions the only gain in thermal efficiency that can be realized is in lower exit-gas temperatures. If, then, the difference in exit-gas temperatures is negligible, the gain in thermal efficiency is negligible. If, instead of placing tubes in the furnace walls, an amount of boiler-tube surface was added to the last bank or last pass, approximately equivalent in area to the area exposed to radiation, the final gas temperature would be about the same as if this area were placed in the furnace.

This is illustrated in Fig. 1 by means of a curve of gas-temperature drop. A curve is plotted with per cent of boiler surface

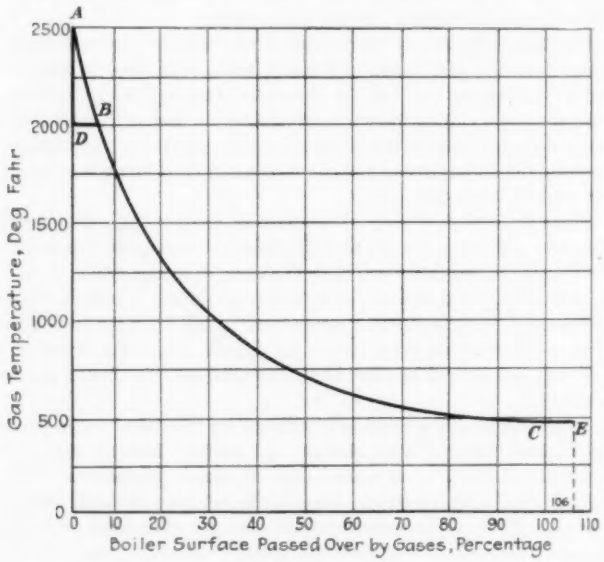


FIG. 1

passed over by gases as abscissas and temperature of gases as ordinates. The curve indicates how the gases drop in temperature as they pass through the boiler setting from the furnace to the outlet. For a boiler unit without water tubes on the furnace walls, the gases enter the first tubes at an assumed temperature of 2500 deg. fahr., and this point is indicated as the point *A* on the curve. On arrival at the boiler outlet 100 per cent of the boiler surface has been passed over by the gases and the temperature of the gases has dropped to an assumed temperature of 460 deg. fahr. This point is represented as *C* on the curve. If water-tube surface is placed on the furnace walls and coal is then burned in the furnace at the same rate as previously, and burned with the same weight of air, the gases will enter the tubes at a lower temperature than in the first case.

Assume that sufficient tube area has been placed in the furnace to reduce the temperature of the gases entering the boiler tubes to 2000 deg. fahr. With respect to the coordinates used on the curve in Fig. 1, this condition of the gas would be represented by the point *D*. A curve could then be drawn from *D* representing the change in gas temperature as boiler surface was passed over. Assume a horizontal line drawn from *D* to intersect the curve *A-C* at *B*. Gases would then be entering the first row of boiler tubes at the same temperature as if they had passed over (*D-B*) percentage of the tube area in the case where no tubes were in the furnace. A curve drawn from *D* would then be the same as the curve *B-C* but transposed a horizontal distance *D-B*, and instead of drawing the curve from *D* the curve *B-C* would be extended to *E*, so that the horizontal distance from *C* to *E* would be equal to *D-B* and the temperature at *E* would represent the final temperature of the gases leaving the boiler. The reduction in gas temperature is equivalent to the addition of tube area in the last pass represented by the horizontal distance from *C* to *E* or from *D* to *B*.

On this assumed curve the increase in area is indicated as about 6 per cent of the boiler-tube area. With a given boiler and with a given operation in which the assumed curve may be realized, the addition of surface in the furnace exposed to radiation, equivalent to 6 per cent of the boiler-tube area, will result in reduction of gas temperature about as indicated by the curves (from 2500 deg. fahr. to 2000 deg. fahr.), but the final gas temperature has only been reduced about 13 deg. fahr. With 14 per cent CO_2 in the gases, this 13 deg. fahr. represents about one-third of 1 per cent of the heat in the coal.

We may say then that water-cooling a furnace does not result in any appreciable increase in thermal efficiency if the same amount of coal is burned with the same amount of air. The same idea may be expressed by saying that the absorption of heat by radiation from the flame and gases in the furnace does not of itself result in a material increase in boiler efficiency. Thermal gain through the use of water tubes in the furnace is obtained through burning coal with a smaller quantity of air or with a higher percentage of CO_2 than would be permissible with refractory walls.

That the amount of boiler tubes exposed is a factor in furnace temperature is well illustrated by certain types of horizontal water-tube boilers which have had tile covering the lower row of tubes in such a way as to present the effect of a flat refractory arch over the fire. If in such cases the tile covering was removed and flat tile laid above the tubes, allowing the tubes to be exposed to radiation, the furnace temperature was reduced. In making calculations in connection with water-cooled furnace walls the exposed boiler tubes should then be included.

When coal is burned, a gas is produced which in weight is equivalent to the weight of air used for combustion plus that portion of the coal combined. The weight of gas produced per pound of coal burned is indicated by the percentage of CO_2 in the gas. The more air per pound of coal the less the percentage of the CO_2; and the less weight of air per pound of coal the greater the percentage of CO_2. Burning a pound of a particular coal results in a particular amount of heat. This heat would all be absorbed by the products of combustion if no heat was lost by radiation. The difference between the heat generated and the heat radiated is that which is absorbed by the products of combustion, and the amount of this absorbed heat determines the temperature of the gas entering the boiler-tube space. If burning a pound of coal results in the release of 14,000 B.t.u. and the production of 15 lb. of gas, and if 5000 B.t.u. are radiated to surrounding surfaces, the absorption of the remaining 9000 B.t.u. will result in a temperature rise of about 2230 deg. fahr., or a total temperature of 2300 deg. fahr. of the gases entering the boiler tubes if air is supplied at 70 deg. fahr. If the ash in the coal softens or fuses below this temperature, trouble will be encountered due to slag in the furnace. The above condition represents about 14 per cent CO_2. If slag results, it will then be desirable either to arrange for more heat to be absorbed by radiation or to operate at a lower percentage of CO_2. It will be seen then that the addition of water tubes in the furnace in such a case will result in high thermal efficiency through permitting higher CO_2 to be maintained. This is practically the only way in which water-cooled walls can result in higher boiler efficiency.

As stated before, the decision as to the use of water-cooled walls and as to the amount to be used must be made for each indi-

vidual case. It is necessary to consider the size of the boiler, the amount of steam to be made, the variation of load, the kind of coal to be burned, and the efficiency expected or economically possible.

For example, assume a boiler of 500 rated horsepower burning coal, the ash of which fuses at 2300 deg. fahr. Assume the projected area of boiler tubes exposed to the furnace to be 200 sq. ft. Assume that it is desired to burn the coal with 14 per cent CO_2. If the boiler operated at 200-lb. pressure, the exit-gas temperature would be about 610 deg. fahr. and the thermal efficiency would be about 79 per cent, developing 1250 boiler horsepower. Under those conditions the temperature in the furnace would be about 2550 deg. fahr. and would result in slagged ash. It would then be necessary to reduce the CO_2 to about 12 per cent or to place water tubes in the furnace walls having an exposed area of about 150 sq. ft. projected area. Prices for such walls vary up to $35 per square foot. At $20 the cost of the water cooling would be $3000. If, instead of spending the money for water cooling, the furnace was operated at 12 per cent, the resulting efficiency would be about 75 per cent. If the coal burned contained 14,000 B.t.u. per lb. and cost $5 per ton delivered, the difference in cost of coal for one year would be about $4000, providing the boiler operated at this load for 24 hr. a day and 300 days a year. Obviously in such a case it would be decided to install the water tubes in the furnace wall.

Assume, however, that 1250 horsepower was a peak which occurred only occasionally and that the total time during which this load was carried amounted to only the equivalent of 30 days continuous operation. If water cooling should be installed, the cost would still be $3000, while the saving in coal cost would only be $400. If off-peak loads did not exceed about 800 to 1000 horsepower, then water tubes would not be installed and the furnace would be operated with 12 per cent CO_2 during peak loads.

In this example if it is assumed that fixed charges on water-wall costs be taken at 15 per cent, the annual charge against the water-wall installation would be $450 per year. If, then, the peak-load period exceeded a total of about 35 days per year, there would be a net gain by spending money for the water walls.

The point that it is desired to bring out in this example is that the furnace temperature may be modified either by using water walls for absorption of heat by radiation or by supplying a greater quantity of air with a given quantity of coal and having a greater weight of gas for absorbing a given quantity of heat. The example is used to illustrate the fact that a charge may exist in either case and that in actual design these charges should be computed and compared.

Frequently a plant is faced with the necessity for greater steam production, with its steam-producing units already operating at the limit of their capacity. The limit may be because of coal-burning capacity, draft, or furnace temperature. Coal-burning capacity may be limited due to furnace temperature, as in the case of clinker formation in the fuel bed with stoker firing. To obtain more capacity by installing additional steam units may cost more than the units alone. In some cases a new boiler house needs to be constructed and in some cases no space is available for any additional units. If furnace temperature is the condition limiting steam production, as it most frequently is, additional capacity may be obtained at the least expense by the addition of water tubes in the furnace. This procedure will undoubtedly become more and more common in the future.

There have been cases in which water tubes have been placed in a furnace for capacity effect and a less number of tubes used in the convection portion of the boiler. It is true that a square foot of tube area exposed to radiation in a furnace will absorb far more heat than the average square foot of surface in the convection portion of the boiler. To take an example again, assume a 500 rated horsepower boiler to deliver 1500 developed horsepower. Assume half the convection-tube surface removed and replaced with tubes in the furnace walls, leaving 2500 sq. ft. of convection surface. In such a case there would not be 2500 sq. ft. put in the furnace, but more probably 600 or 700 sq. ft. in projected area, or about 1000 sq. ft. in total tube area. Gases would enter the tubes at a lower temperature as a result of the water-cooled walls, but would leave the convection surface at a higher temperature as a result of having only half the normal convection surface. The temperature of the gases leaving the convection surface would be about 750 deg., assuming other conditions as in the previous example. This would represent about $3^1/_2$ per cent loss in thermal efficiency. In such a case it is common to express the load in terms of percentage of normal rating of the convection surface; that is, in this example the boiler would be said to operate at 600 per cent of rating.

By means of this arrangement greater capacity is obtained at less cost, as the saving in cost of convection surface will be greater than the cost of the water tubes in the furnace. However, there will be a charge against this saving due to increased coal cost resulting from higher exit-gas temperature. Generally it will pay to add heat-recovery means in the form of an economizer or air preheater. There is a decided tendency toward using less convection surface, absorbing more heat by radiation in the furnace, and recovering more heat from the gases leaving the convection surface.

To sum up, each case must be considered on its own merits, and a few fundamentals should be kept in mind. Water-cooled furnace walls do not appreciably increase overall efficiency through heat absorption and reduction in temperature of furnace gases, but rather through making it possible to carry higher percentages of CO_2 than would otherwise be possible. The expense of water-cooled furnace walls is justified only through savings in coal and cost of equipment. Load factor should be studied in a given case and calculations made to determine whether it is more economical to burn more coal for a short period or to pay for water-cooled surface in the furnace. Increased capacity may be obtained through the use of water-cooled furnace walls, but at the expense of coal cost, unless heat-recovery means are also provided. It may be found possible to get more capacity in a given space and at a lower cost together with greater overall efficiency by using water-cooled furnace walls with less convection surface and the addition of heat-recovery equipment.

Discussion

J. A. Drogue.[2] The author very elaborately goes into economics of water-wall furnaces. The writer would relate some of the operating characteristics and experiences with the water-wall furnace at our plant.

For fifteen months we operated this boiler for six days a week, twenty-four hours per day, at about 300 per cent rating. Our mill goes into production promptly at 6 a.m. on Monday and continues through the week until the following Sunday morning. During this time we had no other boiler to go to in case of emergency, and it was absolutely necessary to carry the mill load at all times on this one boiler. A few words of explanation will counteract any thought you may have on the poor management of the plant in getting caught with only one boiler. Our new layout was such that we had to remove four old boilers (which incidentally would not carry the mill load in event of losing the boiler in question) to make room for new units.

On two occasions in the past several months, due to some work we did or did not do over the previous week end, we were brought

[2] Ball Brothers Company, Noblesville, Ind.

face to face with the necessity of a shutdown or an attempt to run through the week with possible disastrous results. The mill operators were informed of the impending shutdown, and all process work was tapered off to a minimum load. At this point the fire was quenched to a stage when the surface of the fire bed ceased to be incandescent and at the same time the induced-draft fan continued to run. Without the use of a stream of water or a man-cooler two men working in shifts entered the furnace and made necessary adjustments, and the mill was back into production in one hour and thirty minutes. During the past winter we experienced six weeks of close to zero weather, and a lengthy shutdown over the week-end no doubt would have proved costly, since buildings with their various machines and water lines must be kept from freezing, not to mention the necessity of maintaining temperatures in some of our buildings at a point that will insure correct process on starting the mill on Monday morning.

During this cold spell and at various times during the winter it was necessary for us to do our periodical cleaning in and around our one boiler, and this was accomplished in the twenty-four hours shutdown period from Sunday morning until Monday at the same time. A conservative estimate of our money lost per hour in the event of a shutdown is placed at $900.

I feel sure you will agree that it would have been impossible for us to have entered a refractory furnace of similar size, even though it were air cooled, without requiring a much longer time to cool to the point where a man could enter and work.

In the paper mention is made of an item of $4000 as the added cost of water walls on a given job, and my impression is that the author endeavors to show in dollars and cents the justification of a water wall. After the experience of fifteen months as outlined, the thought has come as to what might have been our loss in dollars if we did not have water walls.

A. G. Christie.[3] The subject of water-cooling of furnace walls is one in which the writer has had an intensive interest for many years. The author has clearly stated the ultimate effect of water cooling on furnace performance, and he has shown that if the same amount of coal is burned with the same amount of air, water cooling of the furnace does not result in any appreciable increase in thermal efficiency. While this conclusion is a logical deduction, there are some features of water cooling in a furnace which the author does not discuss.

For instance, in large powdered-coal furnaces it is held that the cooling effect of the water walls does not extend more than a few feet into the furnace and that the temperature of the central mass of the turbulent combustible flame is unaffected by the presence of water-cooled walls. If this is granted, then the effect of the walls is to produce a relatively cold envelope of gases on the outside of the furnace and an intensely hot core. If the gases passed into the boiler tubes in uniform layers, the tube surfaces in the sides and ends of the boilers would receive relatively cool gases while the central tube sections would receive as hot gases as if no water cooling were present. In such a case the tubes would slag over in this portion as fully as if no water-cooled furnace walls had been used.

These conditions are to some extent offset by the turbulence in the furnace which tends to whirl the colder side layers of gases into the interior mass and thus tends to equalize furnace temperature. In spite of this mixing, experience has shown that severe slagging can occur in the central part of the lower boiler tubes even with water-cooled walls when powdered coal is used. Boiler tubes in the form of slag screens have been found desirable in many cases to prevent such slag formation under these conditions.

[3] Professor of Mechanical Engineering, Johns Hopkins University, Baltimore, Md. Mem. A.S.M.E.

The author indicates that any gains in boiler thermal efficiency with water-cooled walls are due to the use of a smaller quantity of air together with a higher percentage of CO_2 than would be permissible with refractory walls. These considerations of decreased air and higher CO_2 must therefore have been assumed in the paper of Messrs. Bennett and Oberholtzer in estimating the increased efficiencies resulting from the use of water-cooled walls.

Melvin D. Engle.[4] The author has very clearly pointed out that the use of water walls in themselves does not result in an appreciable increase in boiler and furnace efficiency. There has been a tendency to make extravagant claims for water-cooled furnace walls, and it is helpful to the industry to have some one point out wherein the use of water walls is actually justified.

There are two factors that the author has failed to mention that are important in considering the advisability of installing water-cooled furnace walls. These are, first, the fact that water walls should increase steam-generating unit availability and, second, that water walls permit more rapid increase in steaming rate.

Greater availability will reduce the unit cost of boiler plants; and the ability to pick up and drop load rapidly without shortening the life of furnace walls is becoming more and more important in the operation of large electric systems. Steam stations that act as standby for systems supplied by hydroelectric plants at considerable distance from the load will find this feature of particular importance.

The author states that the use of water-cooled furnace walls prevents the formation of clinker in the fuel bed on stokers. Operating experience has not fully borne out this contention for stokers of great length and width. This is probably due to the fact that the gases of combustion in a furnace act as a shield to prevent the water-cooled walls from cooling zones at any great distance from the walls.

The author also stresses the point that the use of water-cooled furnace walls materially lowers the temperature of the gases entering the boiler. Is it not a fact that the water wall only lowers the temperature of the gases entering the boiler a few feet from the water-cooled wall? Is it not a fact that the gases from the center of the furnace enter the boiler at practically the same temperature whether the walls are water-cooled or of refractory material? If the gases of combustion act as a shield for the transmission of heat by radiation, the answer to both of these questions must be "yes."

Another statement worthy of further discussion is the statement in the paper that we should limit the furnace temperature to the softening temperature of the slag carried along by the furnace gases. There is reason to believe that this cannot be done in many cases without a serious loss in furnace and boiler efficiency. Would it not be better to design the entrance to our boilers so that they can chill the molten slag below its freezing temperature without "bird nesting?" Such a design will result in more efficient utilization of the heat in the fuel.

The author is to be complimented on his frank discussion of the subject of water-cooled furnace walls. The extravagant claims made by some exponents of the use of such apparatus have been a hindrance rather than a help in the promotion of the use of this very useful boiler-plant equipment.

S. M. Finn.[5] The La Mont wall using small bare tubes approximates European practice, while the covered wall is strictly American. Both are dependent for their proper functioning upon

[4] Station Engineering Department, Edison Electric Illuminating Company of Boston, Boston, Mass. Jun. A.S.M.E.

[5] Vice-President and General Manager, Drake Non-Clinkering Furnace Block Company, New York, N. Y.

circulation—the former, forced circulation; the latter, natural circulation. It is upon this little-understood subject of circulation that these remarks are offered. They are conclusions based on academic investigations of our consulting engineer, Mr. J. Gould Coutant, of New York, and the practical experience of the writer and his colleagues.

The function of circulation in boilers is radically different from the function of circulation in water walls. Circulation in boilers is for the purpose of boiling water and liberating steam, but in water walls circulation within the water-wall tubes is the means of maintaining proper metal temperatures which enable tubes to withstand pressure, by carrying away heat from the zone of high heat concentration to the remote cooler portions of the boiler.

There are two fundamentals that affect circulation, as follows:

(1) Heat absorption by the fluid in the tubes
(2) Hydraulic head.

Heat absorption by water in water-wall tubes is subject to the usual unknown quantities encountered in heat-exchange calculations. It is further complicated by the absorption of radiant heat passing through the walls of the tubes to be absorbed by a very uncertain medium—water, water and steam, saturated steam, or superheated steam. The possibility of error in calculations is great.

Hydraulic head conduces the flow of water; the water is the vehicle for carrying away heat from the hot spots. The rate of flow depends on the difference in head between the heated and unheated columns of water. The flow is restricted by surface friction, eddies, and bends. It varies due to highly localized circumstances, outstanding among which are: variations in local rates of combustion; variations in the transparency of the gases, which affect the transmission of radiant energy; variation in firing.

The science of each of these two fundamentals is well known; combined, together with feedwater conditions which form scale, they present an intangible problem which can be solved only from practice.

In Europe it is typical to avoid evaporating water in walls in contact with radiant heat from fire, thus eliminating possibility of incrustation; also it is typical to use small bare tubes requiring forced circulation or its equivalent, for the purpose of maintaining the allowed safety metal temperature or safety creep stress. Such installations permit absorption of radiant energy up to 80,000 B.t.u. per sq. ft. per hour. Boilers in which this construction is illustrated are the Schmidt, Benson, Loeffler, and Brown-Boveri.

In America it is typical to evaporate water in water walls in contact with direct radiant heat from fire, avoiding the possibility of incrustation by lower rates of heat absorption; also it is typical to use large tubes that permit natural circulation. Large tubes, having thick walls, necessitate lower absorption.

Under standard furnace conditions, as defined by the writer in an earlier paper,[6] fire sends out radiant heat at a rate of intensity of 70,000 to 80,000 B.t.u. per hour per square foot of furnace enclosure. The absorption of this intensity must be reduced by protective tube coverings in zones of intense heat concentration. Bare tubes are used at distant points where radiant heat is diffused as though it were light. The reduction in heat absorption is for the express purpose of avoiding incrustation and maintaining the allowed safety metal temperature or creep stress. Constructions of this kind include the Drake armorclade tube, the Drake thermo fit water-wall block; Bailey & Foster.

Bare tubes (in American practice) are used for ordinary pressures which require relatively thin tubes. It is improbable that they will be used for high pressures, at least in zones of high heat concentration, due to the possibility of overheating the metal and exceeding the safety creep-limit stress.

The reason is obvious. The thicker the metal wall of the tubes the higher the temperature of the metal because of its resistance to the flow of heat through it.

Recent research shows that the tensile strength of metals is subject to another limitation than elastic limit and yield point, known as creep limit. Incidentally, a tube in which the creep-limit stress is exceeded may rupture a long time after the furnace has been in service.

Although the knowledge of "creep limit strength" of materials is insufficient to apply for all practical purposes, it will nevertheless be found to support the following contention, viz.: It is necessary to reduce the radiant-heat intensity to bare tubes as the wall thickness is increased. We submit the following practical demonstration:

Two decades ago 300 lb. was high boiler pressure. Tubes 4 in. in diameter, No. 8 gage, with a wall thickness of 0.165 in., were found satisfactory for exposure to a diffused radiant heat of an intensity of 85,000 B.t.u. per sq. ft. per hour.

Then stokers were installed with bridge walls located midway between the two headers of the boiler. As ratings increased, the radiant-heat intensity increased and tubes began to fail. Thicker tubes, to reduce the metal stresses, were used with some success, but satisfactory conditions were not obtained until the bridge wall was moved toward the rear header, which distributed the radiant heat and reduced the radiant-heat intensity back to 85,000 B.t.u. per sq. ft. per hour.

It is apparent that there is a relation between the safety creep-limit stress, incrustation, and radiant-heat intensity which is nearly proportional to the thickness of the wall of the tube.

For example, boilers built for 450 lb. working pressure having 4-in. tubes, No. 6 gage, 0.205 in. wall thickness, are now usually installed to absorb as a maximum 70,000 B.t.u. per sq. ft. of projected area of boiler-tube surface exposed to radiant heat of the fire per hour.

From the suggested axiom that permissible radiant-heat intensity is directly proportional to the tube thickness we deduce an empirical formula based on the known successful 300-lb. practice as a basis of comparison, as follows:

Tube thickness times intensity of the 300-lb. practice divided by proposed tube-wall thickness equals permissible radiant heat intensity. Set up as an equation applying to the 450-lb. boiler we have:

$$\frac{0.165 \times 85{,}000}{0.203} = 69{,}000 \text{ B.t.u. per sq. ft. per hour.}$$

This checks closely with manufacturers' practice.

A Schmidt boiler (European) with a closed-circuit water wall built for 1360 lb. pressure, has 2-in. diameter bare tubes, 0.188-in. walls and is absorbing between 70,000 and 80,000 B.t.u. per sq. ft. per hour.

From the foregoing formula, $\frac{0.165 \times 85{,}000}{0.188} = 74{,}600$ B.t.u. per sq. ft. per hour is the permissible limit, but no evaporation and the equivalent of forced circulation make higher absorptions possible.

A three-drum bent-tube boiler at Lakeside Station, built for 1390 lb. working pressure, with 3-in. tubes 0.3125 in. thick, was set high above the furnace, in which position 50,000 to 60,000 B.t.u. per sq. ft. of projected tube area could be absorbed. Computing, on the suggested basis, the allowable distributed B.t.u.

[6] "Water and Air Cooled Walls for High-Temperature Furnaces," delivered at the Iowa Engineers Society meeting, January, 1929.

per square foot per hour radiant-heat intensity for tube thickness, we have:

$$\frac{0.165}{0.3125} \times 85{,}000 = 45{,}500 \text{ B.t.u.}$$

It has been pointed out that water circulation in water walls is a combined problem of heat absorption and hydraulics. The water acts as a medium to convey heat away from a hot zone to a remote cool zone and is dependent for flow on the difference in hydraulic head between the heated and cool column of water.

At this point Coutant's Relations become important—that, if steam pressure increases four times, the velocity head is reduced to one-quarter due to smaller steam bubbles caused by the higher pressure and the velocity of the water is reduced one-half.

The bubbles weight increases approximately four times.

The slower velocity of the water in circulation will carry a greater amount of heat to remote cool sections per unit weight, but as boiler pressures increase, the internal diameters of the tubes decrease with increased resistance; therefore no advantage is obtained due to the relatively higher heat-carrying capacity of the high-pressure steam bubbles.

A unit of net tube cross-sectional area has a constant heat-carrying capacity for pressures from 150 to 1600 lb. per sq. in. It has been found that 4-in. tubes on 7-in. centers having approximately 10.5 sq. in. net cross-sectional area are satisfactory for carrying away approximately 50,000 B.t.u. per sq. ft. of water-wall surface per hour; equal to 1 sq. in. net cross-sectional area of tube per 2780 B.t.u. per sq. ft. of water wall per hour. This 1 sq. in. of net cross-sectional area will carry away 2780 B.t.u. per hour for every square foot of water wall, irrespective of the length of the tubes (within practical tube lengths) because of increased velocity caused by the greater hydraulic head due to the longer tubes.

On this basis, 3-in. tubes should be placed on 4-in. centers. If placed on 6-in. centers, the tubes should be recessed in the wall or covered to reduce absorption.

It also indicates that $2^1/_2$-in. tubes on $2^9/_{16}$-in. centers is the minimum limit of natural circulation. Smaller tubes or $2^1/_2$-in. tubes on wider centers require forced circulation.

It is believed that, if the foregoing rules are adhered to, no difficulties will be experienced with water walls due to circulation.

FSP-52-28

A Calorimetric Determination of Thermal Properties of Saturated Water and Steam From 0 Deg. Cent. to 270 Deg. Cent.[1]

By N. S. OSBORNE,[2] H. F. STIMSON,[2] and E. F. FIOCK,[3] WASHINGTON, D. C.

In this paper a description is given of the method, apparatus, measurements, results, and formulations involved in the work on steam which has been completed to date at the Bureau of Standards.

The method has been developed in accordance with the principles set forth in a rigorous analysis of the thermodynamic processes involved in a change in state of a saturated fluid.

The apparatus has been designed and built to provide a means for accurately controlling, observing, and accounting for the amount, the change in state, and the change in energy of a sample of water.

The observations have been carried out as a systematic series of experiments in the range from 0 deg. cent. to 270 deg. cent. (to 55 atmospheres pressure) yielding as a result the values of three characteristic thermal properties of the saturated fluid.

The formulation of the results establishes the values of heat content, latent heat, and entropy, and constitutes an exhibit of the thermal behavior of saturated steam in the region covered by the experiments.

The results are assembled in a coherent, consistent table of the properties of steam, available as a foundation for a more complete steam table.

It is found that, when reduced to a uniform basis, after critical study and analysis, most of the source data on saturated steam are in satisfactory accord.

I—INTRODUCTION

THE measurements here described were undertaken to meet a need for more data to be used as the basis for a steam table suited to present and prospective demands of steam-power engineering. The work was initiated as a result of the conference of engineers and physicists which was held in Cambridge, Mass., on June 23, 1921.[4] The program of experimental work undertaken by the Bureau of Standards was a part of a greater project which has been sponsored and largely supported by The American Society of Mechanical Engineers. Harvard University and Massachusetts Institute of Technology have cooperated in this project by undertaking definite programs of investigation.

The importance of international accord as to the basis of steam-power calculations has been recognized here and abroad. This has resulted in the international cooperation of engineers and investigators in the correlation of their work, with the object of ultimately developing a universally acceptable steam table.

In the program of securing new data for the revision and extension of the steam table, the part which was undertaken by the Bureau of Standards is concerned with the energy changes which occur in the fluid, water, when a mixture of the liquid and vapor phases is subjected to changes of state. The thermal properties which characterize these changes may be determined by calorimetric measurements made upon samples of the fluid under its own vapor pressure, or, in other words, at the saturation limit. In the present work a systematic series of such calorimetric measurements has been carried out. A group of thermal properties has been determined which constitutes a calorimetric survey of the behavior of steam in the saturation region extending from 0 deg. cent. to 270 deg. cent. The theory of the experimental method employed and the analysis of the physical processes are given in two publications,[5] of which the latter gives the more general and complete theoretical treatment. Brief descriptions showing the development of the design and construction of the equipment, the progress of the experimental work, and the tentative results obtained have appeared from time to time as progress reports.[6] Because of the tentative nature of the results previously published, the work has been described only briefly. In this report of the measurements completed to date, a more complete account will be given of the design, construction and use of the apparatus, and of the reduction and formulation of the results.

Recently a study[7] has been made of published source data from previous calorimetric work on water and steam in the saturation state. It is therefore unnecessary to include in this paper any of the details of that study, but the revised values and the appraisals there made serve as the basis for their comparison with the results of this present group of measurements. These results have been formulated into an exhibit of the calorimetric behavior of water as saturated liquid and vapor, with which other groups of source data may be correlated. This formulation, starting with liquid water under its own vapor pressure at 0 deg. cent., extends up to 270 deg. cent., and for this range constitutes a consistent, coherent table of observed thermal properties of saturated steam, and may be regarded as an available foundation for a more extensive formulation of the properties of steam.

II—GENERAL DESCRIPTION OF METHOD AND APPARATUS

Before attempting to describe any of the details of design, construction, and operation of the apparatus, which, it must be admitted, is somewhat complicated, the general scheme will be

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[1] Publication approved by the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Physicist, Bureau of Standards.

[3] Associate Physicist, Bureau of Standards.

[4] MECHANICAL ENGINEERING, vol. 43, August, 1921, p. 557.

Presented at the Second International Steam Table Conference, Berlin, Germany, June 23-26, 1930.

[5] *Jl. Opt. Soc. America & R. S. I.*, vol. 8, no. 4, 1924, p. 519. Also "Calorimetry of a Fluid." By N. S. Osborne. Trans. A.S.M.E., vol. 52 (1930), paper FSP-52-29.

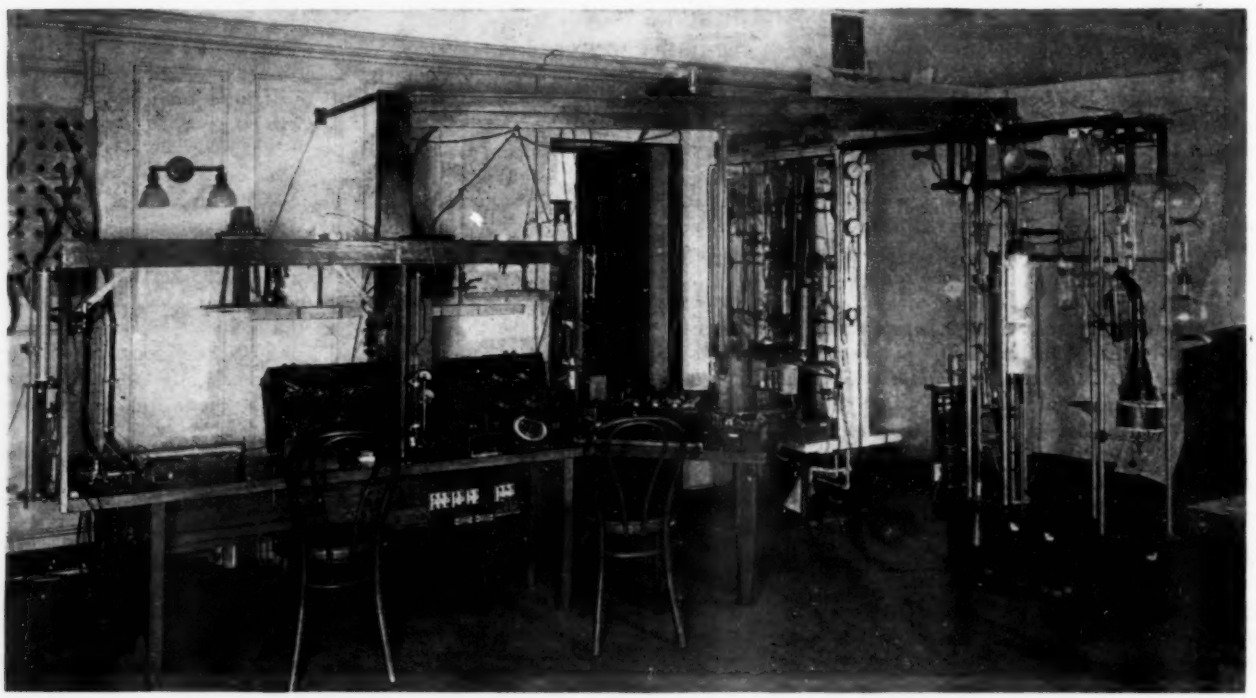
[6] MECHANICAL ENGINEERING, vol. 45 (1923), no. 3, p. 168; vol. 46 (1924), no. 2, pp. 81 and 83, and no. 11a, p. 808; vol. 47 (1925), no. 2, p. 106; vol. 48(1926), no. 2, p. 152; vol. 49 (1927), no. 2, p. 162; vol. 50 (1928), no. 2, p. 152; vol. 51 (1929), no. 2, p. 125; and vol. 52 (1930), no. 2, p. 127.

[7] "A Review of Calorimetric Measurements on Thermal Properties of Saturated Water and Steam." By E. F. Fiock. Trans. A.S.M.E., vol. 52(1930), paper FSP-52-30.

outlined. A general photographic view of the equipment as assembled in the laboratory for operation is shown in Fig. 1.

1—Method

The method makes use of a single calorimetric equipment by means of which measurements are made to determine several of the important thermal properties of the fluid. A sample of the fluid in a closed container at some chosen saturation state is heated electrically to some other chosen saturation state, or else is withdrawn either as saturated liquid or as saturated vapor. The energy required for each of these three processes is added and measured electrically. The amount of energy added per unit mass is characteristic of the particular experimental process observed. The group of measurements yields essentially the heat content or enthalpy, together with several other important properties, and establishes the thermal behavior of the fluid in the region covered by the survey.



Electrical Measuring Instruments Still for Purifying Water Assembled Calorimeter

Fig. 1 General View of Assembled Equipment

2—Apparatus

The apparatus consists essentially of a calorimeter which provides a place where a sample of water may be so isolated from other bodies as to enable its amount, state, and energy to be accounted for. The sample may be made to pass through a chosen, accurately determined change of state while the accompanying gain or loss of energy is likewise accurately determined. The design of the apparatus provides for several such experimental processes selected for their physical simplicity and for their fitness to exhibit the thermal behavior of the fluid. The arrangement of the apparatus is shown in the diagram, Fig. 2.

A quantity of water, part liquid and part vapor, is enclosed in a metal calorimeter shell. The water is circulated rapidly about the interior in such a manner as to distribute heat and promote close approximation to thermal equilibrium. An electric heater continually bathed with flowing water provides a means of adding measured energy which is speedily distributed throughout the calorimeter system. Outlets with valves provide for the introduction or withdrawal either of liquid or of vapor. Detachable receivers suitable for weighing are connected to the outlets to hold the samples of water transferred.

For confining the energy, the calorimeter is well insulated from the influence of external sources of heat and cold. In operation the temperature of an enveloping shell is kept very close to that of the calorimeter shell itself. The heat which passes by leakage to or from the calorimeter system is accounted for as a small correction which is determined. The energy consumed in circulation and added to the system as heat is another small correction which is determined.

Means are provided for observing the following quantities:

- *a* Temperature of the calorimeter and contents
- *b* Vapor pressure in the calorimeter
- *c* Mass of fluid contents of the calorimeter
- *d* Energy added to the system, including electric energy converted to heat and the small corrections for thermal leakage and circulation.

The apparatus is designed to permit three special types of measurements to be made. In the ideal case of perfect manipulation and control of experimental conditions these would consist essentially of observations of the following processes:

1. Heating with fixed amount of contents (Heat-capacity determinations)
2. Isothermal expansion by adding heat, evaporating liquid, and removing *saturated vapor* (Latent-heat experiments)
3. Isothermal expansion by adding heat, evaporating liquid, and removing *saturated liquid*.

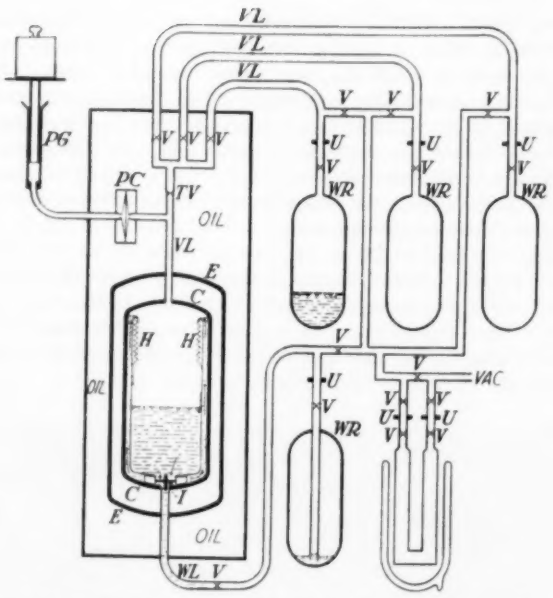


FIG. 2 DIAGRAMMATIC SCHEME OF CALORIMETER EQUIPMENT

C, calorimeter shell	*TV*, throttle valve for steam
E, envelope shell	*V*, valves
I, water-circulating pump	*U*, unions
H, calorimeter heater	*VAC*, vacuum line
WL, water line	*PC*, pressure capsule
VL, vapor line	*PG*, pressure gage
WR, water receiver	*LA*, liquid air trap
OIL, oil bath	

3—RÉSUMÉ OF THEORY OF METHOD

By using the results of the published theory,[8] the measured quantities may be expressed algebraically in terms of familiar thermal properties. The experimental determinations of heat capacity of a sample of water give as a result the increase in the quantity

$$H - L\frac{u}{u' - u}$$

over the observed range of temperature, in which H and L denote, respectively, in energy units, the heat content of saturated liquid and heat of vaporization per unit mass, and u and u' denote specific volume of saturated liquid and vapor, respectively.

In the latent-heat experiments we measure the energy which is added when water is evaporated and withdrawn as saturated vapor at constant temperature. This amount of energy per unit mass is equal to

$$L + L\frac{u}{u' - u}$$

The ratio $u/(u' - u)$ is the fraction of a unit mass evaporated, but remaining within the calorimeter when unit mass is withdrawn.

The quantity $L \times u/(u' - u)$ is complementary both to the determinations of heat capacity and to the determinations involving vapor withdrawal when the values of H and L are desired. This quantity is measured conveniently at any desired temperature by observing in supplementary calorimetric experiments of the third type the energy required to produce the necessary evaporation when a certain quantity of saturated liquid is withdrawn.

Thus the three different types of experiment yield values of three quantities referred to certain temperatures. These quantities,

$$H - L\frac{u}{u' - u}, \quad L + L\frac{u}{u' - u}, \quad \text{and } L\frac{u}{u' - u}$$

are denoted by the symbols, α, γ, and β. From these are derived finally the values of heat content of saturated liquid, H, heat content of saturated vapor, H', and heat of vaporization, L. In addition to these three energy quantities, characteristic of the fluid, there also may be derived from the calorimetric data the entropies of the saturated liquid and vapor; and by use of vapor-pressure data, obtained from other sources, specific volume and internal energy may be calculated.

The properties of water at the saturation limit which may be derived from the calorimetric data, and the formulas by which these derivations are made are as follows:

$$H = \alpha + \beta$$
$$H' = \alpha + \gamma$$
$$L = \gamma - \beta$$
$$\Phi = \int \frac{\alpha d\Theta}{\Theta^2} + \frac{H}{\Theta} = \int \frac{d\alpha}{\Theta} + \frac{\beta}{\Theta}$$
$$\Phi' = \Phi + \frac{L}{\Theta}$$

in which the following notation is used:

H = heat content per unit mass of saturated liquid ($\epsilon + \pi u$), referred to an arbitrary zero
H' = heat content per unit mass of saturated vapor, referred to the same zero as H
L = heat of vaporization per unit mass ($H' - H$)
u = specific volume of saturated liquid
u' = specific volume of saturated vapor

$$\left.\begin{aligned} \alpha &= H - L\frac{u}{u' - u} \\ \beta &= L\frac{u}{u' - u} \\ \gamma &= L\frac{u'}{u' - u} \end{aligned}\right\} \text{measured quantities}$$

Θ = temperature in centigrade absolute
ϵ = internal energy per unit mass
π = vapor pressure at saturation limit
Φ = entropy per unit mass of saturated liquid, referred to an arbitrary zero
Φ' = entropy per unit mass of saturated vapor, referred to the same zero as Φ.

III—DESCRIPTION OF CALORIMETER

The term "calorimeter" will be used here to denote that portion of the apparatus where the observed thermal process occurs. It is the part from which external influences are so isolated that the experimental processes may be accurately controlled and observed. The calorimeter is thus the relatively small part of the apparatus within which we may account accurately for energy added, amount and state of matter, and work done. It forms the nucleus around which the whole experimental equipment is developed.

The calorimeter was designed so that each of the three thermal processes previously mentioned could be separately observed. From these observations the corresponding group of thermal properties of water can be derived. Several general principles have aided in developing the design. These precepts may be enumerated as follows:

[8] Trans. A.S.M.E., vol. 52 (1930), paper FSP-52-20.

a A preliminary analytical study throws important light upon the applications of the thermal processes leading most directly to the desired results.

b Simplification of the experimental processes by refinement in the details of equipment and manipulation makes departures from the ideal experiment small or negligible, and thus promotes reliability.

c Advance consideration of the relative importance of individual refinements is more to be desired than makeshifts to account for and eliminate inconsistency of results.

d The apparatus should be moderately proportioned so as to promote accuracy, and facilitate construction, assembly, and operation.

e The best available materials and workmanship are most economical in the end.

f Proof tests of materials, particular features of design, and of partly assembled portions of the whole, reduce hazards to the completion of the work.

These principles may appear so obvious as to need no mention, but they suggest ideals to be sought and are recalled at this point to indicate a background against which closer details may be viewed in proper perspective.

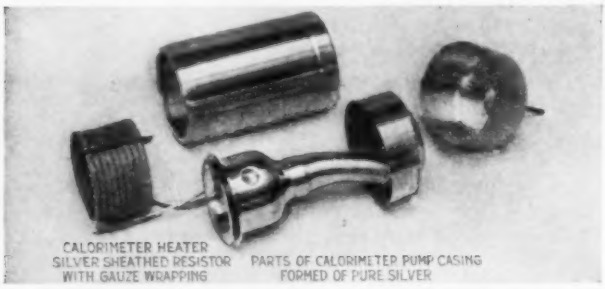


FIG. 3 PARTS OF PUMP CASING AND CALORIMETER HEATER

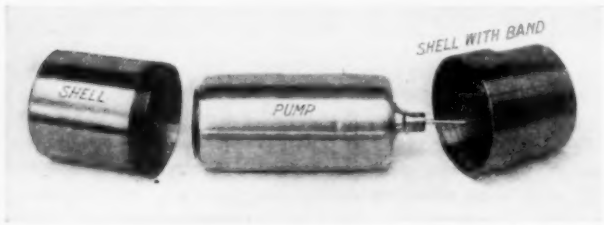


FIG. 4 CALORIMETER PUMP AND SHELL

1—CALORIMETER SHELL

The purpose of the calorimeter shell shown at *C* in Fig. 11 is to provide a receptacle or reservoir for the sample of water under observation. This shell is cylindrical in shape, having spherical ends, and is made of an alloy of 80 per cent copper and 20 per cent nickel. It contains, in addition to the water, the parts which serve as operating organs. These consist of a circulating pump, electric heater, etc., and will be described presently. To provide for installation of these parts the shell is made with a joint normal to its axis. The two main portions of the shell were shaped by cold drawing. Each is accurately machined, one with a right-hand and the other with a left-hand thread at the joint. The parts are brought together by a monel-metal band threaded correspondingly. The joint is made tight by the pressure which a tongue on one portion exerts on a gold-wire gasket lying in a groove in the other portion. To assemble this joint three special powerful wrenches shown in Fig. 5 were provided. Two of these are clamped about the smooth cylindrical surfaces, while the third wrench engages a series of rectangular notches in the monel band. The two clamps are yoked together at the outer end and a threaded drawbar provides the driving force in the final closure of the joint. In this operation sufficient initial stress must be developed in the band to insure a perfect seal against the vapor pressure of the water at any subsequent operating temperature. The amount of force with which the shell is drawn together was determined by making actual proof tests with saturated steam at a higher temperature than the experiments are intended to reach. As expected, these proof tests showed that the joint acted as a safety valve which would open at a rather definite steam pressure, and close again after the pressure was moderately relieved.

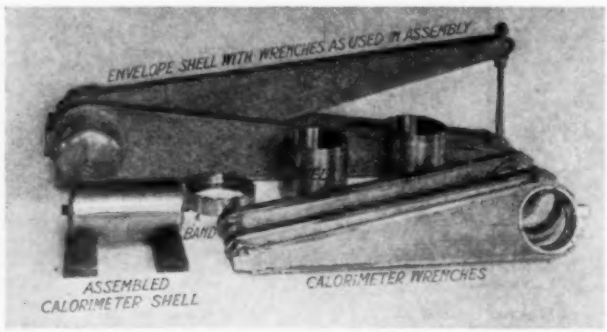


FIG. 5 WRENCHES AND MANNER OF ASSEMBLING SHELLS

The interior of the shell is heavily plated with silver, burnished after each successive plating. This is for the purpose of avoiding contact of the water with the base metal of the shell, silver being more resistant to hot water than the copper-nickel alloy. The exterior is likewise silver plated to give it a bright surface.

The various plugs and fittings which serve as ducts for tubular connections with the outside and for electric power connections are made of alloys of silver, and sealed with gold gaskets in tongue-and-groove joints.

2—CIRCULATING PUMP

The purpose of the circulating pump is twofold. It maintains a rapid circulation of the liquid about the walls of the calorimeter shell for distributing heat and thus promotes thermal equilibrium. It provides at the same time a source of liquid at the top of the calorimeter for keeping the heater surface bathed with water spread out in a thin, flowing stream.

Vigorous circulation lessens the time required to equalize the temperature of calorimeter and contents, and permits observations of steady initial and final temperatures to be made promptly.

Another advantage of the rapid distribution of heat is that, during the time when heat is being added electrically, temperature gradients on the calorimeter surface are kept small. The mean temperature of the surface may be obtained with reference to the opposed envelope surface by means of distributed thermo-elements for accurate evaluation of the thermal leakage.

The method of bathing the heater by a continuous thin flowing layer of liquid brings the evaporating surface close to the energy source and thereby accomplishes quiet evaporation, producing dry saturated vapor with very little disturbance of temperature in the entire system. This feature will be discussed later.

The construction of the pump is indicated in Figs. 3 and 11. It is of the centrifugal type, specially designed for the requirements of this particular case. The pump circulates about six

liters of water per minute, the equivalent of the entire contents passing around every five seconds. The flow circuit divides at the top, the larger part following the wall of the shell and the rest flowing through a port, down over the heater, and thence down to the inside storage space. To circulate the two streams, one of large flow and low head, and the other of small flow and greater head, the pump wheel or impeller is made in two parts rotating together and delivering into the same casing. As a result of tests of the characteristics and efficiencies of a series of model pumps, this final design combining two stages in parallel, each of different characteristics, was developed, and was found to give satisfactory heat-distributing capacity with a tolerably small power input.

The energy consumed in the pump is dissipated as heat added to the calorimeter system, and is objectionable if its amount is indeterminate or too great. Too large a pump energy input would interfere with the accurate observation of the initial and final temperatures. Too much fortuitous variation in the pump energy would impair the accuracy of accounting for energy from this source.

The normal speed of the pump wheel is 900 r.p.m. This speed was changed in certain experiments as found desirable. The method of speed control will be indicated later. The power input is determined calorimetrically and varies from 0.05 to 0.10 watt, depending on controlled conditions. The flow in the inner circuit over the evaporator and through the stage which maintains the head for keeping the main flow channel full is about one liter per minute. The flow in the main channel is about five liters per minute. The efficiency in pumping the water from intake to discharge at the top of the calorimeter was found to be about 28 per cent. The construction of a pump of this size and character designed to operate continuously in water up to 300 deg. cent. with no other lubrication of the running parts than that furnished by the water, built of materials which would be unaffected by the action of the water, involved some interesting problems.

The impeller runs on ball bearings of iridium alloy. The balls are about 1.2 mm. in diameter, ten balls in the lower and seven in the upper bearing. The main part of the runner is built up of silver-palladium alloy. The shroud and clearance ring of the upper stage are made of one piece of silver-gold alloy riveted to the runner. The vanes are machined to streamline curves. The lower stage has neither shroud nor clearance ring.

The five parts of the pump casing and flow channels are formed of pure silver. These parts go together with well-fitting slip joints fastened, where necessary, with pins. The entire pump, discharge tube, and flow-channel casing form a unit which, when assembled, is borne rigidly within the calorimeter shell on three supports of streamline form, machined from a solid piece of silver-gold alloy which fits the central opening at the bottom of the shell.

The pump runner is driven by a square on the lower end of the main runner shaft which engages the square in the upper end of the tubular propeller shaft of silver-palladium alloy. This propeller shaft extends down through the lower tubular support of the shell and on down to the stuffing box, where it engages the stuffing-box shaft of hardened tool steel with another square joint.

3—Stuffing Box

The stuffing box, built to provide for the rotation of the drive shaft for the calorimeter pump without permitting leak of water by it, is shown at *WSB* in Fig. 11. A cylindrical recess 5 mm. in diameter and 12 mm. deep in the monel body was fitted closely with two small ring-shaped packing retainers. The one which goes above the packing is of silver-palladium alloy, and the one below of phosphor-bronze. These retainers fit the 1.0-mm. hardened steel drive shaft within 0.01 or 0.02 mm. on the diameter, so that very little space is left through which the packing can flow. The packing consists of soft kid-leather washers impregnated with paraffin. Pressure is transmitted to the packing in the recess by means of an appropriate number of cupped washers of spring steel. These are compressed to the desired degree by means of a gland threaded into the body of the stuffing box. In order that there shall be no leak, it is necessary that the hydrostatic pressure in the packing be greater than the vapor pressure in the calorimeter. To prevent undue wear on the pump shaft the number of spring washers, the method of combining them, and the amount of deformation imparted to them are adjusted to meet the needs of the experimental conditions. To avoid the difficulties of finding a packing which would be satisfactory at high temperatures, the stuffing box is placed outside the lagging and water cooled. The size of the pump shaft where it engages the packing was reduced to 1 mm. to cut down the wear. Below the packing the shaft is larger and is provided with a ball bearing to take the downward thrust. The race for this bearing is attached to the outside of the body of the stuffing box by a threaded joint and can be removed readily for examination of the packing. A brass collar prevents undue upward motion of the shaft. Power for driving the pump is supplied from below, as will be more fully described later, and transmitted to the pump shaft by a hollow drive shaft having squared slip joints to provide easy assembly yet positive drive.

4—Electric Heater

The electric heater furnishes the means by which the major portion of the measured energy is added to the calorimeter system. As far as the heat-capacity experiments of the first type are concerned, no extraordinary restrictions as to characteristics of this part are necessary, as the exact path by which the change from an initial to a final temperature occurs is of no importance if the energy account is faithfully kept. In the case of evaporation experiments, and particularly those of the second type, much may depend on the way in which the energy is supplied and utilized in the evaporation process. The question as to the actual state of steam used in latent-heat measurements is the moot point which has frequently been raised as affecting the dependability of published values. If the energy could be supplied exactly at the surface where it is required for evaporation, none of it would go to superheating either the liquid or the vapor. Evaporation would then proceed quietly, and dry saturated vapor would be produced. In this calorimeter the electric heater, which is in fact an evaporator, has been developed to effect quiet evaporation and the formation of dry saturated steam with very little disturbance of thermal equilibrium in the calorimeter and contents.

The evaporator-heater as installed is shown in section at *H* in Fig. 11, and photographically in Fig. 3. In this electric heater a resistance wire of constantan 0.4 mm. in diameter, with gold leads of the same size, forms the resistor. This resistor is insulated by embedding it tightly in dry clay which completely fills the pure silver tubular sheath. The portion of the sheath containing the resistor is flattened to a thickness of about 2 mm. The end portions containing the gold leads are drawn to a diameter of about 2 mm. The flattened portion is wrapped spirally with a strip of pure silver fine-wire gauze, fastened by spot welding at a number of points. The flat resistor portion is then bent into the final helical form as shown, and the leads bent so as to pass downward and out of the shell through the fitting.

The purpose of the gauze wrapping is to insure even distribution of the stream of water which flows through the annular

port, and thence down as a thin sheet enveloping the evaporator.

The resistance of the completed heater is about 10 ohms. The insulation resistance between the resistor element and the silver sheath was found to be about 90 megohms when cold.

The evaporator when installed is held in its central position just below the water port by a three-armed silver support carried on the central portion of the discharge tube of the pump. The water is led from this port by guides of silver gauze and brought to the very top of the evaporator. The excess of liquid which flows off the evaporator is led quietly down into the central reservoir by a similar guide or apron of gauze.

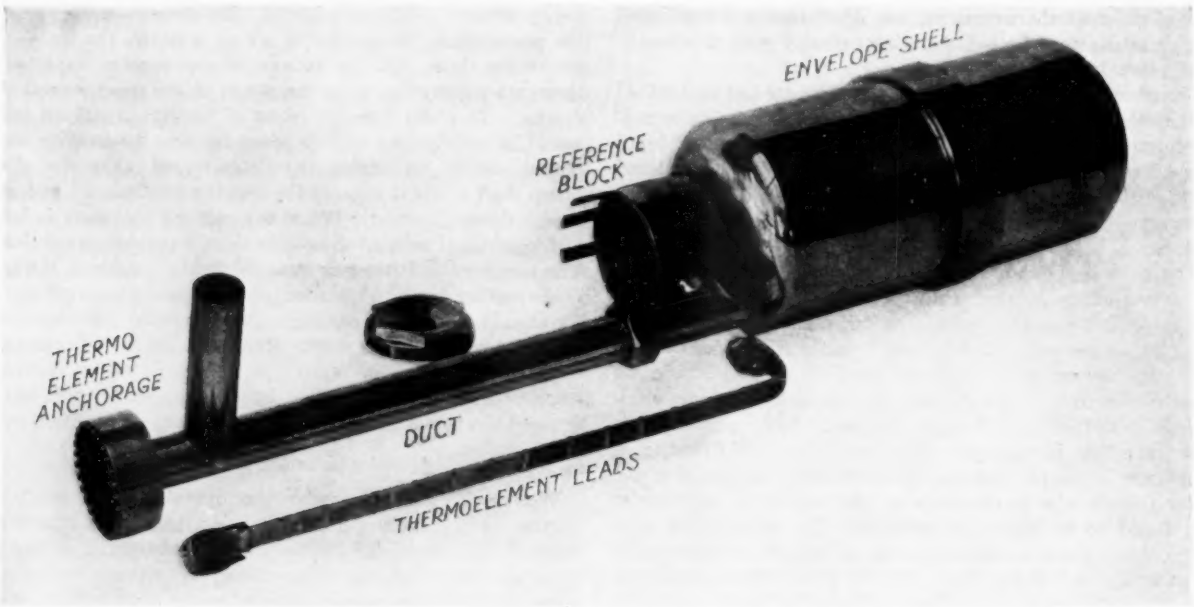


FIG. 6 ENVELOPE SHELL WITH REFERENCE BLOCK, DUCT, AND THERMOELEMENT LEADS

5—ENVELOPE

The chief purpose of the envelope which surrounds the calorimeter shell is to provide a receptacle in which the shell may be installed to isolate it from adventitious or fortuitous sources of energy.

We may regard the space between the calorimeter shell and the envelope as a barrier to heat flow, pervious to only a slight extent. The necessary metal connections between the calorimeter and its surroundings furnish other restricted channels for heat flow. The thermal-leakage rate may be made small by refinement of construction, but can only be annulled completely by avoiding temperature difference. This may not be actually possible, but by first making the thermal insulation very good and then in operation keeping the temperatures under control, it is possible to keep the heat leak extremely small in all cases and to account accurately for the unavoidable small heat leak which does occur. Provision has been made for both these measures in the present calorimeter, as will presently be shown.

The envelope is shown at *E* in Fig. 11. This outer shell is very similar in construction to the shell of the calorimeter. It is of the same material, made in two parts, assembled in the same manner. This shell is made strong as a precautionary measure in case the inner shell should fail to hold the water sample under pressure.

The calorimeter shell is supported axially within the envelope by means of two tubes of silver-palladium alloy, one at the top and one at the bottom. Besides furnishing strong, stiff supports, these tubes serve also as channels for transfer of fluid to or from the calorimeter, either as liquid or vapor. The lower tube also encloses the propeller shaft of the circulating pump. The upper tube serves as an opening through which a tool may be introduced for adjusting parts. The metal of these tubular supports was chosen because it possessed the desired characteristics of resistance to action of hot water, strength, thermal resistivity, and satisfactory workability.

To avoid the danger of injury to the delicate supporting tubes by accidental shock to the system when in place, six metal pegs are fixed in the wall of the outer shell, extending radially inward nearly to the surface of the inner shell but not touching it. These are placed in two planes near the top and bottom of the cylindrical part of the inner shell, and are so adjusted in length as to prevent any eccentric displacement which would endanger the supporting tubes.

Thermal leakage may take place by either of four types of heat flow, namely, radiation, solid conduction, gaseous conduction, and convection. The first of these, radiation, is made small by silver plating and polishing the outer surface of the calorimeter shell. This finish has a tolerably low emissivity in the temperature range of these experiments.

Conduction through the metal parts extending across the insulating space is kept within bounds by choice of materials and proportions of the necessary supporting tubes and electrical leads. The supports have already been described above. The electrical leads which contribute to thermal leakage comprise 21 chromel and 14 copel thermoelement wires each 0.1 mm. in diameter. Each of these materials has low thermal conductivity, and these leads contribute little to the total heat leak. The electric current and potential leads are a larger factor. These are of gold wire. The current leads are about 0.6 mm. in diameter and about 4 cm. long. The potential leads are about 0.2 mm. in diameter and 8 cm. long. They are disposed in the envelope space about as shown in Fig. 11.

Heat leak by gaseous conduction and convection was dealt with at first by providing means for evacuating the envelope

space to a low residual-gas pressure of about 0.0003 mm. of mercury. At this pressure the heat leak by this path is insignificant. The calorimeter was operated this way during much of the earlier work at moderate temperatures. At temperatures of 200 deg. cent. and over, trouble developed due to leaks appearing in the envelope system surrounded by oil. Presence of the small amount of oil which penetrated into this evacuated space, produced thermal effects which were baffling until finally

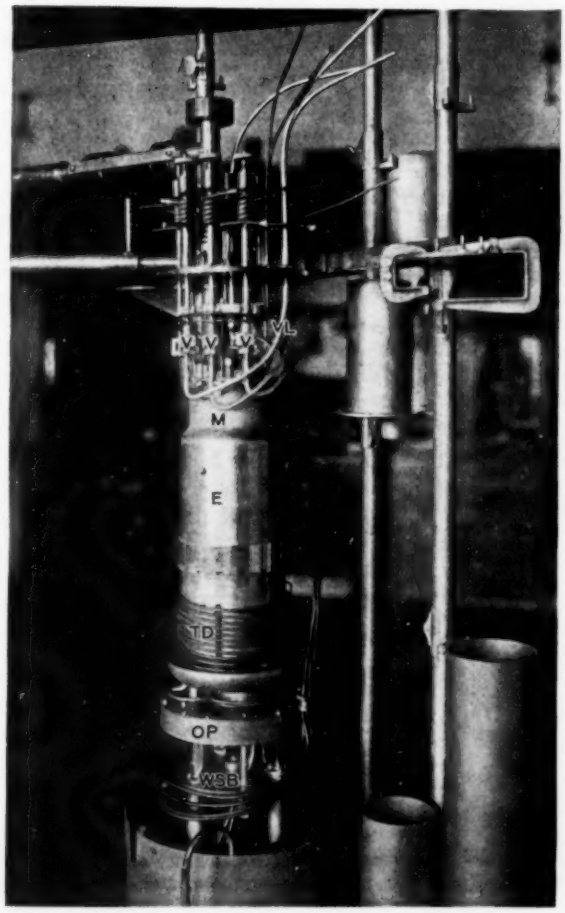


FIG. 7 PARTLY ASSEMBLED CALORIMETER

E, envelope shell
M, mantle for reference block
TD, thermoregulator dilatometer
V, shut-off valves
VL, vapor lines
WSB, water stuffing box
OP, oil pump

traced to their cause. After contending with this difficulty for some time, it was decided to discontinue the evacuation of the envelope. Instead, it was filled with nitrogen at about atmospheric pressure. This practice gave a slightly greater thermal-leakage coefficient, but inasmuch as this coefficient was determined as a part of the systematic routine and the actual heat-leak correction made in every experiment, a small percentage change in an already extremely small correction term was of no consequence.

Supplementing the above-mentioned features of thermal insulation of the calorimeter, there is a thermometric device for indicating the relative temperatures of calorimeter and envelope surfaces, and also a means for controlling these temperatures. The thermoelements which serve for the first of these form a part of the whole thermometric system which will be described in detail presently. The thermal control bath which surrounds the envelope and is the medium for effecting temperature control will next be described.

6—CONTROL BATH

The control bath as shown in Fig. 11 is contained in a brass casing, which, when assembled, supports the calorimeter in its envelope through the axial connecting tube at the bottom. The bottom of the bath casing is a circular brass plate which is supported by three flattened tubular columns. These columns rest on a lower deck, which is a part of the outer casing which holds the lagging. This lower deck is borne on a main support of three columns, which thus carry the main calorimetric unit.

The somewhat involved construction of these enclosures was made necessary in order to provide in advance for an assembly or dismantling without unnecessary disarrangement of delicate or remote parts. This precaution has proved a great advantage, when, in operation, either failure of a part or need for change or readjustment have necessitated access to more or less remote points inside.

A mineral oil of suitable viscosity and flash point is circulated in the bath by the path indicated in Fig. 11. A centrifugal pump drives the oil about this circuit. Each of four electric-heater units supplies energy for some specific purpose. The one for principal heat supply furnishes most of the heat which escapes to the outside, and part of that which is used to raise the temperature of the bath. This so-called "main heater" is located on the outer wall of the bath casing. It is made of nichrome wire, 0.8 mm. in diameter, wound first in helical form, then flattened to a ribbon-like coil. After the outer wall of the bath casing had been covered with a thin layer of alundum cement baked hard, this heating element was wound on, and another coat of alundum cement applied to hold it in place. The two windings have a resistance of 10 ohms each and can be used either singly, in series, or in parallel. This heater will of course not produce a quick response in the bath to a change in current. To provide for quick response an "auxiliary heater" is installed in the upper portion of the oil space above the envelope shell. This auxiliary heater is a copper-sheathed, clay-insulated unit similar in principle to the calorimeter heater but proportioned and shaped for this place. Its resistance is about 10 ohms.

To provide for automatic regulation of the bath temperature a third or "regulating heater" of 10 ohms, similar in construction to the one just described, was installed in the oil return flow just ahead of the pump intake. The current in this heater is automatically varied to maintain constant temperature in the bath by the thermoregulator, which will soon be more fully described.

A fourth heater, called the "central heater," is an auxiliary heater attached to the fitting on the central tube at *CH*, Fig. 11. This enables the operator to keep the upper end of the central supporting tube of the calorimeter at a higher temperature than the calorimeter so as to avoid condensation. The central heater is of the clay-insulated, copper-sheathed type wound spirally on a core of copper to which it is hard-soldered. This core is clamped to the fitting which it is desired to heat.

A refrigerating coil is installed in the bath to permit operation at temperatures down to 0 deg. cent. This consists of a tube as shown at *RC*, Fig. 11, and may be used to carry carbon dioxide, cooled by throttling from high pressure. By connecting the refrigerating coil to the water supply it can be used to cool the system after experiments at high temperature.

The thermoregulator is actuated by an oil dilatometer in which the variation of temperature causes motion of a mercury column in a small tube, thus making or breaking an electric circuit. This impulse operates a relay which effects the change in heating current. The thermoregulator is of the selective-setting type long used in this laboratory. When heating up, a

valve is left open, permitting the escape of expanded fluid. When the desired temperature is reached the valve is closed, heating currents are readjusted, mercury contact finally set, and the temperature is then automatically held.

The dilatometer of the regulator is a long, thin-walled tube wound in helical form and installed in the oil stream where it meets the envelope shell as shown at *TD* in Fig. 11.

The insulating material used for lagging is calorox. No attempt was made to make the heat leak from the bath very small by using great bulk of lagging. It was thought more expedient to conserve space and weight by wasting a little more electric power. The outer casing is water cooled to keep it from getting too hot to the touch.

The various connections extending up more or less rigidly from the calorimeter unit emerge through easily fitting tubular

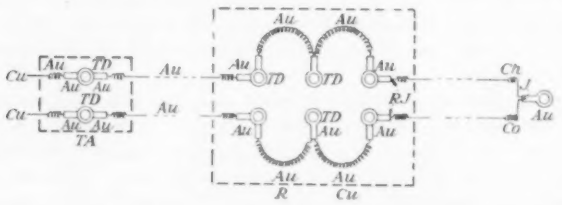


FIG. 8 DIAGRAM OF THERMOELEMENT

J, principal junction
RJ, reference junction
R, reference block
TD, thermoelement lead tie-downs
TA, thermoelement lead anchorage
Co, copel wire
Ch, chromel wire
Au, gold wire and terminals
Cu, copper

wells in the top of the control-bath casing. This permits near closure of the oil space, at the same time allowing for differential expansion of the parts. Flexible connections are brought out through tight fittings. Special wells are provided through which the resistance thermometers are inserted into their receptacles, yet to be described.

The control bath is designed to conserve space and heat capacity by utilizing the circulating oil as a medium for rapid transfer of heat rather than as a heat reservoir. It is necessary to provide for the change in volume of the oil over the temperature range used. An outlet at the top leads through a water-cooled tube to a trap with two oil reservoirs, each of three liters capacity. The supply flask is above, inverted, with spout below the overflow. The other is below to receive oil which overflows. This arrangement keeps the oil level at a definite height in the wells whether temperature is rising or falling.

7—THERMOMETRIC INSTALLATION

Temperatures at various chosen points on the calorimeter and envelope are observed by the combined use of platinum resistance thermometers and thermoelements. A heavy copper reference block located in the bath at *R*, Fig. 11, serves as the thermal union between the resistance thermometers and the reference junctions of the thermoelements. The thermometers thus measure the temperature of the reference junctions, and the thermoelements indicate the small temperature differences which exist between the reference block and the principal junctions located at the chosen points.

The three thermometers fit into copper receptacles which are hard-soldered to the reference block. Thirteen thermoelements, each having reference junctions on this same block, have principal junctions at points on the calorimeter shell and its envelope as shown in Fig. 11. Each element is made of 0.1-mm.-diameter chromel and copel wire, stranded so that two copel and three chromel wires lead to the junctions. Each element has three junctions, a principal one at the place where indication of the temperature is desired, and two reference junctions on the reference block. From these, gold wires lead to the outside. A diagram of a single thermoelement is shown in Fig. 8.

Copel-chromel elements were chosen because of their comparatively high thermoelectric power and relatively low thermal conductivity.

The reference block is especially designed to avoid temperature gradients and keep all the reference junctions and resistance thermometers at the same temperature. It is attached to the calorimeter envelope through a short tube of heavy copper-nickel. From this block a similar tube extends laterally and thence by a right turn vertically upward and out of the bath where it terminates in an airtight insulating seal, *S*, Fig. 11, for the wires, which are led out through this duct. A side connection near the top is used in the evacuation of the envelope or the filling with gas. The reference block itself is a heavy cylindrical mass of copper with circular steps inside like an amphitheater. These steps afford space for attaching the reference junctions of thermoelements. Three flat-tubular copper receptacles are attached to the block to receive the spatula-

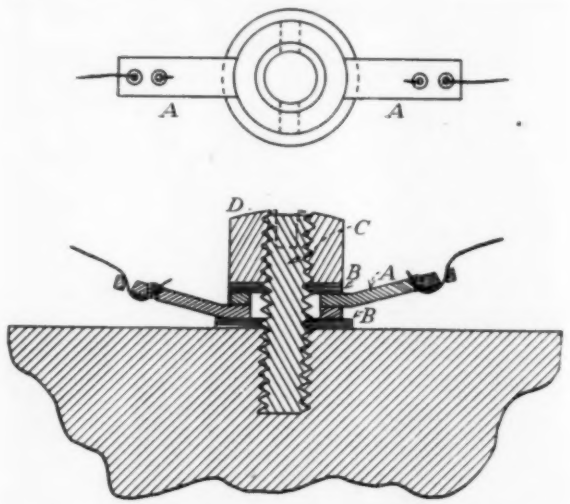


FIG. 9 DETAILS OF THERMOELEMENT ATTACHMENT

A, gold terminal of thermoelement wire, lead, or junction
B, mica insulating washers not less than 0.1 mm. thick
C, threaded stud *D*, nut

(Five times actual size.)

like ends of the resistance thermometers which are inserted from above. This well-conducting copper mass forms a thermal union between the thermometers and the thermoelements in which only extremely small temperature gradients can persist.

However, if left entirely unprotected from the direct influence of the oil stream whose temperature is controlled by the automatic thermoregulator, this block would experience sensible fluctuations of temperature from the controlled mean, which would interfere with precise observations. In order to damp out these fluctuations there is provided a mantle in the form of a figure of revolution fitting the cylinder and the envelope shell at *M*, Fig. 11. It moves vertically in guides so that it can be placed as shown, or raised to a higher level. When in the upper position the oil passes within the mantle and bathes the reference block directly. This is the proper position when the mean temperature of the bath is being changed. When steady temperatures are to be observed the mantle is lowered, thus keeping the reference block more remote thermally from the circulating oil and obliterating the periodic fluctuations, at the same time allowing the block to follow the mean temperature of the oil.

For insulating and supporting the thermoelements within the

envelope, where this cannot be accomplished merely by spacing the bare wires, mica is used. All the junctions are insulated with mica except for the principal junctions of Nos. 9 and 10, which are soldered directly to the central tube, and No. 12 which is assembled without insulation for test purposes. By making suitable connections outside, the thermoelements can be used either as individuals for survey of temperature distribution, or in groups to indicate average surface-temperature difference for heat-leak control. The group of five on the calorimeter can be used in series to refer the calorimeter temperature to the resistance thermometers.

The 28 gold wires leading out from the reference junctions in the reference block are carried through a multicellular duct built up of mica strips in which each wire has its own passageway. The wires which lead downward from the reference block to the principal junctions are guided and supported by threading through appropriate thin strips of mica. The method of attaching junctions to the metal parts so as to insure electric insulation and provide at the same time good thermal union is shown in Fig. 9. The gold terminal to which are hard-soldered the two wires forming the junction is clamped firmly between mica washers to the metal surface under a nut threaded to a stud. By the same device connections are made between the gold wires and the reference junctions, simply leaving out any mica washer from between the gold terminals. Thermal "tie-downs" of the gold wires are made in the reference block to intercept lead conduction which otherwise might reach the reference junctions and vitiate their indications. Each wire is twice tied down before connecting with its reference junction, thus reducing the effect of lead conduction from a possible error of nearly a degree, in an extreme case, to about one ten-thousandth of this amount. A total of 80 tie-downs find place within the reference block, and a total length of about 6 meters of bare wires, coiled in helical spirals, is accommodated in the 32 cu. cm. of space within this block, requiring considerable care in stowage to keep them all clear of each other. After assembly the whole system of insulated thermoelements showed an insulation resistance of several megohms to the calorimeter unit at 500 volts.

The thermoelements are made up of fine stranded wires so that they will have appropriate resistance and still be sufficiently flexible to be manipulated into suitable shapes for stowage. The total series resistance of the group consisting of elements Nos. 1 to 5, inclusive, is about 100 ohms. This resistance gives critical damping with the potentiometer and galvanometer used for observing. The resistances of the other combinations used are made up to similar total values by series resistance coils on the instrument board.

The copper ring *TA*, Fig. 11, furnishes an isothermal anchorage for the junctions between the gold lead-out wires and the copper wires which lead over to the instrument board. These circuits are entirely of copper to avoid parasitic thermal e.m.f.'s. Two special switches each of copper were built to permit the individual thermoelements to be quickly connected to the potentiometer in the desired combinations. One of these is a double-pole twelve-throw dial switch, and the other a quick double-throw six-pole knife switch. Where soldered joints were made, special care was taken to insure against thermal gradients in the joints. Every precaution was taken to reduce parasitic thermal e.m.f.'s in the measuring circuits to a negligible amount.

Four platinum resistance thermometers, specially constructed for this purpose, were used as working standards for the temperature measurements. These are of the four-lead potential-terminal type. They are made small and compact to fit the receptacles described above. The platinum windings are of 0.05 mm. diameter, wound seven turns to the millimeter on a flat plate of mica, notched at the edges to receive the wire. The windings occupy a length of about 10 mm. The thermometers are encased in platinum sheaths 0.5 mm. in outside thickness, 5 mm. wide, and 60 mm. long. These sheaths are hard-soldered to tubes of copper-nickel 3.2 mm. in diameter, through which the gold lead wires threaded through mica strips are led out to the seals at the top where copper wires are joined on.

The platinum windings were annealed by flashing, using an electric current of short duration sufficient to heat the wires to redness momentarily.

The thermometric reference block is closed at the top by a flexible annular diaphragm of silver to allow freedom for differential expansion between the outer envelope and the suspended calorimeter shell with its supporting tubes. The top of the block is terminated by a monel ring hard-soldered to the copper, having a blunt-edged flange upon which the silver diaphragm is clamped by the threaded nut bearing against a washer. At the inner edge the diaphragm is similarly clamped to the fitting which joins the supporting tube below and connects with the distributing tube above, through a union.

8—Throttle and Reheat Tube

This device has not been used up to the present time, but as it was installed in the calorimeter from the first assembly and had some influence on the design, although left inoperative, it will be briefly described. The extension of the silver-palladium outflow tube within the calorimeter is threaded in the lower portion of its length with a pitch of 80 threads to the inch. A plug of silver-gold alloy is fitted to this thread so that it can be placed anywhere in the threaded part. A coil consisting of 170 cm. of silver tube is connected at two points of the outflow tube. By placing the plug between these points, vapor flowing out of the calorimeter is first reduced in pressure and temperature as it leaks by the plug threads, and then flows through the coil where it is reheated by absorbing heat from the saturated vapor which surrounds the tube. This is the arrangement provided for a fourth type of experiment in which the heat content of superheated vapor would be determined.

As the calorimeter was used in the first three types of experiments the throttling plug was left out entirely, leaving free passage for vapor up the outflow tube.

IV—ACCESSORY APPARATUS

1—Flow Lines and Control Valves

The flow lines provide connections between the calorimeter and the outside receivers, and are used to conduct the flow in the transfer of fluid to or from the calorimeter. The lines at the bottom provide for the introduction and withdrawal of liquid, while those at the top are used in the withdrawal of vapor. Fig. 2 shows diagrammatically the various paths along which liquid and vapor can be conducted, and the relative positions of the control valves.

The valves used in the flow lines are all of the diaphragm type, which permits motion for operation without the use of packing, and will be described later. Part of these can be operated as shut-off valves, and two as throttles for reducing pressure and controlling rate of flow.

All flow lines are of silver or silver-palladium tubing. Those which are at the pressure existing inside the calorimeter are of silver-palladium, except one small silver tube at the bottom. This bottom tube emerges from the side of the lower supporting tube, in the lagging space, and leads downward near the stuffing box to the outside as shown at *WL* in Fig. 11. It serves for introduction or removal of liquid, and is the only connection to

the calorimeter which has to be broken for dismantling the control bath. The silver-palladium exit tube at the top of the calorimeter has two horizontal branches, one leading to a throttle valve and one to a pressure-transmitting device or pressure capsule, which will be described later. From the throttle valve a silver-palladium manifold tube leads to three shut-off valves located in the control bath. From each of these a comparatively large silver tube leads out through the top of the control bath and the lagging to a silver tempering coil which is provided with means for evacuation, through a diaphragm valve. The liquid line at the bottom leads through a diaphragm valve which serves either as a liquid throttle or shut-off, to a similar tempering coil. All four tempering coils terminate in unions which fit the threaded portions of the valves on the containers for the water. All four vacuum valves lead to a common vacuum line, which is provided with a metal trap in which water remaining in the connecting lines can be caught and weighed.

The diaphragm valves already spoken of were designed to eliminate the use of packing. One of these will be described to illustrate this feature. In the section shown at *VV* in Fig. 11 it may be seen that the action which opens and closes the valve is the vertical motion of a stem relative to a seat. The face of the stem is a soft silver disk, and the seat in this type is a rather blunt but finely machined edge of silver-palladium alloy. Longitudinal motion of the stem is permitted by an annular silver diaphragm of 0.2 mm. thickness, clamped to seal tightly near the outer and inner edges. Rotation of the stem is thus prevented. The portion of the stem outside the diaphragm is a cylinder fitting freely in a recess in the valve bonnet, so that the stem moves perpendicularly to the plane of the seat. Excessive flexure of the diaphragm is prevented by the metal support behind it, and by limiting the travel of the stem. In this type of valve used on the containers and in the vacuum lines the motion is imparted to the stem by a differential thread arrangement. The differential screw has a square end which can be turned by a detachable key.

Each of the three shut-off valves in the bath, one of which is shown at *V*, is of this same diaphragm type, but the motion is imparted to the stem in a different manner. These valves have to be opened and closed as nearly instantaneously as possible, and no intermediate positions in the motion are required. Therefore, instead of the differentially threaded stem, an entirely different mechanism was designed to accomplish this purpose. Their stems are attached to vertical struts, leading outside the control bath to a spring and lever system shown at *VM*. By means of the adjusting screws illustrated, the mechanism can be set so that any desired force of the stem on the seat will be furnished by deformation of the spring lever *SL*.

It is evident from the diagram that by a single motion of the operating lever *OL* the valve can be completely opened or closed. The provision by which these operations are automatically carried out at a signal from the standard clock will presently be described.

The shut-off valve in the liquid line was changed during the liquid-withdrawal experiments from the type shown at *WV* to a type similar to the vapor shut-off valves in the control bath. Provision was made for the remote control of a slow-motion screw which actuated the handle of the mechanism. The opening of the valve could thus be controlled to regulate the rate of withdrawal of liquid.

The vapor throttle valve is similar to the shut-off valves, but because of the fineness of operation demanded of it, certain changes were necessary. The seat is made flat with a 0.5-mm. aperture, and the silver disk on the stem is made slightly conical where it engages the seat. The operating mechanism outside the oil bath was changed to the form shown in Fig. 11 and was provided with a slow-motion screw to operate the valve handle. This slow-motion screw is operated from the instrument table by a positive mechanical control. The spring lever was replaced by a heavier brass one which acts as a rigid member. All joints are made as free from backlash as possible, and the spring *K* is provided to take up the residual slack. For some of the latent-heat experiments at high pressures, a set of 8 silver-palladium disks, each 0.1 mm. in thickness and with 0.5-mm. holes in the centers, was introduced between the valve stem and the seat to provide more paths for vapor and thus cut down the sensitivity of the valve to motions imparted to it by the observer.

Considerable difficulty was experienced in the evolution of the throttle valves. The two described proved to be the most satisfactory of several which were tried.

2—Containers for Water

The containers for the water, as shown at *WR* in Fig. 11, are made of 1.6-mm. silver in cylindrical form with hemispherical ends. Their capacity is about 425 cu. cm. Those for use in transferring liquid are provided with reentrant silver tubes reaching nearly to the bottom. Each container is provided with an outlet tube of silver bent as indicated in the figure and leading to a diaphragm valve. The body of this diaphragm valve terminates in the male half of a union, threaded to fit any of the unions on the tempering coils. The weight of each container and its valve is about 1000 grams.

3—Pump-Speed Control

The speed of the circulating pump in the calorimeter is accurately controlled to make the rate at which it supplies energy to the water definite and reproducible.

The long external drive shaft for this pump has squared ends which couple the pump positively to a liquid tachometer rotating at the same speed as the pump. The tachometer, of the well-known Veeder type, is driven by a motor which draws its power from a 50-volt storage battery. The impeller maintains a column of kerosene at a height which is determined by the speed. A column of mercury, in a side tube off the kerosene column, is maintained at a level which also varies with the speed. In a manner similar to that of the thermostat, this mercury column, by its motion, serves to make and break an electric circuit. As a result, more or less resistance is placed in the armature circuit of the driving motor, in the direction required to maintain a constant head of mercury. The kerosene column furnishes a sensitive means for the direct observation of the pump speed, which latter is maintained practically constant by the automatic control.

To check the constancy of the pump rate, a revolution counter was geared directly to the pump shaft. Observations with this counter at infrequent intervals showed that the pump rate is maintained constant at any desired value to about 0.1 per cent.

4—Timing Devices

In the α and β experiments the time of electric heating, and in the γ experiments the time of withdrawal of vapor, must be observed. For the first of these there was constructed a double-pole double-throw switch operated by a spring and a release which is actuated by an electric impulse from the standard clock. This automatic switch not only opens or closes within less than a tenth of a second, but also requires the same time interval for action in either direction. In operation it was used to switch the connection from a 30-volt storage battery to either the calorimeter heater or to a spill coil having nearly the same resistance.

To provide for timing the flow in the γ experiments, the

actuating mechanisms of the three previously described shut-off valves are equipped with pulley wheels rigidly attached to the operating levers and rotating about the same axes as these levers. Cords firmly attached to the wheels can be led in either direction around the wheels and thence up to a spring. The spring is permitted to operate by the release of a catch at a signal from the standard clock. The spring thus furnishes the power for opening or closing the valves at a chosen time, either individually or in any desired combination. The time required for complete opening or closing of a valve is less than 0.1 sec.

Signals from the standard clock are received only on the exact seconds, the 59th being omitted. The key in the clock circuit is closed by the operator during the 59th second so that the zero signal is used either for starting and stopping the power or for diverting the vapor flow. The integral number of minutes involved in an observed process is observed on a watch.

5—Pressure-Measuring Instruments

Provision was made for observing pressure, and although no measurements are included in this report, the installation will be described. A side tube from the outflow tube just after it emerges from the vacuum space leads to a pressure capsule *PC*, Fig. 11, which is also in the oil bath to avoid condensation. The pressure capsule is made of monel metal with a 4-cm.-diameter sheet-monel diaphragm, *D*, 0.08 mm. thick, stretched across the capsule. This diaphragm transmits the pressure from the vapor on one side to oil on the other. The diaphragm is allowed a displacement at the center of only 0.25 mm. from the mean position before coming against the solid supporting walls of the capsule. When the highest working pressure is applied, no permanent deformation of the diaphragm will result. The diaphragm, however, is sufficiently flexible for a very small pressure change to deflect it across the chamber. A measurement of the opposing pressure when the diaphragm is balanced midway in the capsule will then give the pressure of the vapor. All the monel surfaces on the vapor side are silver plated to prevent chemical action of the hot vapor with the metal. A 0.2-mm. gold-wire ring was used to make the diaphragm gas-tight in the capsule. Monel screws were used in addition to the monel case and diaphragm so that there would be no differential expansion of the parts, which would cause leaks or change in tension of the diaphragm. Oil is used to transmit the pressure from the diaphragm in the capsule to a gage of the deadweight type with piston rotating in oil.

6—Electrical Measuring Instruments

The electrical measurements comprise observations of thermometer resistance, thermoelectromotive force, and electric power input.

The resistances of the thermometers are measured with a Mueller[9] bridge built by O. Wolff. The bridge coils are immersed in a thermostatted oil bath. A separate commutator switch permits the observation of any chosen one of four thermometers. The bridge coils were recalibrated whenever it was found desirable by the method outlined by Mueller. A measuring current of 4.5 milliamperes was used both in the calibration and in the measurements of temperature. A galvanometer-scale deflection of 1 mm. corresponds to approximately 0.0001 ohm, or about 0.001 deg. cent. for the thermometers used.

The electromotive forces of the thermoelements, amounting in nearly all cases to less than 20 microvolts, are measured on a Wolff potentiometer designed by F. Wenner. By comparison with another instrument which had been calibrated at the Bureau of Standards, the corrections to the indications of this instrument were found to be negligible in the range in which it is used. When the temperature of the calorimeter is being observed, a scale deflection of 1 mm. corresponds to 0.002 deg. cent. or less.

The power input is measured on a Diesselhorst potentiometer built by O. Wolff. It was calibrated at the Bureau of Standards before, during, and after the experimental work, and showed no changes of more than 1 part in 30,000. The scale deflection was greater than 2 mm. per division on the last dial in the measurements on current and potential drop.

The standard cells to which the potentiometric measurements are referred are Eppley unsaturated cadmium cells. They are kept in a cork-lined box and calibrated frequently against the standards maintained at this Bureau. Two of these cells are used with the potentiometer for power measurement to guard against fortuitous changes in voltage between calibrations. Their electromotive forces, as measured and used in this work, are expressed in international volts. This international volt is defined at $^1/_{1.0183}$ of the voltage of the normal Weston cell at 20 deg. cent.

The volt-box ratio and the resistance of the 0.01-ohm manganin standard used in the measurements were verified at this Bureau before, during, and after the experiments, and showed no changes of more than 1 part in 30,000.

7—Weighing Instrument

The balance used in the weighings is a two-kilogram Troemner balance with a sensitivity of one scale division per milligram. Provision was made for suspending the counterpoise and the containers for water in a closed cabinet below the balance pans. All weighings are made by the method of substitution. The weights are of brass, platinum plated, and calibrated at this Bureau. Correction is made for buoyancy of the air on the weights. In this method of weighing, no correction for the air buoyancy on the water sample is necessary. Changes in buoyancy are compensated for by making the volume of the counterpoise approximately the same as the volume of a container. The weighings are made to the nearest milligram.

8—Thermometric Standards

All temperatures used in this work are measured on the international Centigrade scale of 1927,[10] which is defined, in the interval 0 to 660 deg. cent., by assigning the temperatures 0, 100, and 444.60 deg. cent. to the ice point, steam point, and sulphur boiling point, respectively, all at the pressure of one standard atmosphere (760 mm. of mercury); and establishes other points by means of a standard resistance thermometer of pure platinum, making use of a formula of the form:

$$R_\theta = R_0(1 + A\theta + B\theta^2)$$

in which θ is temperature, R_0 the resistance at the ice point, and A and B are constants found by calibration at the steam point and the sulphur boiling point.

Before starting the calorimetric measurements the resistance thermometers were calibrated directly at the ice, steam, and sulphur points as outlined above. Later on in the work it was found expedient to calibrate the thermometers by comparison with a laboratory standard. A standard resistance thermometer of the strain-free type was inserted into a special copper receptacle which is in good thermal contact with the copper reference block in the control bath. This strain-free thermometer is calibrated at the ice, steam, and sulphur points directly, and the constants of the calorimetric spade-type thermometers are determined by direct comparison in position in the control bath. In this way it was possible by holding the bath at a constant temperature to

[9] Mueller, Bureau of Standards Scientific Paper No. 288.

[10] Burgess, *B. S. Jl. of Research*, Research Paper No. 22, p. 635, Oct., 1928.

make very accurate comparisons between the strain-free thermometer and the calorimetric thermometers. However, the strain-free thermometer was not so well adapted for indicating the temperature of the upper reference block during the calorimetric experiments, and for this reason the calorimetric thermometers were used in the actual experiments.

The thermoelements used in measuring the calorimeter temperature were calibrated in place against the resistance thermometers. The accuracy of this calibration is of little importance because of the extremely small values of the observed thermoelectromotive forces.

9—Purification of Water

Dissolved gases are removed from the water samples used in the experiments. In a specially designed still, ordinary distilled water is subjected to another distillation under a pressure of about 0.1 atmosphere. During this redistillation, a pump is continually removing the permanent gases, together with some water vapor from the condenser, at such a rate that there is little opportunity for the permanent gases to redissolve in the condensate. In special test experiments the amount of residual gas remaining in the condensate was found to be less than $^1/_{1000}$ of the amount usually contained in distilled water. After a water sample is thus freed from dissolved gases it is not allowed to come into contact with air again either before or during its use in the experiments, but is always kept and transferred under its own vapor pressure.

V—DESCRIPTION OF MEASUREMENTS

The keynote in the description of the experiments is that an accurate account must be kept of each of three primary observed quantities, namely, the amount of fluid subjected to the process, the change in state, and the amount of energy exchanged. The energy account consists primarily of that which is added and measured electrically, but includes also the energy added by the pump and the energy exchanged by thermal leakage. In the following discussion the method employed for the evaluation of all these factors will be described.

1—Accounting for Mass of Water

The mass of water which is subjected to a change in state enters as a direct factor in the reduction of the data. The results, therefore, can be no more reliable than the determinations of the masses. In addition, a quantitative account of the aggregate mass of the water involved in any set of experiments throws important light upon the reliability of the data derived from the experiments. Equality of the amount put in to the total removed indicates that there has been no gain or loss of fluid through leaks, and that there have been no accidental errors in the determination or recording of weights. In this work such a mass accounting has been carefully kept, and important revelations have resulted from it. From time to time water leaks from the calorimeter and air or oil leaks into the calorimeter have been indicated, located, and eliminated. When the mass accounts balance properly, the results assume an added degree of certainty.

The procedure employed in filling the calorimeter and in removing that water which remained after the completion of a series of experiments is the same for the α, β, and γ experiments. It will therefore be described here once for all.

A container having a reentrant tube is evacuated and filled with a charge of water from the still. The valve on the container is heated to a little above 100 deg. cent. to drive off external moisture, and the whole is cooled in a current of air, from a fan, to the temperature of the room. This treatment was found to bring the containers to a reproducible condition for weighing, and will be spoken of hereafter as "conditioning."

The container and its charge of water are first weighed and then attached to the tempering coil at the bottom of the calorimeter. The calorimeter and coil are then evacuated through the vacuum valve, until a McLeod gage in the system indicates a pressure of less than 0.001 mm. The system is then considered tight and sufficiently gas free.

The vacuum valve is then closed and the container valve opened. The water in the container is forced into the calorimeter by its own vapor pressure when heat is applied with a gas torch. Usually the entire charge is introduced, but the flow can be stopped after any chosen part has gone in if the container is suspended during the filling from one arm of a crude balance.

When the desired mass of liquid has flowed into the calorimeter, the lower calorimeter shut-off valve is closed, the tempering coil is surrounded with boiling water, and the container is immersed in cold running water. Virtually all the water remaining in the lines is thus distilled back into the container. After a few minutes the container valve is closed and the coil is again evacuated, this time through the liquid air trap which has previously been evacuated and weighed. The vacuum valves and the valves on the trap are closed and the container and trap are detached, conditioned, and weighed. The change in weight of the container minus the weight of the water caught in the trap (always less than 1 milligram, and therefore negligible when the operations are properly carried out) gives the mass of the water in the calorimeter system.

After a set of experiments in which part of the water sample may or may not have been removed, the water remaining in the calorimeter is withdrawn into the container. This process is hastened by having the calorimeter and tempering coil hot and the container cool. When as much liquid as will flow out has been collected in the container, its valve is closed and that which remains in the calorimeter is collected in the liquid air trap. The masses thus removed are weighed as described above.

In the constant-mass or α experiments the mass weighed in is equal to the mass removed if the operations have been properly performed. In the withdrawal or β and γ experiments the masses of water removed as saturated liquid or vapor must be included in the mass accounting.

2—Heat-Capacity Determinations

The measurements of the first type which determine the change in value of the quantity $H - L\frac{u}{u' - u_,}$ denoted by the symbol α, were, with a few exceptions, taken in ten-degree intervals. The whole range from 0 deg. cent. to 270 deg. cent. was divided into three sections, 0 to 100, 100 to 200, and 200 to 270, each of which could generally be covered in a single day's work. Starting at the lowest temperature of one of the sections, 0 deg. cent., for instance, the initial temperature is observed. Then electric energy is added to heat the calorimeter and water approximately 10 degrees. The temperature is again observed and energy added for another period. There is thus obtained a series of observed temperatures, between which the added energy is measured. By keeping a continuous account of the energy, including heat leak and pump energy, the change in α from the initial starting point of the series to any stopping point can be determined independently of the intermediate intervals. A single series of measurements of this kind gives as a result the amount of energy, Q, added to change the temperature of the calorimeter and contents from any one to any other of the observed temperatures. A second series carried out in a similar manner over the same intervals of temperature

but with a different mass of water in the system gives as a result a second series of values of Q corresponding to these same temperature intervals. Two such series of measurements suffice to determine a series of values for the changes in α and for the changes in another quantity which is designated as Z, over the temperature intervals covered. This quantity Z, as described more fully in the paper on the analysis,[11] is a characteristic property of the calorimeter as used, and its value depends on the temperature but is independent of the total amount of water in the system. Actually, in each range of temperature several series were made in which the masses of water contained were chosen so as to give large differences in total heat capacity.

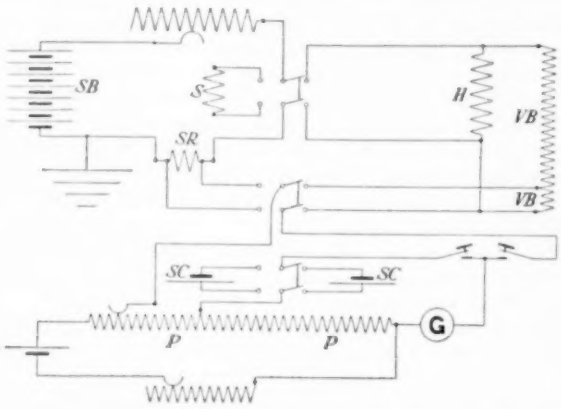


FIG. 10 WIRING DIAGRAM FOR CALORIMETER HEATING POWER AND ITS MEASUREMENT

H, calorimeter heater	VB, volt box
S, spill coil	P, potentiometer
SB, storage battery	SC, standard cells
SR, standard resistance	G, galvanometer

Reduction of the results of such a group as a whole by the method of least squares yields a series of values of α, which is a property of saturated liquid water, and a series of values of Z, which is a property of the calorimeter as it was used. The individual values of α, corresponding to each experiment, can then be calculated for purposes of comparison to show the degree of reproducibility of the measurements of α.

In all the measurements of this type a tight-fitting cap seals the calorimeter at A, Fig. 11. The water samples are introduced and removed as liquid through the connection at the bottom of the calorimeter as previously described.

The shut-off valve in the water line is outside the boundary of the calorimeter, and so a small part of the water introduced into the system is outside the calorimeter and does not enter into the thermal process observed. This fact in no way affects the result because the method of reducing the observations eliminates it as a quantity independent of the amount of filling. Only minor differences, such for instance as fortuitous changes in the temperature of the small emergent portion of the water sample, can affect the results. These effects would appear as accidental errors in a large group of measurements.

The energy imparted by the pump during an experiment is taken into account in computing the results. The pump power is determined by supplementary experiments in which no electric power is supplied and heat leak is determined, as will be described presently.

In the heat-capacity experiments it is not important that the pump power be known accurately. The method of reduction by combining results of experiments differing only in amount of water sample used eliminates the pump energy from the final result except for its variation from one experiment to another Changes in this factor would enter as accidental errors in a series. Experiment showed that no systematic difference in pump power resulted from the use of different amounts of water.

The group of measurements in the range 0 to 100 deg. cent. differed from those in the other higher ranges in several respects. In the first place, it was necessary to cool the calorimeter from the laboratory temperature down to the starting point of the series as a preliminary to the experiments. At best this preliminary cooling was a slow process. On the other hand, the pressures in this range were low and the work was freer from leakage troubles incident to pressure. The measurements from 100 deg. to 200 deg. could be made more expeditiously, but difficulties incident to the higher temperatures and pressures were greater. Above 200 deg. these difficulties increased and other obstacles to operation were encountered. These resulted in a greater percentage of failures, and necessitated occasional suspension of the observations for changes and repairs.

A determination of heat capacity in the range 0 deg. to 100 deg. will be described to illustrate the principal features of the determinations over any temperature range. The procedure is in general as follows:

A weighed sample of water is introduced into the calorimeter. To provide a medium for extracting heat from the calorimeter and expedite the cooling down to the initial temperature of the series, helium is introduced into the space between the calorimeter and its envelope to a pressure of about 10 cm. of mercury. This gas was used because of its relatively high thermal conductivity.

Carbon dioxide is used as the refrigerating medium both for cooling the bath down and for holding it at the low temperature while the calorimeter is being cooled. During this period of cooling the calorimeter the bath is held very near but not below 0 deg. cent., because freezing of the water in the lower tube must be avoided.

The automatic thermoregulator is used during this time to maintain a practically steady temperature of the bath. It operates by compensating the excess refrigeration with electric heating, automatically regulated in the manner previously described.

When the calorimeter has cooled to the desired temperature, the bath temperature is raised to the same value. The helium is then pumped out of the envelope space, leaving the residual pressure less than 0.001 mm. of mercury. The automatic thermoregulation, the pump speed, and the power supply which is to furnish heat to the calorimeter are all finally adjusted. The mantle around the reference block is lowered, and after a short wait for a steady state to be reached the preliminaries are complete and the observations are begun.

The initial temperature of the calorimeter is measured by taking simultaneous readings on the resistance thermometers and the e.m.f.'s of the five thermoelements on the calorimeter in series. Two observations on each of the three thermometers are taken for a determination of a single temperature. Readings are begun two minutes previous to the time chosen for switching on the power to the calorimeter heater, and are continued at twenty-second intervals until the six temperature observations have been made. The order used in reading the thermometers is not important, but was kept the same at the beginning and end of each experiment.

The operator increases the energy supply to the bath a little before switching on the power to the calorimeter heater, so that the envelope temperature and the calorimeter temperature will start to rise at nearly the same time and the same rate. On the chosen instant the automatic switch is thrown and the current started in the calorimeter heater. The operator, guided by indications of the differential thermoelements, keeps the power

[11] See reference in footnote 8.

supply to the bath so adjusted that the rising temperatures of the calorimeter and envelope are kept very nearly together.

At one-minute intervals the small temperature differences as indicated by the differential thermoelements are observed. The recorder observes the current and potential drop in the heater on the alternate minutes.

The heating is continued for an integral number of minutes and stopped on the zero of that minute when the temperature is nearest to 10 deg. By proper preliminary adjustment of the power supply the final temperature is brought in nearly all cases to within less than 0.1 deg. of the chosen even temperature.

Just before the end of the heating period the operator decreases the power supply to the bath so that the bath temperature will not exceed the final temperature of the calorimeter. On the chosen instant the automatic switch is thrown, and the current stopped in the calorimeter heater. As quickly as possible the bath temperature control is shifted to the thermoregulator and a few minutes are allowed for the establishment of a steady state. During this interim, readings of the differential thermoelements are continued. Within 10 seconds after the power is turned off, the calorimeter temperature has become uniform.

When the bath temperature has become steady, the temperature of the calorimeter is observed as previously described. During the interim the power now on the spill is readjusted to take care of the change in the heat capacity of the calorimeter and its contents over the next 10-deg. interval. This makes it possible to stop at very nearly 20 deg. with an integral number of minutes of heating.

The final temperature of the first interval is the initial temperature of the second interval. The experiments are continued in steps of 10 deg. as far as desired.

For temperatures of 60 deg. and above, no refrigeration is required for the satisfactory regulation of the bath temperature.

3—Latent-Heat Determinations

The measurements of the second type which determine the value of the quantity $L + Lu/(u' - u)$, denoted by the symbol γ, were carried out at certain chosen temperatures at which the values of L, the heat of vaporization, were desired.

The process of evaporating and removing saturated vapor is a flow process, and in these experiments the instrument is used as a flow calorimeter with storage for the fluid sample. The experiments are carried out strictly as flow experiments proceeding at a steady state. It was possible to perform them as successive determinations, immediately following one another without disturbance of the steady condition. Sometimes as many as eight separate measurements were made in one series during a day's work.

The procedure for the measurement of latent heat is in general as follows:

A weighed charge of water is introduced into the calorimeter. Three containers are evacuated, conditioned, weighed, and attached to the three vapor lines. The lines up to the shut-off valves are then evacuated and the vacuum valves closed.

In the following discussion the three flow lines for vapor and the corresponding shut-off valves will be designated as Nos. 1, 2, and 3. This numbering is convenient for the description of the manipulation, but has no further significance. In the preliminary operations to establish the steady state the receiver in position 1 is used, and it will be spoken of as the "spill." When in use, each of the tempering coils at the top is immersed in boiling water, and each of the receivers in cold running water.

During the preliminary heating of the calorimeter and bath, the shut-off valve and the receiver valve in the spill line are open. This prevents any leak past the throttle from causing pressure to build up on the diaphragms of the valves in the vapor lines. The calorimeter pump is running during the heating up, and its speed is adjusted to the desired value.

When the temperature chosen for operation is reached by both the calorimeter and the bath, and when the bath temperature is properly held there by the thermoregulator, the withdrawal of vapor can be begun.

The current for the calorimeter heater, previously adjusted to the value to give the desired rate of evaporation, is then turned on. The throttle is opened and gradually adjusted to the rate of flow which will maintain the calorimeter temperature constant and equal to the practically constant temperature of the envelope. The flow into the spill in line No. 1 is continued for about a half-hour while the operator completes the adjustment of the throttle to bring the calorimeter to a steady temperature. This much time at least was allowed to insure that any liquid which had lodged in the tubes could be swept out and a steady state of flow of dry saturated vapor established.

Meanwhile the automatic throw mechanism for the shut-off valves is adjusted so that No. 1 will close and No. 2 open at a chosen signal from the standard clock. The valve on the receiver in line No. 2 is opened to allow passage of vapor when shut off valve No. 2 opens.

Before starting the measurements, a survey of the temperature existing at various points in the system is made to verify the existence of the steady state.

Six temperature observations are taken at times ten, twenty, and thirty seconds before and after the transfer of the flow from line 1 to line 2. For these observations, simultaneous readings are taken of the resistance of a single thermometer and the e.m.f. of the five thermoelements on the calorimeter in series.

On the chosen minute the flow is shifted from line 1 to line 2. Observations of the temperatures of the calorimeter and reference block are made on each succeeding minute, the operator maintaining the calorimeter temperature as nearly constant as possible by adjustment of the throttle valve. Observations are made of current and potential drop in the calorimeter heater on the alternate minutes. Meanwhile determinations are made of the mass of water which has been removed, by weighing the spill and the residual water in the lines which has been caught in the liquid air trap. A weighed receiver is put in position 1 and the tempering coil evacuated. The valve mechanism is reset to be ready for the transfer of the flow from line 2 to line 3.

Water samples removed are usually about 30 grams. The time of an experiment is therefore determined by the rate at which the energy is supplied to the calorimeter. After the desired amount of water has been collected in position 2, temperatures are again measured as before.

The above-described procedure is repeated as long as time or the quantity of water in the calorimeter permits, each experiment constituting an independent determination.

4—Evaporation Factor

The α experiments yield values of the quantity $H - L\,\frac{u}{u' - u}$ and the γ experiments yield values of $L + L\,\frac{u}{u' - u}$. To determine the values of H and L from these quantities it is necessary to know the value of the quantity $L\,\frac{u}{u' - u}$, which has been designated as β.

In the analysis[12] it is shown that this correction term β may be determined calorimetrically as the amount of energy which is supplied per gram of saturated liquid withdrawn at a constant temperature.

[12] See reference to footnote 8.

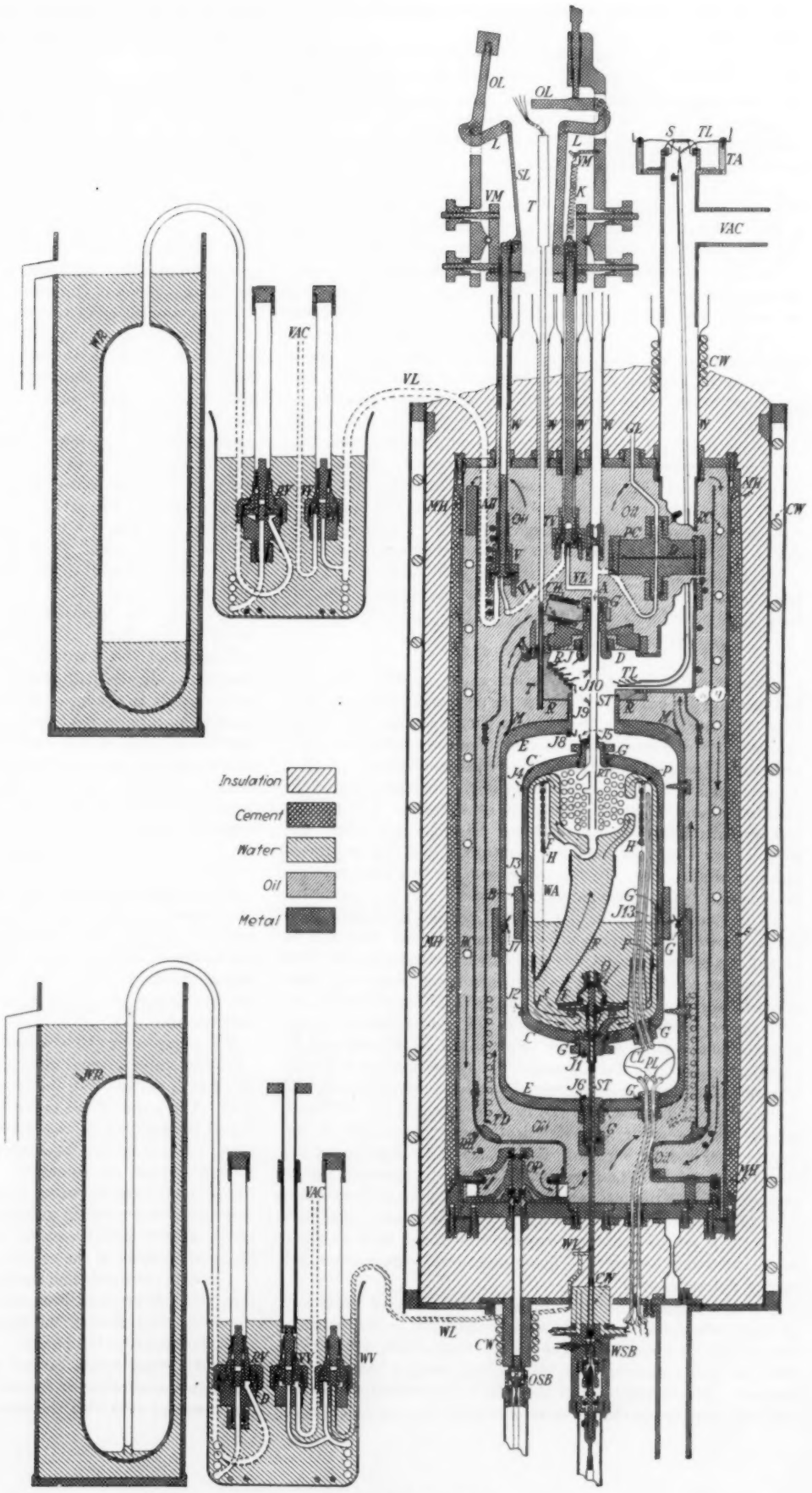


FIG. 11 SECTION OF ASSEMBLED CALORIMETER

(NOTE: This is not a true section, but shows sections of important parts projected on a plane.)

C, calorimeter shell (Cu-Ni)
E, envelope shell (Cu-Ni)
B, threaded band (monel)
G, gold gaskets
ST, support tubes (Ag-Pd) (Water and steam outlets)
I, pump impeller
P, pump casing (Ag)
O, ball bearings (Ir alloy)
P, water port
H, calorimeter heater (Ag)
WA, gauze apron (Ag)
CL, current lead (Au)
PL, potential lead (Au)
PC, pressure capsule (monel)
GL, gage line
WL, liquid water line (Ag)
WSB, water stuffing box
WV, water valve
WR, water receiver
RV, receiver valve
VAC, vacuum line
VV, vacuum valve
VL, vapor line
V, shut-off valve
TV, throttle valve
VM, valve mechanism
OL, operating lever
SL, spring lever
L, link
K, backlash spring
TD, thermoregulator dilatometer
R, reference block (Cu)
D, diaphragm
M, mantle
J1, *J2*, . . ., principal junctions
RJ, reference junctions
TL, thermoelement leads (Au)
S, gastight insulating seal
TA, anchorage for *TL* leads (Cu)
T, Pt resistance thermometer
OIL, oil bath
OP, oil pump
OSB, oil stuffing box
MH, main heater
AH, auxiliary heater
RH, regulating heater
CH, central heater
RC, refrigerating coil
CW, cooling water

(One-quarter actual size.)

The procedure for a determination of β at a given temperature is as follows:

A weighed charge of water is introduced into the calorimeter. The temperatures of the calorimeter and the bath are brought to the value chosen for operation. The electric power on the spill is adjusted so that the proper rate of withdrawal of water will approximate a chosen value. The pump speed is adjusted and controlled as in the α and γ experiments.

A weighed container is attached to the lower tempering coil and the coil evacuated. The shut-off valve in the liquid line is now of the throttling type previously described, with arrangement for control by the operator at the instrument table.

The temperature of the calorimeter is measured as in the α experiments, and the power is similarly thrown on. As quickly as possible the operator opens the throttle for the liquid and adjusts it until the temperature of the calorimeter is constant and practically equal to the bath temperature. As in the α experiments, readings of the differential thermoelements and the power to the heater are recorded at the successive one-minute intervals. At the chosen instant for stopping the experiment the power is cut off. The throttle is closed at such a time that the final temperature of the calorimeter is nearly the same as that of the bath. If the calorimeter is a little too hot, more liquid can be withdrawn until the desired state is reached. The final temperature is then measured as before. The water withdrawn is collected and weighed in the manner already described.

No provision was made for making more than a single β experiment at a time, and therefore it was necessary to measure only the time of electric heating and not the time of flow.

5—Thermal Leakage

When a difference of temperature exists between the calorimeter and its envelope, there is an accompanying transfer of heat by radiation, conduction, and convection. For small temperature differences the amount of energy transferred per unit time is proportional to this temperature difference.

In operation it is impossible to keep the temperatures of the calorimeter and its envelope identical at all times. The existing temperature difference can, however, be kept very small, and over a period of time can be so controlled that the integrated value is almost zero. In the measurements this ideal was approached as the operator gained experience.

To properly correct for the thermal leakage in those experiments where the temperature difference did not integrate to zero, a set of experiments was performed to determine the thermal-leakage coefficient. These experiments are performed under temperature conditions which are identical, as far as is possible, with those prevailing during the α, β, and γ experiments, except that the temperature difference between the calorimeter and its envelope is purposely much greater. The total amount of energy transferred under this exaggerated temperature difference, the value of the difference, and the time over which the difference exists, are observed. No electrical energy is added to the calorimeter, and during most of the time of an experiment the pump is stopped. Thus practically all of the observed energy change is due to thermal leakage. Because the calorimeter was used both with the envelope space evacuated and filled with nitrogen, a set of heat-leak experiments under each of these conditions was necessary. These heat-leak experiments are carried out as follows:

The calorimeter is charged with about 250 grams of water and its temperature and that of the bath are brought to the temperature chosen for operation. The initial temperature is observed with the pump running, and on the next minute the pump is stopped. The temperature of the bath is then raised or lowered until the envelope is either hotter or colder than the calorimeter by an amount corresponding to about 100 microvolts ($^2/_3$ deg.) on the differential or regulating thermoelements. At one-minute intervals readings of this indication are recorded in microvolts over a period of from 30 minutes to one hour. At the end of this time the bath temperature is brought back to that of the calorimeter, and on an integral minute the pump is started to insure uniformity of temperature throughout the calorimeter. The final temperature is then observed as before.

The pump is stopped during most of the time of the experiment, so that the uncertainty in the determination of the energy supplied by the pump does not enter into the determination of the thermal-leakage rate. Small corrections are made for the energy supplied by the pump during the short intervals at the beginning and end of the experiment.

The data obtained in such an experiment, when combined with the previously determined heat capacity of the calorimeter and its contents, yield the amount of heat transferred per microvolt-minute at the temperature of the experiment. In accordance with Newton's law of cooling, this coefficient or thermal-leakage rate can be used as a factor which, when multiplied by the resultant number of microvolt-minutes for any experiment at that temperature, gives directly the net heat exchange due to heat leak. At the higher temperatures where a small difference in temperature corresponds to a much larger difference in vapor pressure, the bath was kept colder than the calorimeter in the heat-leak experiments. This insured against the direct transfer of heat from the bath to the calorimeter by active boiling in the lower support tube.

The heat-leak experiments, on account of the method in which they are performed, furnish an opportunity for the direct calibration of the thermoelements against the resistance thermometers. If the bath temperature is measured both before and after it is changed, simultaneous indications of the five thermoelements on the calorimeter in series can be interpreted directly in terms of the thermometers, since the change in the calorimeter temperature in the short interval of time involved is small. Only the five series thermoelements were so calibrated, because the indications of the others need not be expressed in degrees.

6—Pump Energy

The energy supplied to the calorimeter by the pump appears as a correction which must be applied to obtain the total energy added in the α, β, and γ experiments. As previously shown, the knowledge of the true value of this pump energy is not of vital importance in the determinations of α, but it appears directly in the evaluation of β and γ, and must therefore be carefully determined.

The pump-energy determinations are carried out as follows:

The calorimeter, containing a known mass of water, is brought, along with the bath, to the temperature chosen for the observation. The pump speed is regulated to the desired value. The initial temperature of the calorimeter is measured at a chosen time. At one-minute intervals the indications of the differential thermoelements are recorded. The bath temperature is raised from time to time to reduce the integrated value of the temperature difference between calorimeter and envelope to virtually zero. After a period varying from 30 minutes to one hour the final temperature of the calorimeter is observed.

The data obtained from such an experiment, together with the previously determined heat capacity of the calorimeter and its contents, yield directly the rate at which energy is supplied to the calorimeter by the pump.

Similar experiments carried out with different quantities of water in the calorimeter showed no systematic variation in the pump-energy rate with the mass of the water.

Experiments were performed with the pump running at various speeds, but throughout most of the work the speed employed was about 900 r.p.m.

VI—RESULTS OF MEASUREMENTS

1—Alpha Experiments

In the following record of results of measurements, the experimental data have been assembled in the form of tables. An example of the record of a single determination of heat capacity of the calorimeter and its charge of water, over a ten-degree interval, is given in Table 1. The total time of experiment,

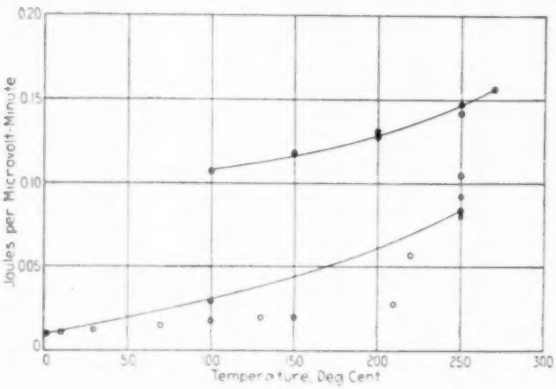


Fig. 12 Heat-Leak Coefficient

(Lower curve with space evacuated; upper curve with nitrogen in space.)

over which the heat-leak correction and the pump energy correction are applied, is the interval between the middle instants of the initial and final periods of temperature readings. The total heat-leak factor for this interval is the sum of the heat-leak indications at one-minute periods during this time. This factor in microvolt-minutes multiplied by the heat-leak coefficient in joules per microvolt-minute gives the heat-leak correction in joules. Numerous determinations of the heat-leak coefficient were made and the values observed have been assembled in the form of a chart, Fig. 12.

The correction for pump energy is computed as the product of the total time of the experiment by the pump power. The values of the pump power taken were those determined from time to time in the special blank experiments for obtaining this quantity. Nearly 200 pump-energy determinations were made throughout the experimental work, and the values observed have been assembled in the form of a chart, Fig. 13. In computing the results of the work on heat capacity the values for pump power have been modified at various times in the light of additional determinations. It has not been necessary to revise the earlier calculations on the basis of a uniform value for pump power, because this factor has so little effect on the final result.

A partial reduction of the observations is given in Table 2. This includes the computation of the initial and final tempera-

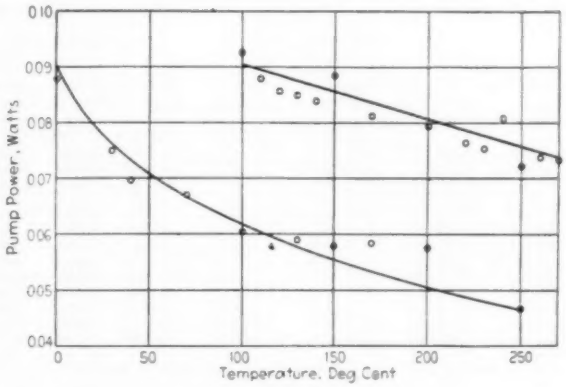


Fig. 13 Pump Power

(Lower curve with tachometer at 51 cm.; upper curve with tachometer at 70 cm.)

TABLE 1 SAMPLE DATA SHEET—DETERMINATION OF HEAT CAPACITY

Experiment No. 193-U. Jan. 26, 1929
200 to 210 Deg. Cent.
Observers: N. S. O., H. F. S., E. F. F.

	—Thermometers—		—Thermoelements—		—Electric Power—		
Time	Designation	Bridge ohms	t.e. 1–5 —microvolts—	Regulating	I amperes	E volts	
12:21:00	2 n	45.2890	9.0	2	....	....	
:20	2 r	45.2939	8.8	...	....	....	
:40	3 r	45.4794	8.8	...	....	....	
:22:00	3 n	45.4735	8.5	...	....	....	
:20	4 n	45.8176	8.6	...	....	....	
:40	4 r	45.8143	8.7	...	....	....	
:23	...	.....	...	3	....	....	On
:24	...	.....	...	0	1.9929	....	
:25	...	.....	...	2	....	20.195	
:26	...	.....	...	2	1.9924		
:27	...	.....	...	1	....	20.195	
:28	...	.....	...	—2	1.9920	....	
:29	...	.....	...	2	....	20.193	
:30	...	.....	...	—5	1.9916	....	
:31	...	.....	...	—3	....	20.192	
:32	...	.....	...	—1	1.9912	....	
:33	...	.....	...	0	....	20.193	
:34	...	.....	...	—3	1.9909	....	
:35	...	.....	...	—2	....	20.192	Off
:36	...	.....	...	3	....	....	
:37	...	.....	...	2	....	....	
:38	...	.....	...	2	....	....	
:39:00	2 n	46.2525	7.1	...	....	....	
:20	2 r	46.2581	7.2	...	....	....	
:40	3 r	46.4470	7.2	...	....	....	
:40:00	3 n	46.4402	7.1	...	....	....	
:20	4 n	46.7902	6.8	...	....	....	
:40	4 r	46.7858	6.8	...	...	....	

Total heat-leak factor = +5 microvolt-minutes
Total time of experiment, 12:22 to 12:40, = 1080 seconds
Total time of electric heating, 12:23 to 12:35, = 720 seconds

TABLE 2 SAMPLE COMPUTATION—DETERMINATION OF HEAT CAPACITY

Experiment No. 193-U 200 to 210 Deg. Cent.

	Initial			Final		
Thermometer No.	2	3	4	2	3	4
$(n + r)/2$	45.2914	45.4764	45.8159	46.2553	46.4436	46.7880
Bridge correction	0.0096	0.0096	0.0096	0.0096	0.0096	0.0096
R_θ	45.3010	45.4860	45.8255	46.2649	46.4532	46.7976
$R_\theta - R_0$	19.6930	19.7575	19.8716	20.6569	20.7247	20.8437
$100(R_\theta - R_0)/(R_{100} - R_0)$	197.090	197.095	197.064	206.737	206.744	206.704
$0.01\theta(0.01\theta - 1)\delta$	2.961	2.965	2.997	3.425	3.429	3.466
Ref. block temp.	200.051	200.060	200.061	210.162	210.173	210.170
Mean ref. block temp.		200.057			210.168	
Mean t.e. 1–5 (microvolts)	8.73			7.03		
Mean t.e. 1–5 (degrees)		—0.030			—0.024	
Temperature of calorimeter		200.027			210.144	

Calculation of Energy

Potentiometer	Current		Voltage
Mean reading	1.99183	Mean reading	20.1933
Pot. correction	0.00020	Pot correction	0.0020
Correction for std. 0.01 ohm	—0.00010	Corr. for volt box factor	0.0067
Corr. for volt box current	—0.00060		
Current I =	1.99133 amperes	Voltage E =	20.2020 volts

Time of electric heating = t = 720 seconds
Total electric energy = IEt = 28,964.8 joules

Total time of experiment	1080	seconds	
Pump power	0.0759	watts	
Pump energy			82.0
Total heat-leak factor	5	microvolt-minutes	
Heat-leak coefficient	0.129	joules/microvolt-minute	
Heat leak			0.6
Entire energy added			29,047.4 joules

TABLE 3 PRINCIPAL DATA FROM α EXPERIMENTS

$$Q]_1^2 = M\alpha]_1^2 + Z]_1^2. \quad \alpha]_1^2 = \left[H - L\frac{\theta}{\theta' - \theta}\right]_1^2$$

Exp. No.	Date	Mass of water M Grams	Initial temp. θ_1 Degrees centigrade	Final temp. θ_2	Total time of exp. Seconds	Time of electric heating	Total electric energy International joules	Pump energy	Heat leak	Entire energy added Q	Energy for even temp. intervals $Q]_1^2$	$\alpha]_1^2$ International joules per gram	Residual $\alpha]_1^2 - \alpha_m$
35-A	6/10/27	487.107	1.402	10.285	1320	540	25,427.3	116.4	2.6	25,546.2	28,769.8	41.982	−0.001
36-A	6/13/27	487.107	1.546	10.380	1440	540	25,277.6	127.0	1.9	25,406.5	28,772.8	41.988	0.005
37-A	6/16/27	487.107	1.504	10.273	2580	1020	24,981.5	228.0	3.2	25,212.8	28,763.2	41.968	−0.015
42-A	7/11/27	326.833	1.627	8.987	1380	480	16,091.9	120.6	1.6	16,214.1	22,032.0	41.953	−0.030
45-A	7/14/27	326.833	2.508	9.209	1380	420	14,645.2	119.8	1.5	14,766.5	22,039.0	41.975	−0.008
46-A	7/16/27	326.833	1.323	9.265	1380	480	17,374.3	120.0	1.6	17,495.7	22,033.8	41.959	−0.024
49-A	7/22/27	493.054	1.602	9.818	1320	480	23,727.8	116.1	1.3	23,845.2	29,033.6	42.010	0.027
51-A	7/25/27	493.054	1.716	10.260	1440	480	24,661.7	125.0	1.9	24,788.6	29,026.6	41.996	0.012
57-A	8/ 2/27	291.001	1.196	9.499	2820	1080	16,818.1	246.7	2.1	17,066.9	20,555.7	42.046	0.063
58-A	8/ 3/27	291.001	1.021	9.802	2760	1080	17,795.3	240.2	2.5	18,038.1	20,543.6	42.005	0.022
59-A	8/ 4/27	291.001	0.908	9.880	2580	1080	18,204.3	225.4	3.5	18,433.2	20,546.7	42.015	0.032
62-A	8/ 9/27	366.801	0.552	10.100	3420	1260[a]	22,340.6	297.7	2.3	22,640.6	23,710.6	41.958	−0.025
63-A	8/10/27	366.801	1.029	10.083	2340	1080[a]	21,256.0	203.3	1.6	21,460.9	23,702.8	41.937	−0.046
	Even temperature interval 0–10°. $Z]_0^{10} = 8320.3$ int. joules. Mean value of $\alpha]_0^{10} = \alpha_m = 41.9832$ int. joules per gram												
35-B	6/10/27	487.107	10.284	20.210	1500	598	28,394.3	120.5	3.8	28,518.6	28,731.0	41.799	−0.003
36-B	6/13/27	487.107	10.380	20.447	1440	600	28,808.9	115.7	1.8	28,926.3	28,732.6	41.802	0.000
37-B	6/16/27	487.107	10.273	20.192	2760	1200	28,273.1	220.6	4.6	28,498.2	28,732.2	41.802	0.000
38-B	6/17/27	487.107	9.955	19.653	2940	1200	27,628.9	236.2	3.3	27,868.5	28,736.1	41.810	0.008
39-B	6/20/27	487.107	9.995	19.824	3240	1200	27,981.0	260.8	3.8	28,245.5	28,736.7	41.811	0.009
42-B	7/11/27	326.833	9.136	20.203	1740	660	24,235.8	140.2	1.6	24,377.6	22,025.8	41.781	−0.021
46-B	7/16/27	326.833	9.265	19.212	1440	600	21,788.4	116.0	1.6	21,906.0	22,024.7	41.778	−0 024
49-B	7/22/27	493.054	9.817	20.445	1560	660	30,677.7	124.4	1.7	30,803.8	28,979.0	41.798	−0.004
51-B	7/25/27	493.054	10.259	19.688	1620	600	27,193.5	127.9	3.1	27,324.5	28,980.9	41.802	0.000
57-B	8/ 2/27	291.001	9.500	19.147	1500	600	19,691.1	120.2	1.3	19,812.6	20,538.7	41.816	0.014
58-B	8/ 3/27	291.001	9.802	19.923	1920	600	20,637.9	153.9	1.0	20,792.8	20,542.2	41.828	0.026
59-B	8/ 4/27	291.001	9.879	20.015	1440	600	20,703.7	113.8	1.7	20,819.2	20,539.5	41.818	0.016
62-B	8/ 9/27	366.801	10.100	20.625	1620	780[a]	24,818.4	129.0	1.0	24,948.4	23,699.9	41.793	−0.009
63-B	8/10/27	366.801	10.083	20.001	1200	600[a]	23,409.1	95.2	1.4	23,505.7	23,699.3	41.791	−0.011
	Even temperature interval 10–20°. $Z]_{10}^{20} = 8370.5$ int. joules. Mean value of $\alpha]_{10}^{20} = \alpha_m = 41.8016$ int. joules per gram												
35-C	6/10/27	487.107	20.210	30.208	1620	600	28,626.1	120.4	3.4	28,749.9	28,754.2	41.713	0.005
36-C	6/13/27	487.107	20.447	30.364	1380	600	28,407.4	102.6	2.0	28,511.9	28,749.9	41.704	−0.004
37-C	6/16/27	487.107	20.191	30.142	1500	600	28,493.8	111.2	2.7	28,607.9	28,748.8	41.702	−0.006
43-C	7/12/27	326.833	20.034	29.581	1500	600	20,948.8	112.0	2.2	21,062.9	22,061.9	41.692	−0.016
46-C	7/16/27	326.833	19.212	29.458	1740	600	22,464.4	130.0	3.8	22,598.2	22,057.9	41.680	−0.028
49-C	7/22/27	493.054	20.445	30.088	1440	600	27,859.5	105.7	2.6	27,967.8	29,005.8	41.720	0.012
51-C	7/25/27	493.054	19.688	29.507	1380	600	28,368.2	101.7	2.8	28,472.7	28,998.2	41.705	−0.003
53-C	7/27/27	493.054	19.809	29.487	1800	600	27,929.1	131.9	3.2	28,064.2	29,002.7	41.714	0.006
57-C	8/ 2/27	291.001	19.147	29.899	1380	660	22,014.6	103.0	1.4	22,119.0	20,572.5	41.707	−0.001
58-C	8/ 3/27	291.001	19.923	29.984	1380	600	20,597.4	102.6	2.1	20,702.1	20,576.8	41.722	0.014
59-C	8/ 4/27	291.001	20.015	30.114	1440	600	20,680.3	106.5	2.0	20,788.8	20,583.8	41.746	0.038
62-C	8/ 9/27	366.801	20.625	28.802	1380	540[a]	19,299.1	102.8	1.4	19,403.3	23,732.4	41.703	−0.005
63-C	8/10/27	366.801	20.001	29.889	1200	600[a]	23,376.3	88.9	1.4	23,466.6	23,731.4	41.700	−0.008
	Even temperature interval 20–30°. $Z]_{20}^{30} = 8435.7$ int. joules. Mean value of $\alpha]_{20}^{30} = \alpha_m = 41.7078$ int. joules per gram												
35-D	6/10/27	487.107	30.209	40.169	1380	600	28,580.2	97.3	3.1	28,680.6	28,793.7	41.703	0.000
36-D	6/13/27	487.107	30.364	40.268	1560	600	28,401.2	110.0	3.8	28,515.0	28,791.7	41.699	−0.004
37-D	6/16/27	487.107	30.142	40.118	1560	600	28,613.4	110.0	3.0	28,726.4	28,794.6	41.705	0.002
43-D	7/12/27	326.833	29.850	39.633	1560	600	21,507.6	109.7	1.7	21,619.0	22,100.6	41.675	−0.028
46-D	7/16/27	326.833	29.458	39.576	1380	600	22,263.8	96.5	1.4	22,361.7	22,103.6	41.684	−0.019
49-D	7/22/27	493.054	30.089	39.914	1380	600	28,440.1	96.8	2.3	28,539.2	29,047.4	41.715	0.012
51-D	7/25/27	493.054	29.507	39.689	1380	600	29,470.7	97.0	2.5	29,570.2	29,043.4	41.706	0.003
53-D	7/27/27	493.054	29.487	39.523	1500	600	29,037.4	105.5	3.4	29,146.3	29,044.9	41.709	0 006
57-D	8/ 2/27	291.001	29.899	40.553	1380	600	21,868.9	96.7	1.7	21,967.3	20,616.8	41.707	0.004
58-D	8/ 3/27	291.001	29.984	40.000	1140	600	20,571.9	79.9	1.9	20,653.7	20,619.7	41.717	0.014
59-D	8/ 4/27	291.001	30.114	40.002	1200	600	20,316.1	83.8	1.3	20,401.2	20,632.5	41.761	0.058
62-D	8/ 9/27	366.801	28.802	40.356	1380	780[a]	27,368.1	96.7	1.6	27,466.4	23,774.0	41.696	−0.007
63-D	8/10/27	366.801	29.889	39.922	1140	600[a]	23,772.2	79.8	1.6	23,853.6	23,765.2	41.672	−0.031
	Even temperature interval 30–40°. $Z]_{30}^{40} = 8480.0$ int. joules. Mean value of $\alpha]_{30}^{40} = \alpha_m = 41.7032$ in. joules per gram												
35-E	6/10/27	487.107	40.169	50.095	1320	600	28,541.7	89.9	2.5	28,634.1	28,846.0	41.708	0.010
36-E	6/13/27	487.107	40.268	50.136	1320	600	28,374.2	89.9	2.5	28,466.5	28,844.4	41.705	0.007
37-E	6/16/27	487.107	40.118	50.072	1380	600	28,607.7	93.9	2.8	28,704.4	28,837.7	41.691	−0.007
43-E	7/12/27	326.833	39.633	49.965	1560	600	22,782.0	104.4	3.0	22,889.4	22,153.5	41.685	−0.013
49-E	7/22/27	493.054	39.914	49.861	1440	600	28,830.6	97.3	1.7	28,929.6	29,084.1	41.688	−0.010
51-E	7/25/27	493.054	39.689	50.055	1440	600	30,055.0	97.5	2.2	30,154.7	29,093.6	41.707	0.009
53-E	7/27/27	493.054	40.084	50.072	1380	600	28,958.3	93.5	2.1	29,053.9	29,089.2	41.699	0.001
57-E	8/ 2/27	291.001	40.553	50.097	1260	540	19,642.6	84.5	2.6	19,729.1	20,667.0	41.709	0.011
58-E	8/ 3/27	291.001	40.001	50.159	1140	600	20,910.0	76.4	1.9	20,988.3	20,660.6	41.687	−0.011
59-E	8/ 4/27	291.001	40.002	49.867	1440	600	20,296.6	96.1	2.1	20,394.8	20,673.2	41.730	0.032
62-E	8/ 9/27	366.801	40.356	50.450	1200	660[a]	23,932.6	80.1	1.0	24,013.7	23,821.9	41.691	−0.007
63-E	8/10/27	366.801	39.922	50.111	1200	600[a]	24,186.9	80.1	2.1	24,269.1	23,818.3	41.681	−0.017
	Even temperature interval 40–50°. $Z]_{40}^{50} = 8529.7$ int. joules. Mean value of $\alpha]_{40}^{50} = \alpha_m = 41.6981$ int. joules per gram												
35-F	6/10/27	487.107	50.095	59.991	1440	600	28,503.5	95.1	3.6	28,602.3	28,903.7	41.685	0.002
36-F	6/13/27	487.107	50.136	60.052	1560	600	28,549.3	103.1	4.5	28,656.9	28,900.0	41.677	−0.006
37-F	6/16/27	487.107	50.072	60.070	1440	600	28,804.7	95.0	3.6	28,903.3	28,909.3	41.696	0.013
44-F	7/13/27	326.833	49.962	60.055	1440	600	22,322.4	93.7	2.0	22,418.1	22,210.1	41.647	−0.036
49-F	7/22/27	493.054	49.881	59.995	1500	600	29,446.5	98.7	2.7	29,547.9	29,158.3	41.695	0.012
51-F	7/25/27	493.054	50.055	60.173	1500	600	29,391.5	98.7	3.1	29,493.3	29,147.2	41.676	−0.007
53-F	7/27/27	493.054	50.072	59.908	1380	600	28,574.7	90.7	3.4	28,668.8	29,148.7	41.679	−0.004
57-F	8/ 2/27	291.001	50.097	60.100	1380	600	20,647.1	88.8	1.9	20,737.8	20,740.0	41.722	0.039
58-F	8/ 3/27	291.001	50.159	60.096	1200	600	20,521 2	77.2	2.8	20,601.2	20,727.9	41.681	−0.002
59-F	8/ 4/27	291.001	49.867	59.855	1380	600	20,618.1	88.6	2.8	20,709.5	20,734.2	41.703	0.020
62-F	8/ 9/27	366.801	50.450	59.584	1320	600[a]	21,731.3	84.9	1.0	21,817.2	23,884.6	41.673	−0.010
63-F	8/10/27	366.801	50.111	60.264	1500	600[a]	24,150.1	96.2	1.1	24,247.4	23,880.7	41.663	−0.020
	Even temperature interval 50–60°. $Z]_{50}^{60} = 8598.6$ int. joules. Mean value of $\alpha]_{50}^{60} = \alpha_m = 41.6833$ int. joules per gram												
35-G	6/10/27	487.107	59.991	69.962	1260	600	28,798.7	81.0	2.9	28,882.6	28,967.7	41.690	0.007
36-G	6/13/27	487.107	60.052	69.997	1380	600	28,714.6	88.7	3.8	28,807.2	28,965.1	41.685	0.002
37-G	6/16/27	487.107	60.070	70.096	1680	600	28,921.8	107.8	4.2	29,033.9	28,958.9	41.672	−0.011
44-G	7/13/27	326.833	60.055	70.368	1560	600	22,887.6	97.8	2.2	22,987.6	22,278.1	41.666	−0.017
49-G	7/22/27	493.054	59.995	70.327	1380	600	30,094.3	88.3	1.7	30.134.3	29,214.1	41.687	0.004
50-G	7/23/27	493.054	59.997	70.202	1500	600	29.714.9	96.0	2.6	29,813.5	29,212.5	41.685	0.002
51-G	7/25/27	493.054	60.173	70.093	1440	600	28,888.6	90.8	2.2	28,981.6	29,212.3	41.685	0.002
52-G	7/26/27	493.054	60.156	70.048	2040	600	28,768.7	130.1	2.4	28,901.2	29,215.7	41.690	0.007
53-G	7/27/27	493.054	59.908	69.821	1350	600	28,869.6	86.4	2.0	28,958.0	29,209.0	41.677	−0.006

[a] Time determined with a sidereal chronometer and must be multiplied by 0.99735 to obtain mean solar seconds.

TABLE 3 PRINCIPAL DATA FROM α EXPERIMENTS—*Continued*

Exp. No.	Date	Mass of water M	Initial temp. θ_1	Final temp. θ_2	Total time of exp.	Time of electric heating	Total electric energy	Pump energy	Heat leak	Entire energy added Q	Energy for even temp. intervals $Q]_1^2$	$\alpha]_1^2$	Residual $\alpha]_1^2 - \alpha_m$
57-G	8/ 2/27	291.001	60.100	70.061	1320	600	20,629.8	82.9	2.2	20,714.9	20,787.0	41.673	−0.010
58-G	8/ 3/27	291.001	60.096	69.990	1140	600	20,498.6	71.5	1.6	20,571.7	20,791.6	41.689	0.006
59-G	8/ 4/27	291.001	59.855	70.135	1260	600	21,299.3	78.9	1.5	21,379.7	20,798.2	41.711	0.028
62-G	8/ 9/27	366.801	59.584	69.592	1380	660[1]	23,875.1	86.4	2.0	23,963.5	23,947.5	41.678	−0.005
63-G	8/10/27	366.801	60.264	70.195	1140	600[1]	23,711.2	71.5	1.3	23,949.7	23,947.4	41.677	−0.006
	Even temperature interval 60–70°. $Z]_{60}^{70}$ = 8660.2 int. joules. Mean value of $\alpha]_{60}^{70} = \alpha_m$ = 41.6828 int. joules per gram												
35-H	6/10/27	487.107	69.962	80.078	1320	600	29,281.6	82.6	3.2	29,367.4	29,030.1	41.677	−0.008
36-H	6/13/27	487.107	69.997	80.439	1560	600	29,071.5	97.6	4.2	29,173.3	29,037.9	41.693	0.008
37-H	6/16/27	487.107	70.096	80.201	1440	600	29,248.8	90.0	3.2	29,342.0	29,034.0	41.685	0.000
44-H	7/13/27	326.833	70.474	80.725	1500	600	22,816.3	91.5	2.5	22,910.2	22,342.4	41.653	−0.032
49-H	7/22/27	493.054	70.327	80.621	1440	600	30,059.6	89.6	4.3	30,153.5	29,285.6	41.693	0.008
50-H	7/23/27	493.054	70.202	80.363	1440	600	29,665.7	89.8	1.9	29,757.4	29,280.5	41.683	−0.002
52-H	7/26/27	493.054	70.048	79.885	1440	600	28,717.5	89.7	2.0	28,809.2	29,281.0	41.684	−0.001
53-H	7/27/27	493.054	69.821	80.169	1350	600	30,216.3	84.3	1.8	30,302.4	29,282.3	41.686	0.001
57-H	8/ 2/27	291.001	70.061	79.982	1380	600	20,610.7	84.4	1.9	20,697.0	20,861.5	41.693	0.008
58-H	8/ 3/27	291.001	69.990	80.016	1260	600	20,834.2	77.0	1.7	20,912.9	20,858.0	41.681	−0.004
59-H	8/ 4/27	291.001	70.135	80.218	1440	600	20,955.7	87.9	3.1	21,046.7	20,869.7	41.721	0.036
62-H	8/ 9/27	366.801	69.592	80.268	1260	660[2]	25,561.6	77.0	1.4	25,640.0	24,016.4	41.678	−0.007
63-H	8/10/27	366.801	70.195	80.087	1140	600[2]	23,686.3	69.6	1.1	23,757.0	24,015.3	41.675	−0.010
	Even temperature interval 70–80°. $Z]_{70}^{80}$ = 8728.8 int. joules. Mean value of $\alpha]_{70}^{80} = \alpha_m$ = 41.6847 int. joules per gram												
35-I	6/10/27	487.107	80.078	90.269	1380	600	29,580.7	84.2	3.2	29,668.2	29,111.9	41.673	0.007
36-I	6/13/27	487.107	80.043	89.743	1440	600	29,297.4	87.9	4.2	29,389.5	29,110.4	41.670	0.004
37-I	6/16/27	487.107	80.201	90.333	1500	600	29,402.7	91.5	2.9	29,497.3	29,111.7	41.673	0.007
44-I	7/13/27	326.833	80.725	90.341	1680	600	21,466.1	109.8	1.9	21,568.8	22,420.2	41.635	−0.031
50-I	7/23/27	493.054	80.363	90.484	1440	600	29,623.2	87.5	3.1	29,713.8	29,350.0	41.654	−0.012
52-I	7/26/27	493.054	79.885	89.879	1500	600	29,247.9	91.2	2.1	29,341.2	29,356.5	41.667	0.001
53-I	7/27/27	493.054	80.169	90.355	1320	600	29,825.0	80.3	2.6	29,907.9	29,357.5	41.669	0.003
56-I	8/ 1/27	291.001	79.908	89.551	1440	600	20,098.1	86.6	2.3	20,187.0	20,936.7	41.671	0.005
57-I	8/ 2/27	291.001	79.981	90.006	1380	600	20,910.8	83.1	1.3	20,995.2	20,941.0	41.678	0.012
58-I	8/ 3/27	291.001	80.016	89.981	1140	600	20,793.9	68.6	1.1	20,863.6	20,934.6	41.656	−0.010
59-I	8/ 4/27	291.001	80.218	90.242	1140	600	20,931.7	68.6	2.0	21,002.3	20,948.7	41.705	0.039
62-I	8/ 9/27	366.801	80.268	89.928	1140	600[2]	23,206.5	68.6	1.2	23,276.3	24,093.9	41.661	−0.005
63-I	8/10/27	366.801	80.086	89.938	1080	600[2]	23,665.6	64.9	1.4	23,731.9	24,088.1	41.645	−0.021
	Even temperature interval 80–90°. $Z]_{80}^{90}$ = 8812.6 int. joules. Mean value of $\alpha]_{80}^{90} = \alpha_m$ = 41.6656 int. joules per gram												
35-J	6/10/27	487.107	90.269	100.480	1560	600	29,722.6	93.0	4.2	29,819.9	29,198.6	41.689	−0.001
36-J	6/13/27	487.107	90.139	100.239	1800	600	29,388.1	107.4	4.1	29,499.6	29,205.7	41.704	0.014
37-J	6/16/27	487.107	90.333	100.495	1440	600	29,582.9	85.4	3.1	29,671.4	29,195.1	41.682	−0.008
44-J	7/13/27	326.833	90.341	100.073	1440	600	21,814.7	85.2	2.1	21,901.9	22,500.2	41.638	−0.052
50-J	7/23/27	493.054	90.484	100.361	1500	600	28,999.0	89.0	3.5	29,091.5	29,450.9	41.698	0.008
52-J	7/26/27	493.054	89.879	100.022	1440	600	29,781.8	85.7	2.0	29,869.5	29,447.4	41.691	0.001
53-J	7/27/27	493.054	90.355	100.212	1380	600	28,943.5	82.1	2.2	29,027.8	29,444.5	41.685	−0.005
56-J	8/ 1/27	291.001	89.551	100.097	1440	660	22,090.6	85.4	3.4	22,179.4	21,030.2	41.714	0.024
57-J	8/ 2/27	291.001	90.007	100.164	1320	600	21,279.8	78.1	1.8	21,359.7	21,029.1	41.710	0.020
58-J	8/ 3/27	291.001	89.981	100.065	1340	600	21,118.0	79.2	0.8	21,198.0	21,023.0	41.689	−0.001
59-J	8/ 4/27	291.001	90.242	100.217	1170	600	20,910.2	69.3	1.0	20,980.5	21,028.8	41.709	0.019
62-J	8/ 9/27	366.801	89.928	100.894	1200	660[2]	26,448.4	71.4	1.4	26,521.2	24,180.6	41.682	−0.008
63-J	8/10/27	366.801	89.992	99.976	1140	600[2]	24,071.3	67.9	1.5	24,140.7	24,178.3	41.676	−0.014
	Even temperature interval 90–100°. $Z]_{90}^{100}$ = 8891.5 int. joules. Mean value of $\alpha]_{90}^{100} = \alpha_m$ = 41.6896 int. joules per gram												
38-K	6/17/27	487.107	99.716	109.876	1920	600	29,644.7	112.2	3.4	29,760.3	29,292.5	41.686	0.008
39-K	6/20/27	487.107	99.606	109.837	1380	600	29,878.6	80.7	3.9	29,963.2	29,288.4	41.677	−0.001
52-K	7/26/27	493.054	100.126	110.223	1320	600	29,749.8	76.8	3.5	29,830.1	29,536.8	41.678	0.000
53-K	7/27/27	493.054	100.212	109.985	1440	600	28,775.3	83.7	2.0	28,861.0	29,534.2	41.673	−0.005
56-K	8/ 1/27	291.001	100.097	109.936	1440	600	20,694.7	84.7	2.2	20,781.6	21,123.2	41.704	0.026
60-K	8/ 5/27	291.001	100.049	110.218	1320	600	21,402.1	77.3	0.8	21,480.2	21,119.3	41.691	0.013
87-K	11/ 8/27	475.903	100.135	110.796	1800	1080	30,626.1	104.5	2.2	30,732.8	28,823.8	41.682	0.004
89-K	11/10/27	475.903	100.104	110.293	1800	1200	29,267.8	104.4	0.0	29,372.2	28,825.7	41.686	0.008
90-K	11/11/27	475.903	100.066	110.309	1740	1200	29,427.5	101.0	0.0	29,528.5	28,825.4	41.686	0.008
93-K	11/15/27	263.730	100.103	110.107	1740	1200	19,881.9	100.9	−0.5	19,982.3	19,973.2	41.656	−0.022
94-K	11/16/27	263.730	100.030	110.044	1740	1200	19,911.6	100.9	−1.1	20,011.4	19,984.0	41.697	0.019
95-K	11/25/27	267.764	100.217	110.046	1740	1200	19,700.8	101.0	−0.4	19,801.4	20,144.5	41.669	−0.009
96-K	11/26/27	267.764	100.095	110.044	1800	1200	19,938.7	104.6	−2.0	20,041.3	20,144.5	41.669	−0.009
97-K	11/28/27	267.764	100.188	110.051	1680	1200	19,780.0	97.6	−2.4	19,875.2	20,150.3	41.690	0.012
112-K	1/30/28	301.182	100.197	110.164	1800	1200	21,366.3	103.5	2.1	21,471.9	21,542.1	41.686	0.008
113-K	1/31/28	410.025	100.032	110.331	1800	1260	26,747.7	109.8	0.4	26,857.9	26,077.0	41.680	0.002
114-K	2/ 1/28	410.025	99.974	110.452	1860	1260	27,208.5	103.5	1.3	27,313.3	26,066.3	41.654	−0.024
115-K	2/ 6/28	419.333	100.049	110.178	2100	1260	26,669.4	128.3	1.4	26,799.1	26,457.4	41.662	−0.016
	Even temperature interval 100–110°. $Z]_{100}^{110}$ = 8987.2 int. joules. Mean value of $\alpha]_{100}^{110} = \alpha_m$ = 41.6782 int. joules per gram												
38-L	6/17/27	487.107	109.876	120.052	1500	600	29,817.4	85.9	3.6	29,906.9	29,389.0	41.676	−0.008
39-L	6/20/27	487.107	109.837	120.106	1380	600	30,097.7	79.1	3.8	30,180.6	29,390.4	41.678	−0.006
52-L	7/26/27	493.054	110.223	120.277	1320	600	29,725.2	75.5	3.6	29,804.3	29,640.4	41.683	−0.001
56-L	8/ 1/27	291.001	109.936	119 903	1380	600	21,073.2	80.5	2.8	21,156.5	21,266.5	41.711	0.027
60-L	8/ 5/27	291.001	110.218	120.156	1380	600	21,013.1	80.4	2.1	21,095.6	21,224.1	41.703	0.019
87-L	11/ 8/27	475.903	110.796	120.192	1800	1080	27,081.0	102.8	2.1	27,185.9	28,927.0	41.686	0.002
89-L	11/10/27	475.903	110.293	120.441	1980	1200	29,247.3	113.1	0.5	29,360.9	28,929.7	41.691	0.007
90-L	11/11/27	475.903	110.309	120.338	1800	1200	28,912.4	102.7	0.5	29,015.6	28,928.8	41.690	0.006
93-L	11/15/27	263.730	110.107	120.073	2300	1200	19,867.5	131.3	−0.4	19,998.4	20,065.5	41.622	−0.062
95-L	11/25/27	267.764	110.046	119.826	1980	1200	19,692.9	113.6	−0.5	19,806.0	20,251.9	41.691	0.007
96-L	11/26/27	267.764	110.044	119.937	1740	1200	19,938.1	99.8	−2.5	20,035.4	20,249.9	41.684	0.000
97-L	11/28/27	267.764	110.051	119.867	1800	1200	19,775.7	103.2	−3.6	19,875.3	20,247.5	41.675	−0.009
112-L	1/30/28	301.182	110.164	119.951	1920	1200	21,073.4	114.7	1.8	21,189.9	21,650.2	41.708	0.024
113-L	1/31/28	410.025	110.331	119.623	1620	1140	24,232.0	96.8	1.4	24,330.2	26,183.7	41.691	0.007
114-L	2/ 1/28	410.025	110.452	120.030	1740	1140	24,967.8	103.9	0.8	25,072.5	26,173.2	41.667	−0.017
115-L	2/ 6/28	419.333	110.178	120.033	1740	1200	26,077.4	103.9	1.8	26,183.1	26,565.8	41.679	−0.005
	Even temperature interval 110–120°. $Z]_{110}^{120}$ = 9088.6 int. joules. Mean value of $\alpha]_{110}^{120} = \alpha_m$ = 41.6836 int. joules per gram												
38-M	6/17/27	487.107	120.053	130.284	2760	600	30,029.9	155.3	5.1	30,190.4	29,507.1	41.672	−0.005
39-M	6/20/27	487.107	120.106	130.389	1320	600	30,269.9	74.3	3.8	30,347.0	29,508.4	41.674	−0.003
52-M	7/26/27	493.054	120.277	130.077	1320	600	29,089.1	74.1	3.6	29,166.8	29,754.5	41.671	−0.006
56-M	8/ 1/27	291.001	119.903	130.342	1380	600	22,195.4	79.7	4.5	22,279.6	21,340.9	41.692	0.015
60-M	8/ 5/27	291.001	120.156	130.027	1380	600	20,985.4	79.8	2.7	21,067.9	21,340.1	41.689	0.012
87-M	11/ 8/27	475.903	120.192	130.062	1740	1140	28,571.1	98.2	1.9	28,671.2	29,046.4	41.685	0.008
89-M	11/10/27	475.903	120.440	130.388	1920	1200	28,792.1	108.2	−0.1	28,900.2	29,046.2	41.684	0.007
90-M	11/11/27	475.903	120.338	130.138	1740	1200	28,370.4	98.1	0.2	28,468.7	29,047.9	41.688	0.011
95-M	11/25/27	267.764	119.826	129.792	1800	1200	20,194.2	101.7	−1.2	20,294.7	20,365.8	41.668	−0.009
96-M	11/26/27	267.764	119.937	129.894	1680	1200	20,185.0	94.8	−2.9	20,276.9	20,364.9	41.665	−0.012
97-M	11/28/27	267.764	119.867	129.957	1620	1200	20,462.4	91.4	−3.0	20,550.8	20,369.0	41.680	0.003
112-M	1/30/28	301.182	119.951	129.945	1920	1200	21,634.5	112.0	1.0	21,747.5	21,760.8	41.676	−0.001
113-M	1/31/28	410.025	119.623	129.890	1800	1200	26,889.1	105.0	1.2	26,995.3	26,297.6	41.678	0.001
114-M	2/ 1/28	410.025	120.029	130.054	1680	1200	26,259.7	98.0	0.2	26,357.9	26,291.8	41.664	−0.013
115-M	2/ 6/28	419.333	120.033	130.053	1740	1200	26,633.2	101.5	1.3	26,736.0	26,681.6	41.669	−0.008
	Even temperature interval 120–130°. $Z]_{120}^{130}$ = 9208.6 int. joules. Mean value of $\alpha]_{120}^{130} = \alpha_m$ = 41.6767 int. joules per gram												

TABLE 3 PRINCIPAL DATA FROM α EXPERIMENTS—*Continued*

Exp. No.	Date	Mass of water M	Initial temp. θ_1	Final temp. θ_2	Total time of exp.	Time of electric heating	Total electric energy	Pump energy	Heat leak	Entire energy added Q	Energy for even temp. intervals $Q]_{t^1}$	$\alpha]_{t^1}$	Residual $\alpha]_{t^1} - \alpha_m$
87-N	11/ 8/27	475.903	130.062	140.403	1920	1200	30,060.6	106.8	1.5	30,168.9	29,177.4	41.712	0.011
89-N	11/10/27	475.903	130.388	139.984	1860	1200	27,895.6	104.2	−0.9	27,998.9	29,175.5	41.708	0.007
90-N	11/11/27	475.903	130.137	139.936	2700	1200	28,433.5	150.4	1.7	28,585.6	29,170.2	41.697	−0.004
93-N	11/15/27	263.730	129.997	139.964	1860	1200	20,153.8	103.7	−1.3	20,256.2	20,324.3	41.700	−0.001
95-N	11/25/27	267.764	129.793	139.792	1740	1200	20,392.7	97.0	−1.3	20,488.4	20,493.7	41.704	0.003
96-N	11/26/27	267.764	129.895	140.089	1740	1200	20,796.7	97.0	−3.6	20,890.1	20,492.7	41.701	0.000
97-N	11/28/27	267.764	129.957	139.988	1740	1200	20,461.7	97.0	−1.6	20,557.1	20,493.8	41.705	0.004
112-N	1/30/28	301.182	129.945	140.009	1860	1200	21,921.4	106.3	2.3	22,030.0	21,887.6	41.705	0.004
113-N	1/31/28	410.025	129.890	140.017	2100	1200	26,635.6	120.0	1.6	26,757.2	26,423.3	41.696	−0.005
114-N	2/ 1/28	410.025	130.054	140.019	1680	1200	26,228.0	96.0	0.8	26,324.8	26,417.9	41.683	−0.018
115-N	2/ 6/28	419.333	130.053	140.017	1740	1200	26,616.5	99.4	1.3	26,717.2	26,812.7	41.700	−0.001
	Even temperature interval 130–140°. $Z]_{130}^{140}$ = 9326.7 int. joules. Mean value of $\alpha]_{130}^{140} = \alpha_m$ = 41.7008 int. joules per gram												
87-O	11/ 8/27	475.903	140.403	150.687	1920	1200	30,043.7	105.7	2.0	20,151.4	29,305.7	41.701	−0.012
89-O	11/10/27	475.903	139.984	149.949	1980	1200	29,103.2	109.0	−2.1	29,210.1	29,312.8	41.716	0.003
90-O	11/11/27	475.903	139.937	149.957	1860	1200	29,274.1	99.2	−0.6	29,372.7	29,314.5	41.719	0.006
93-O	11/15/27	263.730	139.964	150.226	1800	1200	20,893.6	99.2	−2.2	20,990.6	20,452.5	41.680	−0.033
95-O	11/25/27	267.764	139.792	149.969	1740	1200	20,906.2	95.9	−2.6	20,999.5	20,633.6	41.728	0.015
96-O	11/26/27	267.764	140.089	150.219	1980	1200	20,789.2	109.2	−3.4	20,895.0	20,625.5	41.698	−0.015
97-O	11/28/27	267.764	139.988	150.068	1800	1200	20,698.9	99.2	−1.8	20,796.3	20,631.5	41.721	0.008
112-O	1/30/28	301.182	140.010	150.173	1980	1200	22,275.7	111.0	2.7	22,389.4	22,031.1	41.738	0.025
113-O	1/31/28	410.025	140.017	150.213	1860	1200	26,983.7	104.3	1.7	27,089.7	26,567.4	41.722	0.009
114-O	2/ 1/28	410.025	140.019	149.930	2100	1200	26,204.1	117.8	0.9	26,322.8	26,559.3	41.703	−0.010
115-O	2/ 6/28	419.333	140.017	149.990	2100	1200	26,759.6	117.8	2.1	26,879.8	26,951.9	41.713	0.000
	Even temperature interval 140–150°. $Z]_{140}^{150}$ = 9460.3 int. joules. Mean value of $\alpha]_{140}^{150} = \alpha_m$ = 41.7127 int. joules per gram												
88-P	11/ 9/27	475.903	150.712	160.406	1920	1140	28,458.7	104.5	2.3	28,565.5	29,459.1	41.699	−0.004
89-P	11/10/27	475.903	149.949	160.148	1740	1200	29,960.7	94.8	−1.6	30,053.9	29,466.3	41.714	0.011
90-P	11/11/27	475.903	149.957	160.362	1860	1200	30,556.0	101.4	−0.4	30,657.0	29,461.4	41.704	0.001
93-P	11/15/27	263.730	150.226	159.779	1740	1140	19,595.1	94.8	−1.7	19,688.2	20,609.4	41.690	−0.013
95-P	11/25/27	267.764	149.970	160.073	1740	1200	20,893.5	94.8	−2.8	20,985.5	20,763.0	41.636	−0.067
96-P	11/26/27	267.764	150.219	159.963	1800	1200	20,159.8	98.0	−3.8	20,254.0	20,785.8	41.721	0.018
97-P	11/28/27	267.764	150.068	160.066	1920	1200	20,696.3	104.6	−2.8	20,798.1	20,795.4	41.757	0.053
112-P	1/30 28	301.182	150.173	160.256	1860	1200	22,263.3	102.0	2.7	22,368.0	22,180.1	41.721	0.018
113-P	1/31/28	410.025	150.212	160.292	1920	1200	26,824.0	105.2	3.3	26,932.5	26,714.0	41.704	0.001
114-P	2/ 1/28	410.025	149.930	159.959	1920	1200	26,679.6	105.2	−0.7	26,784.1	26,706.4	41.685	−0.018
	Even temperature interval 150–160°. $Z]_{150}^{160}$ = 9614.5 int. joules. Mean value of $\alpha]_{150}^{160} = \alpha_m$ = 41.7027 int. joules per gram												
88-Q	11/ 9/27	475.903	160.406	170.543	1860	1200	29,934.6	100.5	2.1	30,037.2	29,624.3	41.712	0.000
89-Q	11/10/27	475.903	160.148	170.158	1860	1200	29,539.6	100.5	−2.0	29,638.1	29,623.7	41.711	−0.001
90-Q	11/11/27	475.903	160.362	170.390	1740	1200	29,622.0	94.0	−0.3	29,715.7	29,627.2	41.719	0.007
93-Q	11/15/27	263.730	159.779	170.247	1860	1260	21,643.2	100.5	−0.8	21,742.9	20,769.1	41.694	−0.018
95-Q	11/25/27	267.764	160.073	170.086	1680	1200	20,882.1	90.7	−1.5	20,971.3	20,950.2	41.742	0.030
96-Q	11/26/27	267.764	159.963	169.858	1680	1200	20,641.3	90.7	−4.0	20,728.0	20,945.6	41.725	0.013
97-Q	11/28/27	267.764	160.066	169.993	1740	1200	20,686.6	94.0	−4.1	20,776.5	20,933.6	41.680	−0.032
112-Q	1/30/28	301.182	160.256	170.258	1920	1200	22,247.7	103.2	2.1	22,353.0	22,341.1	41.728	0.016
113-Q	1/31/28	410.025	160.292	170.027	1860	1200	26,062.1	100 0	1.8	26,163.9	26,874.1	41.707	−0.005
114-Q	2/ 1/28	410.025	159.960	170.161	1920	1200	27,310.3	103.2	2.2	27,415.7	26,873.3	41.705	−0.007
	Even temperature interval 160–170°. $Z]_{160}^{170}$ = 9773.3 int. joules. Mean value of $\alpha]_{160}^{170} = \alpha_m$ = 41.7120 int. joules per gram												
88-R	11/ 9/27	475.903	170.543	180.113	1980	1140	28,412.7	106.0	2.4	28,521.1	29,798.0	41.691	−0.010
89-R	11/10/27	475.903	170.152	180.220	1800	1200	29,937.3	100.9	−4.0	30,034.2	29,813.2	41.723	0.022
90-R	11/11/27	475.903	170.389	180.213	1800	1200	29,189.0	96.3	0.1	29,285.4	29,805.0	41.706	0.005
93-R	11/15/27	263.730	170.248	180.126	1800	1200	20,599.2	96.3	−0.1	20,695.4	20,945.9	41.667	−0.034
95-R	11/25/27	267.764	170.086	180.007	1740	1200	20,872.7	93.2	−2.1	20,963.8	21,129.4	41.725	0.024
96-R	11/26/27	267.764	169.859	179.789	1980	1200	20,919.0	106.0	−4.1	21,020.9	21,128.8	41.722	0.021
97-R	11/28 27	267.764	169.993	180.424	1800	1260	21,944.9	96 3	−5.0	22,036.2	21,122.2	41.698	−0.003
112-R	1/30/28	301.182	170.259	180.053	1800	1200	21,961.0	95.0	2.4	22,058.4	22,519.1	41.709	0.008
113-R	1/31/28	410.025	170.027	179.951	1980	1200	26,742.4	104.9	0.5	26,847.8	27,052.5	41.694	−0.007
114-R	2/ 1/28	410.025	170.161	180.245	1800	1200	27,179.8	95.3	1.8	27,276.9	27,045.7	41.677	−0.024
	Even temperature interval 170–180°. $Z]_{170}^{180}$ = 9957.1 int. joules. Mean value of $\alpha]_{170}^{180} = \alpha_m$ = 41.7009 int. joules per gram												
88-S	11/ 9/27	475.903	180.113	190.113	1980	1200	29,884.1	105.2	0.7	29,990.0	29,988.4	41.660	0.002
89-S	11/10/27	475.903	180.225	190 223	1680	1200	29,912.0	89.2	−4.5	29,996.7	29,997.7	41.680	0.022
90-S	11/11/27	475.903	180.213	190.040	1680	1200	29,377.6	89.3	−0.6	29,466.3	29,983.1	41.649	−0.009
93-S	11/15/27	263.730	180.126	189.651	1800	1140	20,040.2	95.7	−2.4	20,133.5	21,142.4	41 634	−0.024
95-S	11/25/27	267.764	180.007	189.949	1800	1200	21,109.9	95.6	−3.1	21,202.4	21,326.3	41.693	0.035
96-S	11/26/27	267.764	179.789	189.870	1800	1200	21,352.7	95.6	−3.9	21,444.4	21,319.6	41.668	0.010
97-S	11/28/27	267.764	180.424	189.889	1680	1140	20,095.1	89.2	−5.5	20,178.8	21,315.9	41.654	−0.004
112-S	1/30/28	301.182	180.053	189.885	1860	1200	22,228.2	96.5	0.3	22,325.0	22,706.9	41.651	−0.007
113-S	1/31/28	410.025	179.952	189.804	1860	1200	26,734.6	96.5	0.7	26,831.8	27,238.4	41.646	−0.012
114-S	2/ 1/28	410.025	180.245	189.753	1860	1140	25,798.9	96.5	1.6	25,897.0	27,238.1	41.646	−0.012
	Even temperature interval 180–190°. $Z]_{180}^{190}$ = 10,162.4 int. joules. Mean value of $\alpha]_{180}^{190} = \alpha_m$ = 41.6578 int. joules per gram												
88-T	11/ 9/27	475.903	190.112	200.327	2040	1200	30,744.5	107.5	−1.1	30,850.9	30,197.2	41.657	0.005
89-T	11/10/27	475.903	190.223	200.149	1740	1200	29,887.9	91.7	−2.5	29,977.1	30,196.1	41.655	0.003
90-T	11/11/27	475.903	190.040	200.129	1740	1200	30,388.6	91.7	−1.9	30,478.4	30,208.7	41.681	0.029
93-T	11/15/27	263.730	189.651	200.059	1860	1260	22,132.1	98.0	−2.7	22,227.4	21,357.3	41.651	−0.001
95-T	11/25/27	267.764	189.949	199.910	1860	1200	21,354.3	98.0	−3.1	21,449.2	21,535.9	41.691	0.039
96-T	11/26/27	267.764	189.870	200.046	1800	1200	21,814.9	94.8	−5.9	21,903.8	21,523.3	41.644	−0.008
97-T	11/28/27	267.764	189.889	200.144	1680	1200	21,987.7	88.5	−4.7	22,071.5	21,531.3	41.674	0.022
112-T	1/30/28	301.182	189.885	199.763	1860	1200	22,529.8	94.8	3.2	22,627.8	22,909.8	41.627	−0.025
113-T	1/31/28	410.025	189.804	199.744	1980	1200	27,167.3	101.1	2.1	27,270.5	27,436.9	41.618	−0.034
114-T	2/ 1/28	410.025	189.755	199.677	1860	1200	27,126.8	94.8	1.7	27,223.3	27,440.4	41.626	−0.026
	Even temperature interval 190–200°. $Z]_{190}^{200}$ = 10,372.7 int. joules. Mean value of $\alpha]_{190}^{200} = \alpha_m$ = 41.6514 int. joules per gram												
88-U	11/ 9/27	475.903	200.327	210.173	1860	1200	29,858.4	97.3	−3.3	29,952.4	30,414.3	41.576	0.011
89-U	11/10/27	475.903	200.150	210.146	1740	1200	30,312.9	91.0	−5.1	30,398.8	30,406.1	41.558	−0.007
90-U	11/11/27	475.903	200.129	210.145	1800	1200	30,367.6	94.1	−2.1	30,459.6	30,408.1	41.563	−0.002
92-U	11/14/27	263.730	200.530	210.347	1800	1200	21,122.8	94.1	−2.9	21,214.0	21,599.4	41.600	0.035
93-U	11/15/27	263.730	200.060	210.002	1800	1200	21,380.0	94.1	−3.0	21,471.1	21,595.4	41.585	0.020
95-U	11/25/27	267.764	199.910	209.991	1680	1200	21,851.6	87.8	−5.4	21,934.0	21,757.7	41.564	−0.001
96-U	11/26/27	267.764	200.046	209.954	1740	1200	21,467.8	90.9	−4.9	21,553.8	21,755.5	41.556	−0.009
97-U	11/28/27	267.764	200.145	210.019	1680	1200	21,403.3	87.8	−5.7	21,485.4	21,746.4	41.522	−0.043
	Group I, interval 200–210°. $Z]_{200}^{210}$ = 10,628.3 int. joules. Weight one to five												
173-U	12/17/28	292.871	200.641	210.265	1440	720	22,483.2	109.2	0.0	22,592.4	23,464	41.490	−0.013
174-U	12/18/28	408.332	200.143	209.898	1260	720	27,453.0	95.5	3.1	27,551.6	28,243	41.462	−0.041
175-U	12/19/28	414.265	200.153	210.104	1320	720	28,250.3	100.1	−0.9	28,349.5	28,486	41.455	−0.048
176-U	12/20/28	279.315	200.076	212.371	1200	720	28,100.7	91.0	2.3	28,194.0	22,900	41.484	−0.019
177-U	12/21/28	413.267	200.147	210.167	1320	720	28,430.4	100.1	1.5	28,532.0	28,472	41.521	0.018
178-U	12/26/28	288.124	200.157	209.947	1140	720	22,690.9	86.4	−1.0	22,776.3	23,264	41.479	−0.024
179-U	12/27/28	415.134	200.092	210.123	1500	720	28,512.2	113.7	1.0	28,626.9	28,537	41.491	−0.012

TABLE 3 PRINCIPAL DATA FROM α EXPERIMENTS—*Continued*

Exp. No.	Date	Mass of water M	Initial temp. θ_1	Final temp. θ_2	Total time of exp.	Time of electric heating	Total electric energy	Pump energy	Heat leak	Entire energy added Q	Energy for even temp. intervals $Q]_1^3$	$\alpha]_1^2$	Residual $\alpha]_1^2 - \alpha_m$
181-U	12/29/28	419.612	200.173	209.923	1140	720	27,922.1	86.4	−0.4	28,008.1	28,725	41.496	−0.007
183-U	1/ 2/29	285.531	200.187	209.956	1240	720	22,513.7	94.0	3.0	22,610.7	23,144	41.436	−0.067
184-U	1/ 3/29	286.319	200.150	210.187	1080	720	23,200.7	81.9	−0.3	23,282.3	23,192	41.489	−0.014
189-U	1/22/29	420.292	200.137	210.344	1140	720	29,293.2	63.0	1.7	29,357.9	28,757	41.505	0.002
190-U	1/23/29	277.212	200.207	210.231	1260	720	22,819.9	69.7	1.8	22,891.4	22,832	41.554	0 051
191-U	1/24/29	278.960	200.317	210.323	1080	720	22,826.4	82.0	0.1	22,908.5	22,888	41.494	−0.009
193-U	1/26/29	419.181	200.027	210.144	1080	720	28,964.8	82.0	0.8	29,047.6	28,716	41.517	0 014
195-U	1/31/29	421.532	199.989	210.035	1080	720	28,844.0	82.8	1.9	28,928.7	28,796	41.475	−0.028
			Group II, interval 200–210°. $Z]_{200}^{210}$ = 11,312.8 int. joules. Weight five to one										
			Even temperature interval 200–210°. Mean value of $\alpha]_{200}^{210} = \alpha_m$ = 41.5026 int. joules per gram										
173-V	12/17/28	292.871	210.265	220.213	1200	720	23,493.2	89.6	0.7	23,585.5	23,701	41.405	0.014
174-V	12/18/28	408.332	209.898	219.874	1200	720	28,312.3	89.6	0.1	28,402.0	28,473	41.383	−0.008
175-V	12/19/28	414.265	210.104	220.110	1140	720	28,655.4	85.1	−2.5	28,738.0	28,718	41.382	−0.009
177-V	12/21/28	413.267	210.167	220.239	1200	720	28,795.3	89.6	2.0	28,886.9	28,675	41.378	−0.013
178-V	12/26/28	288.124	209.947	219.928	1200	720	23,369.9	89.6	−1.2	23,458.3	23,504	41.403	0.012
179-V	12/27/28	415.134	210.123	220.153	1140	720	28,763.9	85.1	−3.5	28,845.5	28,755	41.385	−0.006
181-V	12/29/28	419.612	209.923	220.051	1200	720	29,235.1	89.6	−2.0	29,322.7	28,953	41.415	0.024
183-V	1/ 2/29	285.531	209.956	219.918	1200	720	23,202.1	89.6	1.2	23,292.9	23,384	41.359	−0.032
184-V	1/ 3/29	286.319	210.187	220.189	1080	720	23,356.7	80.6	−0.7	23,436.6	23,428	41.399	0.008
189-V	1/22/29	420.292	210.344	220.417	1200	720	29,141.5	65.4	2.7	29,209.6	28,989	41.434	0.043
190-V	1/23/29	277.212	210.231	220.137	1140	720	22,778.1	62.2	1.5	22,841.8	23,054	41.409	0.018
191-V	1/24/29	278.960	210.323	220.222	1200	720	22,798.2	89.7	0.5	22,888.4	23,115	41.369	−0.022
192-V	1/25/29	419.602	210.339	220.278	1260	720	28,670.7	94.2	1.2	28,766.1	28,935	41.373	−0.018
193-V	1/26/29	419.181	210.144	219.963	1080	720	28,318.4	80.7	0.5	28,399.6	28,916	41.369	−0.022
195-V	1/31/29	421.532	210.035	220.048	1140	720	28,973.2	86.5	3.3	29,063.0	29,024	41.395	0.004
			Even temperature interval 210–220°. $Z]_{210}^{220}$ = 11,574.8 int. joules. Mean value of $\alpha]_{210}^{220} = \alpha_m$ = 41.3908 int. joules per gram										
173-W	12/17/28	292.871	220.213	230.171	1200	720	23,775.0	88.0	−0.8	23,862.2	23,957	41.355	0.017
174-W	12/18/28	408.332	219.874	230.201	1260	720	29,562.2	92.4	0.3	29,654.9	28,715	41.313	−0.025
175-W	12/19/28	414.265	222.371	230.407	1020	600	23,221.6	74.8	−2.4	23,294.0	28,954	41.299	−0.039
176-W	12/20/28	279.315	220.264	229.968	1140	720	22,623.9	83.6	−6.5	22,701.0	23,390	41.322	−0.006
177-W	12/21/28	413.267	220.239	230.305	1140	720	29,054.7	83.6	−0.5	29,137.8	28,941	41.367	0.029
178-W	12/26/28	288.124	219.928	230.140	1200	720	24,176.4	88.0	−1.0	24,263.4	23,759	41.349	0.011
179-W	12/27/28	415.134	220.153	230.264	1140	720	29,249.2	83.6	−2.2	29,330.6	29,004	41.333	−0.005
181-W	12/29/28	419.612	220.051	230.114	1140	720	29,307.0	83.6	1 6	29,392.2	29,200	41.359	0.021
183-W	1/ 2/29	285.531	219.918	230.993	1320	780	26,093.7	96.8	−3.8	26,186.7	23,632	41.280	−0.058
184-W	1/ 3/29	286.319	220.189	230.244	1260	720	23,730.0	92.4	−0.1	23,822.3	23,686	41.355	0.017
189-W	1/22/29	420.292	220.417	230.316	1080	720	28,885.6	57.7	1.2	28,944.5	29,230	41.363	0.025
190-W	1/23/29	277.212	220.137	230.131	1140	720	23,238.9	60.9	0.8	23,300.6	23,312	41.364	0.026
191-W	1/24/29	278.960	220.222	230.182	1140	720	23,205.3	83.6	−0.5	23,288.4	23,376	41.334	−0.004
193-W	1/26/29	419.181	219.963	229.940	1140	720	29,015.5	83.6	1.4	29,100.5	29,170	41.330	−0.008
195-W	1/31/29	421.532	220.048	229.960	1080	720	28,934.3	79.2	0.5	29,014.0	29,272	41.341	0.003
			Even temperature interval 220–230°. $Z]_{220}^{230}$ = 11,845.4 int. joules. Mean value of $\alpha]_{220}^{230} = \alpha_m$ = 41.3384 int. joules per gram										
173-X	12/17/28	292.871	230.171	240.112	1320	720	23,998.6	95.0	0.6	24,094.2	24,233	41.345	0.043
174-X	12/18/28	408.332	230.201	240.262	1140	720	29,087.1	82.1	0.0	29,169.2	28,986	41.294	−0.008
175-X	12/19/28	414.265	230.407	240.242	1260	720	28,663.1	90.7	−1.5	28,752.3	29,226	41.282	−0.020
176-X	12/20/28	279.315	229.968	239.850	1080	720	23,304.6	77.8	0.6	23,383.0	23,665	41.318	0.016
177-X	12/21/28	413.267	230.305	240.364	1140	720	29.294.1	82.1	−1.5	29,374.7	29,193	41.302	0.000
178-X	12/26/28	288.124	230.140	240.229	1260	720	24.147.0	90.7	0.3	24,238.0	24,019	41.284	−0.018
179-X	12/27/28	415.134	230.264	240.361	1080	720	29,481.5	77.8	−2.9	29,556.4	29,264	41.287	−0.015
181-X	12/29/28	419.612	230.116	240.268	1200	720	29,830.1	86.4	−1.1	29,915.4	29,461	41.316	0.014
183-X	1/ 2/29	285.531	230.993	240.422	1320	660	22,460.2	95.0	−2.0	22,553.2	23,900	41.241	−0.061
184-X	1/ 3/29	286.319	230.244	240.245	1080	720	23,890.6	77.7	−1.4	23,966.9	23,958	41.331	0.029
189-X	1/22/29	420.292	230.316	240.120	1080	720	28,865.7	56.6	−0.9	28,921.4	29,494	41.328	0.026
190-X	1/23/29	277.212	230.131	240.215	1200	720	23,716.6	62.9	0.7	23,780.2	23,577	41.314	0.012
191-X	1/24/29	278.960	230.182	240.213	1080	720	23,640.8	77.7	0.8	23,719.3	23,640	41.281	−0.021
193-X	1/26/29	419.181	229.940	239.904	1140	720	29.251.3	82.1	−2.9	29,330.5	29,438	41.304	0.002
195-X	1/31/29	421.532	229.960	239.937	1080	720	29,386.8	77.7	1.0	29,465.5	29,534	41.301	−0.001
			Even temperature interval 230–240°. $Z]_{230}^{240}$ = 12,124.2 int. joules. Mean value of $\alpha]_{230}^{240} = \alpha_m$ = 41.3025 int. joules per gram										
173-Y	12/17/28	292.871	240.112	250.217	1200	720	24,688.6	85.0	−0.7	24,772.9	24,511	41.245	0.003
174-Y	12/18/28	408.332	240.262	250.209	1140	720	29,030.4	80.7	−3.3	29,107.8	29,257	41.205	−0.037
175-Y	12/19/28	414.265	240.242	250.142	1140	720	29,147.9	80.7	−3.6	29,225.0	29,515	41.238	−0.004
176-Y	12/20/28	279.315	239.850	249.857	1140	720	23,886.7	80.8	−0.6	23,966.9	23,955	41.256	0.014
177-Y	12/21/28	413.267	240.364	250.363	1080	720	29,405.1	76.5	−1.5	29,480.1	29,472	41.234	−0.008
178 Y	12/26/28	288.124	240.229	250.289	1200	720	24,379.1	85.0	−1.6	24,462.5	24,308	41.220	−0.022
179-Y	12/27/28	415.134	240.361	250.359	1380	720	29,446.7	97.7	−2.2	29,542.2	29,537	41.205	−0.037
181-Y	12/29/28	419.612	240.268	250.295	1080	720	29,769.5	76.5	−1.4	29,844.6	29,757	41.289	0.047
183-Y	1/ 2/29	285.531	240.422	250.441	1200	720	24,166.5	85.0	−0.1	24,251.4	24,192	41.188	−0.054
184-Y	1/ 3/29	286.319	240.245	250.234	1140	720	24,143.0	80.7	−1.0	24,222.7	24,242	41.249	0.007
189-Y	1/22/29	420.292	240.120	249.914	1020	720	29,116.2	52.7	0.7	29,169.6	29,783	41.284	0.042
190-Y	1/23/29	277.212	240.215	250.225	1140	720	23,852.5	58.9	0.4	23,911.8	23,881	41.302	0.060
191-Y	1/24/29	278.960	240.213	250.331	1200	720	24,135.6	85.0	3.8	24,224.4	23,934	41.234	−0.008
193-Y	1/26/29	419.181	239.904	249.947	1080	720	29,760.1	76.5	−0.4	29,836.2	29,711	41.222	−0.020
195-Y	1/31/29	421.532	239.937	250.017	1080	720	29,974.3	76.5	1.0	30,051.8	29,814	41.236	−0.006
			Even temperature interval 240–250°. $Z]_{240}^{250}$ = 12,431.5 int. joules. Mean value of $\alpha]_{240}^{250} = \alpha_m$ = 41.2423 int. joules per gram										
173-Z	12/17/28	292.871	250.217	260.287	1200	720	24.926.0	83.6	−2.7	25,006.9	24,825	41.162	0.035
174-Z	12/18/28	408.332	250.209	260.169	1260	720	29,356.5	87.8	−0.9	29,443.4	29,555	41.107	−0.020
175-Z	12/19/28	414.265	250.142	260.129	1200	720	29,690.6	83.6	−3.1	29,771.1	29,805	41.122	−0.005
176-Z	12/20/28	279.315	249.857	260.426	1140	720	25,563.4	79.4	−1.8	25,641.0	24,255	41.120	−0.007
177-Z	12/21/28	413.267	250.363	260.285	1260	720	29,464.0	87.9	−1.6	29,550.2	29,773	41.144	0.017
178-Z	12/26/28	288.124	250.289	260.266	1260	720	24,469.0	87.8	−1.6	24,555.2	24,603	41.070	−0.057
179-Z	12/27/28	415.134	250.359	260.241	1200	720	29,405.6	83.6	−2.5	29,486.7	29,830	41.096	−0.031
181-Z	12/29/28	419.612	250.295	260.163	1260	720	29,555.1	87.8	0.0	29,642.9	30,032	41.139	0.012
183-Z	1/ 2/29	285.531	250.441	260.077	1260	720	23,545.0	87.8	0.7	23,633.5	24,517	41.142	0.015
184-Z	1/ 3/29	286.319	250.234	260.181	1080	720	24,345.3	75.2	−0.3	24,420.2	24,544	41.123	−0.004
189-Z	1/22/29	420.292	249.914	259.932	1080	720	30,067.2	54.9	0 0	30,122.1	30,071	41.165	0.038
190-Z	1/23/29	277.212	250.225	260.210	1320	720	24,094.0	67.2	1 0	24,162.2	24,192	41.204	0.077
191-Z	1/24/29	278.960	250.331	260.486	1560	720	24,504.0	108.8	0.9	24,613.7	24,226	41.068	−0.059
195-Z	1/31/29	421.532	250.017	260.048	1140	720	30,115.8	79.4	0.3	30,195.5	30,102	41.117	−0.010
			Even temperature interval 250–260°. $Z]_{250}^{260}$ = 12,769.7 int. joules. Mean value of $\alpha]_{250}^{260} = \alpha_m$ = 41.1273 int. joules per gram										
173-&	12/17/28	292.871	260.287	270.374	1460	720	25,288.1	100.0	−5.2	25,382.9	25,152	41.096	0.034
174-&	12/18/28	408.332	260.169	270.165	1340	720	29,790.8	91.8	−6.9	29,875.7	29,882	41.059	−0.003
175-&	12/19/28	414.265	260.129	270.232	1320	720	30,357.5	90.4	1.1	30,449.0	30,133	41.077	0.015
176-&	12/20/28	279.315	260.426	270.044	1560	720	23,552.5	106.9	−1.4	23,658.0	24,590	41.078	0.016
177-&	12/21/28	413.267	260.285	270.200	1320	720	29,748.4	90.4	0.6	29,839.4	30,087	41.065	0.003
178-&	12/26/28	288.124	260.266	270.249	1820	720	24,773.7	124.7	3.9	24,902.3	24,937	41.026	−0.036
179-&	12/27/28	415.134	260.241	270.114	1380	720	29.673.9	94.5	−2.3	29.766.1	30,143	41.015	−0.047
183-&	1/ 2/29	285.531	260.077	269.831	1500	720	24,121.0	102.7	2.0	24,225.7	24,839	41.056	−0.006
184-&	1/ 3/29	286.319	260.181	270.165	1140	720	24,770.1	78.1	0.8	24,849.0	24,883	41.096	0.034
189-&	1/22/29	420.292	259.932	270.044	1440	720	30,650.3	71.8	0.1	30,722.2	30,382	41.080	0.018
190-&	1/23/29	277.212	260.210	270.122	1380	720	24.217.7	68.8	3.5	24,290.0	24,500	41.065	0.003
191-&	1/24/29	278.960	260.486	270.255	1080	720	23,927.6	73.9	0.5	24,092.0	24,557	41.012	−0.050
195-&	1/31/29	421.532	260.048	269.958	1080	720	30,080.7	74.0	1.4	30,156.1	30,429	41.071	0.009
			Even temperature interval 260–270°. $Z]_{260}^{270}$ = 13,116.3 int. joules. Mean value of $\alpha]_{260}^{270} = \alpha_m$ = 41.0618 int. joules per gram										

TABLE 3 (APPENDIX) PRINCIPAL DATA FROM α EXPERIMENTS CONDUCTED OVER INTERVALS OF MORE THAN 10 DEGREES

Exp. No.	Date	Mass of water M Grams	Initial temp. θ_1 Degrees	Final temp. θ_2 centigrade	Total time of exp. Seconds	Time of electric heating Seconds	Total electric energy	Pump energy	Heat leak	Entire energy added Q	Energy for even temp. interval $Q]_1^2$	Even temp. interval, degrees centigrade
							International joules					
45	7/14/27	326.833	9.209	40.071	2760	1860	67,878.1	205.7	2.6	68,086.4	66,189.6	10–40
45	7/14/27	326.833	40.071	70.327	2880	1800	67,030.5	186.0	6.9	67,223.4	66,649.2	40–70
45	7/14/27	326.833	70.327	99.590	2760	1740	65,455.7	212.0	5.3	65,673.0	67,323.7	70–100
46	7/16/27	326.833	39.576	69.960	3060	1740	67,285.2	196.0	6.9	67,488.1	66,638.6	40–70
46	7/16/27	326.833	69.960	99.945	2580	1740	67,094.2	154.0	5.9	67,254.1	67,284.3	70–100
51	7/25/27	493.054	70.093	100.031	2940	1800	87,736.7	178.0	4.9	87,919.6	88,094.9	70–100
53	7/27/27	493.054	1.628	19.809	3720	1080	52,505.8	222.4	3.5	52,731.7	58,020.6	0–20
54	7/28/27	493.054	0.826	99.790	8040	6180	288,001.5	593.6	12.1	288,607.2	291,612.9	0–100
55	7/29/27	493.054	0.604	99.477	8160	6120	287,696.3	595.3	10.8	288,302.4	291,588.2	0–100
56	8/ 1/27	291.001	6.304	58.267	3120	2100	106,856.5	222.8	7.3	107,086.6	123,643.4	0–60
56	8/ 1/27	291.001	58.267	79.908	6120	1380	44,869.7	190.7	5.9	45,066.3	41,657.6	60–80
60	8/ 5/27	291.001	0.566	100.048	6780	5940	205,734.5	464.9	4.0	206,203.4	207,253.0	0–100
67	9/26/27	495.653	1.041	99.483	7380	6300	287,605.5	505.2	9.0	288,119.7	292,688.8	0–100
68	9/29/27	507.031	0.949	99.607	6900	5760	292,942.9	479.5	10.3	293,432.7	297,430.0	0–100
69	9/30/27	507.031	0.682	99.950	6960	5880	294,750.4	474.7	14.2	295,239.3	297,413.0	0–100
70	10/ 3/27	275.167	0.560	99.755	7680	6420	198,535.0	524.4	15.3	199,074.7	200,689.6	0–100
71	10/ 4/27	275.167	0.766	100.153	7560	5820	198,896.7	506.2	13.7	199,416.6	200,629.6	0–100
72	10/ 5/27	275.167	0.445	99.716	7560	5820[a]	198,594.6	540.8	18.7	199,154.1	200,620.5	0–100
73	10/ 6/27	498.832	0.817	99.728	7800	6120[a]	290,209.4	549.2	17.0	290,775.6	293,974.5	0–100
74	10/ 7/27	498.832	0.826	100.325	8460	6120[a]	291,980.6	512.6	15.9	292,509.1	293,965.5	0–100
75	10/ 8/27	498.832	0.795	100.311	8700	6180[a]	291,935.5	594.0	20.0	292,549.5	293,958.1	0–100
77	10/13/27	258.324	0.549	100.160	8700	5940[a]	192,202.0	639.8	18.7	192,860.5	193,595.3	0–100
78	10/14/27	258.324	0.595	100.319	9480	6000[a]	192,542.5	523.2	17.4	193,083.1	193,594.7	0–100
80	10/24/27	268.054	0.834	99.893	8040	6120[a]	195,210.7	581.5	19.7	195,811.9	197,658.5	0–100
81	10/25/27	268.054	0.789	99.787	7740	6000[a]	195,104.4	559.9	15.7	195,680.0	197,652.1	0–100
82	10/26/27	268.054	0.670	100.042	7980	6060[a]	195,854.1	577.2	15.0	196,446.3	197,672.1	0–100
83	10/27/27	268.054	0.506	99.943	8100	6000[a]	195,974.6	582.0	13.6	196,570.2	197,675.6	0–100
84	10/28/27	475.116	0.786	100.353	8100	6060	282,255.8	592.8	13.5	282,862.1	284,074.3	0–100
85	10/29/27	475.116	0.927	100.068	8220	6000	281,037.0	593.9	12.4	281,643.3	284,073.6	0–100
86	11/ 7/27	475.903	100.523	198.456	8340	7320	287,655.2	457.1	25.9	288,138.2	294,320.1	100–200
91	11/12/27	475.903	100.091	200.368	13140	8220	294,532.8	533.7	− 7.9	295,058.6	294,403.2	100–200
115	2/ 6/28	419.333	149.990	199.855	6480	5880	136,532.9	343.0	− 0.8	136,875.1	137,246.6	150–200
116	2/11/28	388.753	99.958	199.817	1086	10200	257,020.1	605.0	2.2	257,627.3	258,010.8	100–200
185	1/ 4/29	418.892	200.124	270.546	5820	5100	207,129.6	419.0	− 6.4	207,542.2	206,249.0	200–270

[a] Time determined with a sidereal chromometer and must be multiplied by 0.99735 to obtain mean solar seconds.

tures of the calorimeter, and the computation of the electric energy added. The indication of each of the three platinum resistance thermometers is computed separately. The mean of the three is then taken as the temperature of the reference block. The mean of the six readings of the five series thermo-elements on the calorimeter, reduced to degrees, gives the difference in temperature between the reference block and the calorimeter. The observed temperature of the reference block minus this difference, equals the calorimeter temperature at the middle instant of the series of temperature observations.

The computation of temperature from the observed resistance of the platinum thermometers is made by use of the Callendar formula:

$$\theta = 100(R_\theta - R_0)/(R_{100} - R_0) + 0.01\,\theta\,(0.01\,\theta - 1)\delta$$

The constants, R_0, $R_{100} - R_0$, and δ are determined, as previously described, by calibration either directly or indirectly, using ice, steam, and boiling sulphur as fixed points.

In computing the electric energy, corrections are applied for potentiometer calibration, standard 0.01-ohm resistance, volt-box ratio, and for the fraction of the total measured current which is carried by the volt box as a shunt across the heater. These corrections are added to the mean values of the potentiometer readings for current and voltage. The product of the corrected mean values of current and voltage, multiplied by the number of seconds of electric heating, gives the total energy added electrically. An approximation is involved in taking the means of the readings before taking the products, but in all these experiments the power was so nearly constant that even in the extreme case this is less than 1 part in 100,000, and therefore negligible.

The principal data from the entire group of experiments of this type are given in Table 3. These experiments are arranged in groups, each of which includes all those covering a certain temperature interval. The masses given are corrected to the nearest milligram for error in the weights. These masses are the amounts of water introduced into the system, including the portion filling the water line out to the shut-off valve. The initial and final temperatures and total electric energy are computed as described above.

The entire energy added, Q, is the sum of the total electric energy, pump energy, and heat leak. This entire energy actually added is adjusted to the value it would have had for the even temperature interval which was approximated, giving the values of $Q]_1^2$. The calculation of α_m, the mean value of α for an entire group of experiments, follows next. This is a least-square calculation using the formula:

$$Q]_1^2 = [M\alpha + Z]_1^2$$

This reduction, using the entire group of values of M and $Q]_1^2$ for a single temperature interval, yields the mean value α_m, and $Z]_1^2$. Using this value of $Z]_1^2$, which is a constant for the calorimeter as used, the individual values of $\alpha]_1^2$ for each experiment are next computed, and compared with the mean for the group.

In the last section of Table 3 the principal data are assembled from those α experiments which were carried out in intervals of more than 10 deg. cent. A considerable number of these, particularly between 0 deg. and 100 deg., were made in 100-deg. non-stop experiments which serve as control experiments, adding greater certainty to the values at 100 deg. and 200 deg. cent. than the experiments in shorter instalments alone give.

In order best to utilize the whole aggregate of α experiments, the results are assembled in Table 4 for the purpose of computing the values of the change in α from 0 to 100, from 100 to 200, and from 200 to 270 from the entire series of energy measurements. With very few exceptions the energy account for each of these three large intervals was kept continuously, the stops at the intermediate 10-deg. points for observing temperatures being made for the purpose of independently establishing these intermediate values of α, but without interruption of the heat-leak and pump-energy control.

The independent determinations of $\alpha]_0^{100}$, $\alpha]_{100}^{200}$, and $\alpha]_{200}^{270}$ are computed in Table 4 after summing up the total energy added in each entire experiment. The calculations for α_m and Z are made for these groups of data by least-square reduc-

TABLE 4

Calculation of Mean Value of $\left[H - L\frac{M}{M'-M}\right]_0^{100}$

Exp. No.	Mass of water M, grams	Energy added for even temperature interval, $Q]^2$, int. joules: 0–10°	10–20°	20–30°	30–40°	40–50°	50–60°	60–70°	70–80°	80–90°	90–100°	$\Sigma Q]^2 = Q]_0^{100}$	$\alpha]_0^{100}$	Residual $\alpha]_0^{100} - \alpha_m$
42	326.833	22,032.0	22,025.8	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....
43	326.833	.....	.....	22,061.9	22,100.6	22,153.5	.....	.....	.....	.....	.....	.....	.....	.....
44	326.833	.....	.....	.....	.....	.....	22,210.1	22,278.1	22,342.4	22,420.2	22,500.2	222,124.8	417.125	−0.232
45	326.833	22,039.0	←	66,189.6	→	←	66,649.2	→	←	67,323.7	→	222,201.5	417.360	0.003
46	326.833	22,033.8	22,024.7	22,057.9	22,103.6	←	66,638.6	→	←	67,284.3	→	222,142.9	417.181	−0.176
49	493.054	29,033.6	28,979.0	29,005.8	29,047.4	29,084.1	29,158.3	29,214.1	29,285.6	.....	.....	.....	.....	.....
50	493.054	.....	.....	.....	.....	.....	.....	29,212.5	29,280.5	29,350.0	29,450.9	291,603.7	417.417	0.060
51	493.054	29,026.6	28,980.9	28,998.2	29,043.4	29,093.6	29,147.2	29,212.3	←	88,094.9	→	291,597.1	417.404	0.047
53	493.054	←	58,020.6	→	29,002.7	29,044.9	29,089.2	29,148.7	29,209.0	29,282.3	29,357.5	29,444.5	291,599.4	417.408	0.051
54	493.054	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	291,612.9	417.436	0.079
55	493.054	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	291,588.2	417.386	0.029
56	291.001	←	.....	.....	123,643.4	.....	→	←	41,657.6	→	20,936.7	21,030.2	207,267.9	417.433	0.076
57	291.001	20,555.7	20,538.7	20,572.5	20,616.8	20,667.0	20,740.0	20,787.0	20,861.5	20,941.0	21,029.1		207,309.3	417.575	0.218
58	291.001	20,543.6	20,542.2	20,576.8	20,619.7	20,660.6	20,727.9	20,791.6	20,858.0	20,934.6	21,023.0		207,278.0	417.468	0.111
59	291.001	20,546.7	20,539.5	20,583.8	20,632.5	20,673.2	20,734.2	20,798.2	20,869.7	20,948.7	21,028.8		207,355.3	417.733	0.376
60	291.001	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		207,253.0	417.382	0.025
62	366.801	23,710.6	23,699.9	23,732.4	23,774.0	23,821.9	23,884.6	23,947.5	24,016.4	24,093.9	24,180.6		238,861.8	417.303	−0.054
63	366.801	23,702.8	23,699.3	23,731.4	23,765.2	23,818.3	23,880.7	23,947.4	24,015.3	24,088.1	24,178.3		238,826.8	417.208	−0.149
67	495.653	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		292,688.8	417.418	0.061
68	507.031	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		297,430.0	417.402	0.045
69	507.031	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		297,413.0	417.368	0.011
70	275.167	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		200,689.6	417.546	0.189
71	275.167	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		200,629.6	417.329	−0.028
72	275.167	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		200,620.5	417.296	−0.061
73	498.832	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		293,974.5	417.335	−0.022
74	498.832	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		293,965.5	417.317	−0.040
75	498.832	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		293,958.1	417.302	−0.055
77	258.324	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		193,595.3	417.308	−0.049
78	258.324	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		193,594.7	417.306	−0.051
80	268.054	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		197,658.5	417.319	−0.038
81	268.054	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		197,652.1	417.295	−0.062
82	268.054	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		197,672.1	417.370	0.013
83	268.054	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		197,675.6	417.383	0.026
84	475.116	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		284,074.3	417.329	−0.028
85	475.116	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....		284,073.6	417.328	−0.029

Even temperature interval 0–100°. $Z]_0^{100}$ = 85,794.5 int. joules. Mean value of $\alpha]_0^{100} = \alpha_m$ = 417.357 int. joules per gram

NOTE: The results of experiments 42 to 60, inclusive, were obtained in the earlier stages of the work and were given one-quarter the weight of the later results.

Calculation of Mean Value of $\left[H - L\frac{M}{M'-M}\right]_{100}^{200}$

Exp. No.	Mass of water M, grams	Energy added for even temperature interval, $Q]^2$, int. joules: 100–110°	110–120°	120–130°	130–140°	140–150°	150–160°	160–170°	170–180°	180–190°	190–200°	$\Sigma Q]^2 = Q]_{100}^{200}$	$\alpha]_{100}^{200}$	Residual $\alpha]_{100}^{200} - \alpha_m$
86	475.903	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	294,320.1	416.817	−0.056
87	475.903	28,823.8	28,927.0	29,046.4	29,177.4	29,305.7	.....	.....	.....	.....	.....	.....	.....	.....
88	475.903	.....	.....	.....	.....	.....	29,459.1	29,624.3	29,798.0	29,988.4	30,197.2	294,347.3	416.874	0.001
89	475.903	28,825.7	28,929.7	29,046.2	29,175.5	29,312.8	29,466.3	29,623.7	29,813.2	29,997.7	30,196.1	294,386.9	416.957	0.084
90	475.903	28,825.4	28,928.8	29,047.9	29,170.2	29,314.5	29,461.4	29,627.2	29,805.0	29,983.1	30,208.7	294,372.2	416.927	0.054
91	475.903	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	294,403.2	416.992	0.119
95	267.764	20,144.5	20,251.9	20,365.8	20,493.7	20,633.6	20,763.0	20,950.2	21,129.4	21,326.3	21,535.9	207,594.3	416.929	0.056
96	267.764	20,144.5	20,249.9	20,364.9	20,492.7	20,625.5	20,785.8	20,945.6	21,128.8	21,319.6	21,523.3	207,580.6	416.878	0.005
97	267.764	20,150.3	20,247.5	20,369.0	20,493.8	20,631.5	20,795.4	20,933.6	21,122.2	21,315.9	21,531.3	207,590.5	416.915	0.042
112	301.182	21,542.1	21,650.2	21,760.8	21,887.6	22,031.1	22,180.1	22,341.1	22,519.1	22,706.9	22,909.8	221,528.8	416.935	0.062
113	410.025	26,077.0	26,183.7	26,297.6	26,423.3	26,567.4	26,714.0	26,874.1	27,052.5	27,238.4	27,436.9	266,864.9	416.827	−0.046
114	410.025	26,066.3	26,173.2	26,291.8	26,417.9	26,559.3	26,706.4	26,873.3	27,045.7	27,238.1	27,440.4	266,812.4	416.698	−0.175
115	419.333	26,457.4	26,565.8	26,681.6	26,812.7	26,951.9	←		137,246.6		→	270,716.0	416.758	−0.115
116	388.753	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	258,010.8	416.859	−0.014

Even temperature interval 100–200°. $Z]_{100}^{200}$ = 95,955.6 int. joules. Mean value of $\alpha]_{100}^{200} = \alpha_m$ = 416.873 int. joules per gram

Calculation of Mean Value of $\left[H - L\frac{M}{M'-M}\right]_{200}^{270}$

Exp. No.	Mass of water M, grams	Energy added for even temperature interval, $Q]^2$, int. joules: 200–210°	210–220°	220–230°	230–240°	240–250°	250–260°	260–270°	$\Sigma Q]^2 = Q]_{200}^{270}$	$\alpha]_{200}^{270}$	Residual $\alpha]_{200}^{270} - \alpha_m$
173	292.871	23,464	23,701	23,957	24,233	24,511	24,825	25,152	169,843	289.276	0.199
174	408.332	28,243	28,473	28,715	28,986	29,257	29,555	29,882	203,111	288.952	−0.122
175	414.265	28,486	28,718	28,954	29,226	29,515	29,805	30,133	204,837	288.980	−0.097
176	279.315	22,900	(23,131)[a]	23,390	23,665	23,955	24,255	24,590	165,886	289.148	0.071
177	413.267	28,472	28,675	28,941	29,193	29,472	29,773	30,087	204,613	289.136	0.059
178	288.124	23,264	23,504	23,759	24,019	24,308	24,603	24,937	168,394	289.013	−0.064
179	415.134	28,537	28,755	29,004	29,264	29,537	29,830	30,143	205,070	288.937	−0.140
183	285.531	23,144	23,384	23,632	23,900	24,192	24 517	24,839	167,608	288.885	−0.192
184	286.319	23,192	23,428	23,686	23,958	24,242	24,544	24,883	167,933	289.225	0.148
185	418.892	.....	.....	.....	.....	.....	.....	.....	206,249	289.159	0.089
189	420.292	28,757	28,989	29,230	29,494	29,783	30,071	30,382	206,706	289.283	0.206
190	277.212	22,832	23,054	23,312	23,577	23,881	24,192	24,500	165,348	289.401	0.324
191	278.960	22,888	23,115	23,376	23,640	23,934	24,226	24,557	165 636	288.620	−0.457
192	419.602	.....	28,935	.....	.....	.....	.....	.....	.....	.....	.....
195	421.532	28,796	29,024	29,272	29,534	29,814	30,102	30,429	206,971	289.061	−0.016

Even temperature interval 200–270°. $Z]_{200}^{270}$ = 85,122.5 int. joules. Mean value of $\alpha]_{200}^{270} = \alpha_m$ = 289.077 int. joules per gram

[a] Calculated from experiment 192.

tion, in the same manner as for the 10-deg. groups in Table 3. The value for each individual experiment is then calculated and the deviations from the mean found.

The next step in Table 5 in the reduction is the adjustment of the 10-deg. values of α, taking into account the results of the last calculation of the change in α between the temperatures of 0, 100, 200, and 270 deg. cent. This adjustment consists in distributing the discrepancy between the 10-deg. steps and the 100- or 70-deg. steps proportionately so as to make the adjusted values of the 10-deg. steps add up to the same value as the overall. Having so adjusted the intermediate values, they are next summed up, giving as the result of this entire group of α experiments the table of values of $H - L\frac{u}{u' - u}$ at each multiple of ten degrees reckoned from 0 deg. cent.

No adjustment for smoothing the trend of values has, up to this point, been introduced into the reduction. This will be undertaken under the head of "Formulation," to be described later.

2—Gamma Experiments

An example of the record of a single determination of γ, equal to $L\frac{u'}{u' - u}$, is given in Table 6. The total time of the experiment in this case is the time between the automatic shifting of the flow of vapor from the spill to the receiver and the shifting to the next receiver, and applies to the electric heating, the pump energy, and the heat leak. The temperature readings are so distributed as to give the initial and final temperatures at these instants of starting and stopping the experiment. The readings of the thermometer with leads in the n position and of the thermoelements 1–5 were taken at one-minute intervals during the experiment. These readings serve to determine the variation of temperature of the calorimeter during the experiment.

A reduction, showing the calculation of γ, is given in Table 7. This reduction makes use of the formula:

$$\gamma_1 = \frac{1}{\Delta M}(Q_E + Q_P + Q_L - [Z + M_2\alpha]_1^2) + \frac{\Delta H'}{\Delta\theta}(\theta_1 - \theta_m)$$

in which Q_E, Q_P, and Q_L denote the amounts of energy added electrically by the pump and by heat leak, respectively. The term $[Z + M_2\alpha]_1^2$ expresses the energy correction for the difference of the final temperature of the calorimeter and its contents from the initial. The last term expresses the energy correction

TABLE 5 ADJUSTMENT OF VALUES OF $\left[H - L\frac{u}{u' - u}\right]$ AND ASSEMBLY OF FINAL VALUES RECKONED FROM 0 DEG. CENT.

Temperature interval deg. cent.	Mean value of $[\alpha]_1^2$	Adjusted value of $[\alpha]_1^2$	Final adjusted value of $[\alpha]_0^\theta$	Temperature deg. cent.
	International joules per gram			
0–10	41.984	41.989	41.989	10
10–20	41.802	41.808	83.797	20
20–30	41.708	41.714	125.511	30
30–40	41.704	41.709	167.220	40
40–50	41.698	41.704	208.924	50
50–60	41.683	41.688	250.612	60
60–70	41.683	41.689	292.301	70
70–80	41.685	41.690	333.991	80
80–90	41.666	41.671	375.662	90
90–100	41.690	41.695	417.357	100
Total 0–100	417.303	417.357		
	417.357			
100–110	41.678	41.678	459.04	110
110–120	41.684	41.683	500.72	120
120–130	41.677	41.676	542.39	130
130–140	41.700	41.700	584.09	140
140–150	41.713	41.712	625.81	150
150–160	41.703	41.702	667.51	160
160–170	41.712	41.712	709.22	170
170–180	41.701	41.701	750.92	180
180–190	41.658	41.658	792.58	190
190–200	41.651	41.651	834.23	200
Total 100–200	416.877	416.873		
	416.873			
200–210	41.503	41.519	875.75	210
210–220	41.391	41.407	917.16	220
220–230	41.338	41.354	958.51	230
230–240	41.302	41.318	999.83	240
240–250	41.242	41.258	1041.09	250
250–260	41.127	41.143	1082.23	260
260–270	41.062	41.078	1123.31	270
Total 200–270	288.966	289.077		
	289.077			

TABLE 6 SAMPLE DATA SHEET—DETERMINATION OF $L\frac{u'}{u' - u}$

Experiment No. 273-A — Nov. 16, 1929

at 270 deg. cent.

Observers: N. S. O., E. F. F., D. C. G.

Time	Thermometers, bridge readings, ohms: $2n$	$2r$	Thermoelements t.e. 1–5 regulating, microvolts	I amperes	E volts	
1:32:00						
:30	51.8960		9.7			
:40	59		7.8			
:50	59		6.0			
1:33:00			(−3)[a]			Start
:10		51.9027	4.0			
:20		28	3.6			
:30		27	3.8			
1:34	51.8960		5	−2		21.999
:35	58		8	(−1)	2.1549	
:36	64		10	1		97
:37	66		10	−1	47	
:38	65		7	(−2)		96
:39	64		9	0	47	
:40	64		12	(1)		96
:41	65		9	(−1)	47	
:42	64		13	(2)		95
:43	71		13	(2)	46	
:44	70		9	(−1)		95
:45	58		11	(1)	45	
:46	57		13	(2)		93
:47	58		6	−4	44	
:48	56		6	(−2)		92
:49	56		8	(−1)	43	
:50	58		7	(−2)		91
:51	57		8	(−1)	42	
:52	56		10	(0)		90
:53	56		10	1	41	
:54	51		11	(1)		89
1:55:00	51		7	(−2)	41	
:30	51.8954		6.8			
:40	57		6.4			
:50	58		6.9			
1:56:00			(−2)			Stop
:10		51.9028	8.7			
:20		28	9.2			
:30		30	9.8			

Total heat-leak factor = −14 microvolt-minutes.

Total time of experiment and electric heating = 1380 seconds.

[a] Values in parentheses calculated from indications of t.e. 1–5.

TABLE 7 SAMPLE COMPUTATION—DETERMINATION OF $L\frac{u'}{u' - u}$

Experiment No. 273-A

	Initial	Final
Temperature		
$(n + r)/2$ on Bridge	51.89933	51.89925
Bridge correction	.1168	0.01168
$R\theta$	51.91101	51.91093
$R\theta - R_0$	26.30321	26.30313
$100(R\theta - R_0)/(R_{100} - R_0)$	263.2320	263.2313
$0.01\theta\,(0.01\theta - 1)\delta$	6.7999	6.7998
Reference block temperature	270.0319	270.0311
Mean t.e. 1–5 (microvolts)	5.82	7.97
Mean t.e. 1–5 (degrees)	−0.0189	−0.0259
Temperature of calorimeter	270.0130	270.0052

Calculation of Energy

Potentiometer	Current		Voltage
Mean reading	2.15447	Mean reading	21.9939
Potentiometer correction	.20	Potentiometer correction	20
Correction for standard 0.01 ohm	11	Correction for volt-box factor	73
Correction for volt-box current	66	Voltage = E =	22.0032
Current = I =	2.15390		

Time of experiment = t = 1380 seconds		
Total electric energy = IEt =	65,401.9	joules
Pump power = 0.0477 watt		
Pump energy =	65.8	
Total heat-leak factor = −14 microvolt-minutes		
Heat-leak coefficient = 0.156 joule per microvolt-minute		
Heat leak =	−2.2	
Change in calorimeter temp. = −0.0068 deg.		
Factor for calorimeter temp. change = 2636 joules per degree		
Energy for calorimeter temp. change =	17.9	
Total energy for evaporation	65,483.4	joules
Mass withdrawn = 39.364 grams		
$L\frac{u'}{u' - u}$ at 270.0130 deg.	1663.54	joules per gram

TABLE 8 PRINCIPAL DATA FROM γ EXPERIMENTS

$$\gamma_{\theta_1} = \frac{1}{\Delta M}\left\{Q_E + Q_P + Q_L - [Z + M_{\theta_1}\alpha]_{\theta_1}^{\theta_2}\right\} + \frac{\Delta H'}{\Delta\theta}(\theta_1 - \theta_m)$$

Exp. No.	Date	Initial temp. θ_1	Final temp. θ_2	Total time of exp.	Mass of vapor removed ΔM	Mass of water remaining M_2	Total electric energy Q_E	Pump energy Q_P	Heat leak Q_L	Energy correction for final temp. diff. $-[Z + M_{\theta_1}\alpha]_{\theta_1}^{\theta_2}$	Entire energy added	Entire energy per gram γ_1	$\gamma\theta$ corrected to even temp.	Residual $\gamma\theta - \gamma_m$
		Degrees centigrade		Minutes	Grams		International joules					International joules per gram		
228-A	5/25/29	99.956	99.941	60	28.631	360.9	64,434.8	226.2	−6.5	36.2	64,690.7	2259.46	2259.35	0.64
B		99.941	99.946	60	28.591	332.3	64,380.9	226.2	−8.9	−12.4	64,585.8	2258.95	2258.80	0.09
C		99.946	99.941	60	28.576	303.7	64,345.3	226.2	−1.8	11.1	64,580.8	2259.96	2259.82	1.11
229-A	5/28/29	99.959	99.957	30	28.905	348.3	65,170.9	113.1	−1.2	40.9	65,323.7	2259.94	2259.83	1.12
B		99.957	99.962	30	28.903	319.4	65,189.4	113.1	−1.6	−11.7	65,289.3	2258.91	2258.80	0.09
C		99.962	99.984	30	28.896	290.5	65,192.5	113.1	−5.9	−48.3	65,251.4	2258.15	2258.05	−0.66
D		99.984	99.969	16	15.433	275.0	34,770.2	60.3	−1.1	32.4	34,861.8	2258.91	2258.87	0.16
231-A	6/ 8/29	99.862	99.844	30	29.400	320.7	66,257.3	113.1	−2.0	42.4	66,410.8	2258.87	2258.50	−0.21
B		99.844	99.833	30	29.395	291.3	66,230.4	113.1	3.4	23.0	66,369.9	2257.86	2257.45	−1.26
232-A	6/11/29	99.966	99.949	20	30.651	374.7	69,101.1	75.4	−5.5	42.3	69,213.3	2258.11	2258.02	−0.69
B		99.949	99.949	21	32.165	342.5	72,558.9	79.2	1.3	− 0.2	72,639.2	2258.33	2258.19	−0.52
C		99.949	99.938	20	30.634	311.9	69,095.4	75.4	1.9	25.9	69,198.6	2258.88	2258.74	0.03
D		99.938	99.957	20	30.604	281.3	69,087.9	75.4	−1.1	−40.9	69,121.3	2258.57	2258.41	−0.30
233-A	6/12/29	100.010	100.020	22	22.562	407.4	50,903.5	82.9	−0.4	−25.5	50,960.5	2258.69	2258.72	0.01
B		100.001	99.987	21	32.178	373.9	72,566.8	79.2	−2.5	36.7	72,680.2	2258.69	2258.69	−0.02
C		99.987	99.982	20	30.646	343.3	69,112.4	75.4	−1.8	11.5	69,197.5	2257.96	2257.93	−0.78
D		99.982	99.995	20	30.611	312.7	69,113.2	75.4	0.0	−30.3	69,158.3	2259.26	2259.21	0.51
E		99.995	100.001	20	30.615	282.1	69,114.9	75.4	−3.3	−13.2	69,173.8	2259.47	2259.46	0.75
234-A	6/13/29	99.986	100.000	40	39.074	311.3	88,105.2	150.8	−0.4	−30.4	88,225.2	2257.90	2257.86	−0.85
Even temperature 100°. Z_{100} = 956.3 int. joules per deg. cent. α_{100} = 4.169 int. joules per gram deg. cent. Mean value of $\gamma_{100} = \gamma_m$ = 2258.71 int. joules per gram														
213-A	4/22/29	129.939	129.949	30	29.708	367.8	64,536.6	153.6	−6.0	−24.7	64,659.5	2176.50	2176.33	−0.82
B		129.949	129.950	30	29.723	338.1	64,535.9	153.6	−4.0	− 1.9	64,683.6	2176.21	2176.07	−1.07
225-A	5/22/29	129.915	129.908	30	31.207	375.1	67,802.1	109.8	7.7	16.0	67,935.6	2176.93	2176.69	−0.46
B		129.908	129.924	30	31.151	343.9	67,813.0	109.8	−6.9	−37.8	67,878.1	2179.00	2178.74	1.59
C		129.924	129.912	30	31.206	312.7	67,789.9	109.8	7.6	26.9	67,934.2	2176.96	2176.75	−0.40
D		129.912	129.932	30	31.143	281.6	67,740.3	109.8	3.0	−41.6	67,811.5	2177.42	2177.13	−0.02
226-A	5/23/29	129.943	129.934	30	30.982	344.9	67,326.1	109.8	−2.9	22.6	67,455.6	2177.25	2177.09	−0.06
B		129.934	129.944	30	30.963	313.9	67,315.7	109.8	−1.8	−23.9	67,399.8	2176.79	2176.60	−0.55
C		129.944	129.931	30	30.953	283.0	67,258.9	109.8	−0.5	28.7	67,396.9	2177.39	2177.23	0.08
D		129.931	129.949	15	15.447	267.5	33,627.7	54.9	0.1	−37.3	33,645.4	2178.12	2177.93	0.78
227-A	5/24/29	129.944	129.934	60	31.037	347.0	67,343.2	219.6	−9.3	22.9	67,576.4	2177.29	2177.13	−0.02
B		129.937	129.906	15	28.583	300.6	62,108.1	54.9	−1.4	69.6	62,231.2	2177.21	2177.03	−0.12
C		129.906	129.917	15	28.527	272.0	62,076.4	54.9	−1.3	−23.2	62,106.8	2177.12	2176.87	−0.28
235-A	6/14/29	129.986	129.986	20	32.752	379.3	71,267.5	73.2	−1.2	1.0	71,340.5	2178.20	2178.16	1.01
B		129.986	129.967	20	32.773	346.5	71,251.1	73.2	−1.0	44.1	71,367.4	2177.63	2177.59	0.44
C		129.967	129.979	20	32.739	313.8	71,227.1	73.2	−0.3	−26.8	71,273.2	2177.01	2176.92	−0.23
D		129.979	129.976	20	32.728	281.0	71,189.7	73.2	−0.7	7.3	71,269.5	2177.63	2177.57	0.42
E		129.976	129.972	20	32.707	248.3	71,154.6	73.2	−1.0	7.7	71,234.5	2177.96	2177.90	0.75
236-A	6/17/29	129.994	130.000	31	30.877	350.6	67,132.3	113.5	4.1	−15.5	67,234.4	2177.49	2177.47	0.32
B		130.000	129.989	31	30.904	319.7	67,139.0	113.5	1.1	28.9	67,282.5	2177.15	2177.15	0.00
C		129.989	129.985	31	30.885	288.8	67,145.6	113.5	−2.3	9.0	67,265.8	2177.94	2177.91	0.76
D		129.985	130.000	31	30.877	257.8	67,128.4	113.5	4.9	−37.4	67,209.4	2176.68	2176.64	−0.51
264-A	10/12/29	130.021	130.015	30	29.317	372.9	63,681.2	109.8	−4.5	17.0	63,803.5	2176.33	2176.39	−0.76
B		130.015	129.997	30	29.284	343.6	63,593.6	109.8	−4.9	42.7	63,741.2	2176.66	2176.70	−0.45
C		129.997	130.015	30	29.207	314.4	63,530.6	109.8	−5.5	−41.4	63,593.5	2177.34	2177.33	0.18
D		130.015	129.996	30	29.246	285.2	63,487.4	109.8	−5.7	41.2	63,632.7	2175.58	2175.62	−1.53
E		129.996	130.024	30	29.167	256.0	63,452.9	109.8	−5.1	−56.8	63,500.8	2177.15	2177.14	−0.01
F		130.024	130.009	30	29.200	226.8	63,425.9	109.8	−2.8	28.7	63,561.6	2176.77	2176.84	−0.31
G		130.009	129.989	30	29.189	197.6	63,402.2	109.8	−3.3	36.9	63,545.6	2177.04	2177.07	−0.08
Even temperature 130°. Z_{130} = 992.0 int. joules per deg. cent. α_{130} = 4.169 int. joules per gram deg. cent. Mean value of $\gamma_{130} = \gamma_m$ = 2177.15 int. joules per gram														
246-A	7/24/29	150.004	149.997	20	19.876	294.8	42,054.4	71.8	2.0	15.8	42,144.0	2120.35	2120.36	0.72
B		149.997	149.997	20	19.860	274.9	42,040.2	71.8	−0.6	− 1.1	42,110.3	2120.36	2120.35	0.71
247-A	7/25/29	149.960	149.966	60	29.036	387.4	61,344.6	215.4	5.3	−15.3	61,550.0	2119.78	2119.66	0.02
B		150.027	150.010	20	39.983	304.9	84,592.9	71.8	−0.1	40.3	84,704.9	2118.52	2118.60	−1.04
C		150.010	149.999	21	42.193	262.8	89,334.7	75.4	3.5	22.7	89,436.3	2119.70	2119.73	0.09
248-A	7/26/29	150.013	150.006	15	30.331	367.8	64,209.0	53.9	−0.7	19.9	64,282.1	2119.35	2119.39	−0.25
B		150.006	150.012	15	30.309	337.4	64,190.7	53.9	2.2	−14.8	64,232.0	2119.24	2119.26	−0.38
C		150.006	150.010	15	30.304	276.7	64,173.6	53.9	−1.7	−10.4	64,215.4	2119.04	2119.06	−0.58
D		150.010	150.010	15	30.296	246.4	64,160.2	53.9	3.0	1.2	64,218.3	2119.70	2119.73	0.09
261-A	9/26/29	149.943	149.976	30	31.091	367.1	65,909.7	97.8	−6.8	−83.7	65,917.0	2120.13	2119.96	0.32
B		149.976	149.960	31	32.158	334.9	68,040.2	101.1	−9.4	37.9	68,169.8	2119.84	2119.77	0.13
C		149.960	149.948	30	31.083	303.9	65,782.3	97.8	−3.6	28.4	65,904.9	2120.29	2120.17	0.53
D		149.948	149.977	30	31.008	272.8	65,734.9	97.8	−8.5	−62.8	65,761.4	2120.79	2120.64	1.00
E		149.977	149.983	30	31.028	241.8	65,708.4	97.8	−7.0	−13.4	65,785.8	2120.21	2120.14	0.50
F		149.983	149.985	30	31.027	210.8	65,697.4	97.8	−6.6	− 3.4	65,785.2	2120.26	2120.21	0.57
263-A	10/11/29	149.959	149.978	30	29.457	327.6	62,356.5	107.7	−0.4	−45.8	62,418.0	2118.95	2118.83	−0.81
B		149.966	149.962	30	29.407	268.6	62,196.1	107.7	−1.1	9.4	62,312.1	2118.95	2118.85	−0.79
C		149.962	149.961	30	29.371	239.2	62,111.2	107.7	−3.4	0.6	62,216.1	2118.28	2118.17	−1.47
D		149.961	149.959	30	29.303	209.9	62,038.6	107.7	−1.6	4.0	62,148.7	2120.90	2120.79	1.15
E		149.959	149.959	30	29.287	180.6	61,961.4	107.7	−2.0	− 0.4	62,066.7	2119.26	2119.14	−0.50
Even temperature 150°. Z_{150} = 1021 int. joules per deg. cent. α_{150} = 4.171 int. joules per gram. deg. cent. Mean value of $\gamma_{150} = \gamma_m$ = 2119.64 int. joules per gram														
249-A	7/30/29	169.989	170.010	15	30.777	354.7	63,345.9	52.7	5.0	−52.4	63,351.2	2058.39	2058.36	−0.42
B		170.010	169.990	42	86.210	268.4	177,355.2	147.4	7.0	41.5	177,551.1	2059.51	2059.53	0.75
C		169.990	169.998	15	30.775	237.6	63,356.2	52.7	4.4	−14.5	63,398.8	2060.07	2060.04	1.26
250-A	7/31/29	169.999	170.012	19	18.508	199.4	38,073.1	66.7	0.9	−26.0	38,114.7	2059.36	2059.36	0.58
251-A	8/ 1/29	170.008	170.008	30	29.370	377.6	60,363.3	96.0	−2.1	− 1.3	60,455.9	2058.42	2058.45	−0.33
B		170.008	170.008	30	29.351	348.2	60,336.6	96.0	0.0	1.5	60,434.1	2059.01	2059.04	0.26
C		170.008	170.000	30	29.342	318.9	60,296.5	96.0	−0.4	17.6	60,409.7	2058.81	2058.84	0.06
D		170.000	170.017	32	31.262	287.6	64,282.8	102.4	−0.1	−37.4	64,347.7	2058.34	2058.34	−0.44
E		170.017	170.000	30	29.328	258.3	60,242.8	96.0	3.5	38.1	60,380.4	2058.80	2058.85	0.07
252-A	8/ 2/29	170.036	170.018	30	30.465	375.8	62,563.2	105.3	−1.8	46.4	62,713.1	2058.53	2058.64	−0.14
B		170.018	170.004	30	30.445	345.3	62,528.0	105.3	3.6	35.1	62,672.0	2058.53	2058.59	−0.19
C		170.004	170.022	30	30.284	314.9	62,499.1	105.3	−3.1	−43.3	62,558.0	2058.91	2058.92	0.14
D		170.022	170.008	30	30.407	284.5	62,460.2	105.3	−2.4	30.9	62,594.0	2058.54	2058.61	−0.17
E		170.008	170.013	30	30.374	254.1	62,430.3	105.3	−0.6	−10.8	62,524.2	2058.48	2058.51	−0.27
F		170.013	170.014	30	30.372	223.8	62,403.8	105.3	2.1	− 1.6	62,509.6	2058.13	2058.17	−0.61
G		170.014	170.009	30	30.361	193.4	62,381.1	105.3	0.4	9.5	62,496.3	2058.44	2058.48	−0.30
H		170.009	170.008	33	33.382	160.0	68,596.3	115.8	3.1	1.7	68,716.9	2058.50	2058.53	−0.25
Even temperature 170°. Z_{170} = 1055 int. joules per deg. cent. α_{170} = 4.171 int. joules per gram deg. cent. Mean value of $\gamma_{170} = \gamma_m$ = 2058.78 int. joules per gram														

TABLE 8 PRINCIPAL DATA FROM γ EXPERIMENTS—*Continued*

Exp. No.	Date	Initial temp. θ_1	Final temp. θ_2	Total time of exp.	Mass of vapor removed ΔM	Mass of water remaining M_2	Total electric energy Q_E	Pump energy Q_P	Heat leak Q_L	Energy correction for final temp. diff. $-[Z + M_{12}\alpha]t^2$	Entire energy added γ_1	Entire energy per gram γ_1	$\gamma\theta$ Corrected to even temp.	Residual $\gamma\theta - \gamma_m$
253-A	8/ 3/29	200.023	200.030	15	30.034	274.1	58,767.8	46.5	0.3	−15.1	58,799.5	1957.76	1957.84	−0.12
B		200.030	200.026	20	40.055	230.0	78,318.4	62.0	2.6	8.2	78,391.2	1957.09	1957.20	−0.76
C		200.026	200.012	15	30.032	204.0	58,717.2	46.5	0.9	26.6	58,791.2	1957.62	1957.71	−0.25
254-A	8/ 5/29	200.000	200.008	31	32.908	311.9	64,374.5	96.1	1.0	−20.5	64,451.1	1958.52	1958.52	0.56
B		200.008	200.012	29	30.790	281.1	60,184.7	89.9	0.9	− 8.2	60,267.3	1957.37	1957.39	−0.57
C		200.012	200.008	30	31.823	249.3	62,233.0	93.0	0.3	9.3	62,335.6	1958.82	1958.86	0.90
D		200.008	199.997	30	31.836	217.4	62,204.2	93.0	0.6	21.4	62,319.2	1957.51	1957.53	−0.43
E		199.997	200.006	30	31.796	185.6	62,179.4	93.0	−0.1	−17.2	62,255.1	1957.95	1957.94	−0.02
255-A	8/ 7/29	199.991	199.978	30	31.486	349.4	61,526.4	93.0	−0.6	34.2	61,653.0	1958.11	1958.08	0.12
B		199.978	199.980	30	31.447	317.9	61,488.5	93.0	−1.3	− 3.9	61,576.3	1958.10	1958.02	0.06
C		199.980	199.979	30	31.426	286.5	61,448.6	93.0	−2.8	1.6	61,540.4	1958.26	1958.19	0.23
D		199.979	199.981	30	31.399	255.1	61,407.7	93.0	0.8	− 3.5	61,498.0	1958.60	1958.53	0.57
E		199.981	199.979	30	31.381	223.7	61,363.5	93.0	−1.0	− 2.9	61,452.6	1958.27	1958.20	0.24
F		199.979	199.984	10	10.454	213.2	20,444.5	30.4	−0.4	−10.4	20,464.1	1957.54	1957.47	−0.49

Even temperature 200°. Z_{200} = 1119 int. joules per deg. cent. α_{200} = 4.159 int. joules per gram deg. cent. Mean value of $\gamma_{200} = \gamma_m$ = 1957.96 int. joules per gram

256-A	8/ 8/29	219.974	219.986	30	31.248	359.1	58,783.8	91.2	5.9	−29.7	58,851.2	1883.36	1883.26	−0.16
B		219.986	220.001	30	31.214	327.9	58,734.7	91.2	5.9	−37.8	58,788.1	1883.39	1883.34	−0.08
C		220.001	220.004	31	32.226	295.7	60,626.3	94.2	0.5	− 7.4	60,713.6	1883.99	1883.99	0.57
D		220.004	219.995	30	31.174	264.5	58,606.6	91.2	−1.1	20.6	58,717.3	1883.53	1883.55	0.13
E		219.995	220.000	30	31.122	233.4	58,552.7	91.2	−1.7	−12.2	58,630.0	1883.85	1883.86	0.44
F		220.000	220.001	30	31.110	202.3	58,493.0	91.2	−0.1	− 0.8	58,583.3	1883.10	1883.10	−0.32
G		220.001	219.992	32	33.157	169.1	62,347.1	97.3	0.5	15.7	62,460.6	1883.78	1883.78	0.36
257-A	8/ 9/29	218.207	218.208	30	30.958	315.9	58,433.7	91.2	1.6	− 2.2	58,524.3	1890.44	1883.50	0.08
B		218.208	218.213	31	31.964	283.9	60,337.6	94.2	0.5	−11.5	60,420.8	1890.28	1883.35	−0.07
C		218.213	218.204	30	30.926	253.0	58,339.1	91.2	0.8	19.7	58,450.8	1890.02	1883.10	−0.32
D		218.204	218.202	30	30.880	221.1	58,280.1	91.2	0.1	5.0	58,376.4	1890.42	1883.46	0.04
257-E	8/ 9/29	218.202	218.218	30	30.836	191.3	58,229.9	91.2	2.3	−31.2	58,292.2	1890.39	1883.43	0.01
F		218.218	218.214	19	19.524	171.7	36,852.3	57.8	2.1	6.2	36,918.4	1890.92	1883.98	0.56
G		218.214	218.206	13	13.355	158.4	25,203.0	39.5	2.4	15.3	25,260.2	1891.44	1884.48	1.06
258-A	8/10/29	220.010	220.000	15	29.149	345.0	54,820.2	45.6	0.5	25.7	54,892.0	1883.15	1883.19	−0.23
B		220.000	220.011	15	29.127	315.9	54,811.7	45.6	2.5	−25.5	54,834.3	1882.59	1882.59	−0.83
C		220.011	220.015	15	29.119	286.8	54,801.1	45.6	1.7	−11.1	54,837.3	1883.21	1883.25	−0.17
D		220.015	220.016	15	29.116	257.7	54,794.8	45.6	1.3	− 1.8	54,839.9	1883.50	1883.56	0.14
E		220.016	220.011	16	31.067	226.6	58,442.9	48.6	2.4	10.7	58,504.6	1883.18	1883.24	−0.18
F		220.011	220.010	15	29.119	197.5	54,786.0	45.6	1.6	2.4	54,835.6	1883.16	1883.20	−0.22
G		220.010	220.009	15	29.104	168.4	54,786.4	45.6	2.0	0.9	54,834.9	1884.10	1884.14	0.72
259-A	8/12/29	220.008	220.005	60	30.388	373.2	57,044.4	182.4	−2.2	7.3	57,231.9	1883.37	1883.40	−0.02
B		220.005	219.999	60	30.429	342.7	57,037.1	182.4	1.9	14.0	57,235.4	1880.95	1880.97	−2.45
C		219.999	220.012	60	30.357	312.4	57,030.3	182.4	4.0	−30.8	57,185.9	1883.78	1883.78	0.36
D		220.012	220.018	60	30.365	282.0	57,020.8	182.4	3.4	−13.3	57,193.3	1883.53	1883.58	0.16
E		220.021	220.005	60	30.366	220.1	56,986.8	182.4	−1.6	34.7	57,202.3	1883.76	1883.84	0.42

Even temperature 220°. Z_{220} = 1169 int. joules per deg. cent. α_{220} = 4.138 int. joules per gram deg. cent. Mean value of $\gamma_{220} = \gamma_m$ = 1883.42 int. joules per gram

265-A	10/23/29	250.005	250.023	30	33.462	322.6	58,818.2	96.9	− 9.4	−45.8	58,859.9	1759.01	1759.03	0.15
B		250.023	250.011	31	34.610	288.0	60,760.7	100.1	−10.8	27.6	60,877.6	1758.96	1759.06	0.18
C		250.016	250.004	31	34.594	219.9	60,717.7	100.1	− 7.8	27.7	60,837.7	1758.62	1758.69	−0.19
266-A	10/24/29	250.027	249.999	32	34.514	329.6	60,534.2	103.4	−12.3	74.0	60,699.3	1758.69	1758.81	−0.07
B		250.034	250.037	30	32.227	240.4	56,592.1	96.9	− 6.8	− 5.8	56,676.4	1758.66	1758.81	−0.07

Even temperature 250°. Z_{250} = 1259 int. joules per deg. cent. α_{250} = 4.120 int. joules per gram deg. cent. Mean value of $\gamma_{250} = \gamma_m$ = 1758.88 int. joules per gram

269-A	11/12/29	269.968	269.996	28	29.112	341.4	48,513.1	88.5	− 1.8	−76.0	48,523.8	1666.80	1666.64	1.43
B		269.996	269.950	30	31.323	310.1	51,958.1	94.8	− 8.0	117.9	52,162.8	1665.32	1665.30	0.09
C		269.950	269.985	30	31.225	278.9	51,943.4	94.8	− 3.0	−85.2	51,950.0	1663.73	1663.48	−1.73
D		269.985	269.975	30	31.221	247.7	51,925.4	94.8	− 5.3	22.7	52,037.6	1666.75	1666.68	1.47
E		269.975	269.997	30	31.189	216.5	51,909.5	94.8	− 8.0	−49.0	51,947.3	1665.56	1665.44	0.23
270-A	11/13/29	270.025	270.039	30	30.623	329.8	50,949.0	94.8	− 8.7	−37.5	50,997.6	1665.34	1665.46	0.25
B		270.033	270.010	30	30.667	299.1	50,922.7	94.8	−10.4	74.8	51,081.9	1665.70	1665.89	0.68
C		270.010	270.035	30	30.639	268.5	50,908.5	94.8	−12.5	−60.7	50,930.1	1662.26	1662.31	−2.90
D		270.035	270.016	30	30.616	237.9	50,902.1	94.8	−14.7	41.7	51,023.9	1666.58	1666.75	1.54
E		269.998	270.020	30	30.619	186.8	50,894.9	94.8	−11.2	−44.2	50,934.3	1663.49	1663.48	−1.73
F		270.020	269.974	20	20.433	166.4	33,917.8	63.2	− 9.7	92.0	34,063.3	1667.07	1667.17	1.96
271-A	11/14/29	270.045	270.006	31	32.365	350.9	53,676.4	98.0	− 2.2	108.4	53,880.6	1664.78	1665.00	−0.21
B		270.006	270.012	30	31.250	319.7	51,925.1	94.8	6.9	−17.1	52,009.7	1664.31	1664.34	−0.87
C		270.012	270.006	30	31.264	288.4	51,904.3	94.8	−10.6	15.1	52,003.6	1663.37	1663.43	−1.78
D		270.006	270.002	31	32.257	256.2	53,618.4	98.0	5.1	10.9	53,732.4	1665.76	1665.79	0.58
E		270.002	269.999	30	31.188	224.9	51,868.6	94.8	− 8.1	6.1	51,961.4	1666.07	1666.08	0.87
272-A	11/15/29	270.047	270.036	60	31.410	364.3	52,090.9	189.6	−12.0	30.5	52,299.0	1665.04	1665.27	0.06
B		270.036	269.995	60	31.433	332.8	52,016.2	189.6	− 1.2	110.6	52,315.2	1664.34	1664.52	−0.69
C		270.039	270.013	43	22.441	279.0	37,223.4	135.9	−18.4	70.0	37,410.9	1667.07	1667.26	2.05
273-A	11/16/29	270.013	270.005	23	39.359	319.4	65,401.9	72.7	− 2.2	17.9	65,490.3	1663.92	1663.98	−1.23
B		270 005	269.999	22	37.604	281.8	62,528.8	69.5	− 3.7	14.9	52,609.5	1664.97	1664.99	−0.22
C		269 999	269.996	20	34.120	247.6	56,772.0	63.2	− 2.7	8.2	56,840.7	1665.91	1665.91	0.70
D		269.996	270.014	20	34.104	213.5	56,738.4	63.2	− 1.9	−41.2	56,758.5	1664.28	1664.26	−0.95
E		270.014	270.006	16	27 288	186.2	45,388.8	50.6	− 2.7	17.1	45,453.8	1665.71	1665.78	0.57
F		270.006	270.021	15	25.572	160.7	42,561.2	47.4	− 0.3	−28.6	42,579.7	1665.09	1665.11	−0.10

Even temperature 270°. Z_{270} = 1328 int. joules per deg. cent. α_{270} = 4.100 int. joules per gram deg. cent. Mean value of $\gamma_{270} = \gamma_m$ = 1665.21 int. joules per gram

for variation in the temperature of steam withdrawn. This correction was found to be in every case less than 0.00001 of the whole, and has been omitted from the final calculation of γ.

The principal data from the gamma experiments are assembled in Table 8 and the computation of γ completed for the entire series.

3—Beta Experiments

An example of the record of a single determination of β, equal to $L\frac{u}{u'-u}$, is given in Table 9. This is similar to the record for a gamma experiment, but differs from it in that the total time of the experiment over which the energy account is kept is longer than the time of electric heating and removal of liquid.

Table 10 shows the calculation of β. This calculation makes use of the following formula:

$$\beta_1 = \frac{1}{\Delta M}(Q_R + Q_P + Q_L - [Z + M_2\alpha]_1^2) + \frac{\Delta H}{\Delta\theta}(\theta_1 - \theta_m)$$

The principal data from the β experiments are assembled in Table 11 and the computation of β completed for the entire series.

VII—FORMULATION

1—Formulation of Calorimetric Data

In what has gone before we have the record of the experiments and reduction of the data yielding the values of the three characteristic properties of the fluid, namely:

$$H - L\frac{u}{u'-u}, \qquad L\frac{u'}{u'-u}, \qquad \text{and } L\frac{u}{u'-u}$$

denoted by the symbols α, γ, and β, respectively. As a first step in the formulation of the more familiar and more useful thermal properties of steam for engineering purposes, the experimental results are assembled, the values smoothed by means of empirical equations, and the blank spaces filled in by interpolation or by extension, using other experimental data where necessary. This assembly and formulation is given in Table 12. The empirical equations used are also given there. It has not yet been found possible to obtain empirical equations of a simpler form to represent the experimental results to an accuracy warranted by the experimental precision. There is a possibility that a simple formula for expressing the thermal behavior of steam may eventually be discovered. Such a formula would be extremely useful, particularly so if of a form well adapted to thermodynamic calculations and if in agreement with physical facts. In the meanwhile, equations lacking this ideal simplicity of form may be tolerated in the operations of obtaining numerical tabulations of thermal properties which result from experiment. In the present instance fidelity to observation has not been sacrificed to superficial simplicity.

Two formulas have been used for α, one of which fits the results in the range from 0 deg. to 170 deg. cent., and the other in the range from 170 deg. to 270 deg. cent. The formula for β fits the results in the range from 100 deg. to 270 deg. cent. The values of β in the range from 0 deg. to 100 deg. cent. have been computed from previously published vapor-pressure and liquid-density data.

The experimental values of L have been obtained from the observed values of γ and the smoothed or calculated values of β, using the relation $L = \gamma - \beta$. The empirical formula for L fits the experimental results to better than one part in 2300. The form of this equation has been chosen in accordance with the usually accepted belief that at the critical temperature (374 deg. cent.) the heat of vaporization becomes 0 and approaches this value at a negative infinite rate, and that no real values exist at higher temperatures. Below 100 deg. cent. the values given by the equation are in accord with results of other observers.

TABLE 9 SAMPLE DATA SHEET—DETERMINATION OF $L\frac{u}{u'-u}$

Experiment No. 207-C — Feb. 21, 1929
at 240 Deg. Cent.
Observers: N. S. O., H. F. S., E. F. F.

Time	Thermometers, bridge readings ohms $2n$	$2r$	Thermoelements t.e. 1–5 regulating microvolts		Electric power I amperes	E volts	
2:28:00	49.0813	.....	6.1	...	.....	....	
:10	14	.....	6.1	...	.....	....	
:20	15	.....	6.2	...	.....	....	
:30	.....	.....	...	(1)*	.....	....	
:40	.....	49.0878	6.1	...	.....	....	
:50	.....	80	6.2	...	.....	....	
2:29:00	.....	79	6.0	...	.....	....	On
:30	49.0814	.....	7	(1)	.....	....	
:30:00	.....	49.0877	8	(2)	0.97450	....	
:30	11	.....	5	(0)	.....	....	
:31:00	.....	73	5	1	.....	9.9160	
:30	.....	.....	6	(0)	.....	....	
:32:00	.....	71	6	(0)	0.97450	....	
:30	07	.....	4	(−1)	.....	....	
:33:00	.....	70	5	(0)	.....	9.9164	Off
:30	49.0806	.....	2.2	...	.....	...	
:40	05	.....	2.0	...	.....	....	
:50	06	.....	2.1	...	.....	....	
:34:00	.....	.....	...	(−2)	.....	....	
:10	.....	49.0870	2.0	...	.....	....	
:20	.....	71	2.1	...	.....	....	
:30	.....	70	1.9	...	.....	....	

Total heat leak factor = +3 microvolt-minutes
Total time of experiment, 2:28:30 to 2:34:00, = 330 seconds
Total time of electric heating, 2:29:00 to 2:33:00, = 240 seconds

* Values in parentheses calculated from indications of t.e. 1–5.

TABLE 10 SAMPLE COMPUTATION—DETERMINATION OF $L\frac{u}{u'-u}$

Experiment No. 207-C

	Initial	Final
Temperature		
$(n + r)/2$ on bridge	49.08465	49.08380
Bridge correction	0.00969	0.00969
R_θ	49.09434	40.09349
$R_\theta - R_0$	23.48654	23.48569
$100(R_\theta - R_0)/(R_{100} - R_0)$	235.0439	235.0354
$0.01\theta(0.01\theta - 1)\delta$	4.9774	4.9768
Reference block temperature	240.0213	240.0122
Mean t.e. 1–5 (microvolts) 6.12	2.05	
Mean t.e. 1–5 (degrees)	0.0203	0.0068
Temperature of calorimeter	240.0010	240.0054

Calculation of Energy

Potentiometer	Current		Voltage
Mean reading	0.97452	Mean reading	9.9162
Potentiometer correction	2	Potentiometer correction	2
Correction for standard 0.01 ohm	−5	Correction for volt-box factor	33
Correction for volt-box current	−30	Voltage = E =	9.9197
Current = I =	0.97419		

Time of electric heating = 240 seconds
Total electric energy = IEi = 2319.3 joules
Total time of experiment = 330 seconds
Pump power = 0.0748 watts
Pump energy = 24.7
Total heat leak factor = +3 microvolt-minutes
Heat-leak coefficient = 0.142 joules per microvolt-minute
Heat leak = 0.4
Change in calorimeter temp. = + 0.0044 deg.
Factor for calorimeter temp. change = 2949 joules per degree
Energy for calorimeter temp. change = −11.8

Total energy supplied = 2332.6 joules
Mass withdrawn = 62.722 grams
Energy per gram = 37.190 joules per gram
Mean bridge reading during experiments = 48.08421
Mean ref. block temp. during experiment = 240.0169 deg.
Mean t.e. 1–5 during experiment = 5.78 microvolts = 0.0192 deg.
Mean calorimeter temperature during experiment = 239.9977 deg.
Heat content at initial minus heat content at mean $(H_1 - H_m)$ = 0.016 joules per gram

$L\frac{u}{u'-u}$ at 240.0010 deg. = 37.206 joules per gram

TABLE 11 PRINCIPAL DATA FROM β EXPERIMENTS

$$\beta_1 = \frac{1}{\Delta M}\left[Q_E + Q_P + Q_L - [Z + M_2\alpha]_1^2\right] + \frac{\Delta H}{\Delta\theta}(\theta_1 - \dot{\theta}_m)$$

Exp. No.	Date	Initial temp. θ_1 Degrees centigrade	Final temp. θ_2 centigrade	Total time of exp. Minutes	Mass of water removed ΔM Grams	Mass of water remaining M_2 Grams	Total electric energy Q_E International joules	Pump energy Q_P International joules	Heat leak Q_L International joules	Energy correction for final temp. diff. $-[Z + M_2\alpha]_1^2$ International joules	Total energy added International joules	Energy added per gram removed International joules per gram	$\theta_1 - \theta_m$, Deg. cent.	Corr. for temp. variation, $\frac{\Delta H}{\Delta\theta}(\theta_1 - \theta_m)$ International joules per gram	Corr. to even temp. International joules per gram	β Corrected to even temp. International joules per gram
180-A	12/28/28	100.000	100.007	19.000	189.123	230.5	178.6	100.3	−1.4	−13.4	264.1	1.396	−0.002	−0.008	0.000	1.388
182-A	12/31/28	100.019	100.018	14.000	140.217	278.3	121.8	73.9	−1.5	2.1	196.3	1.400	0.002	0.008	−0.001	1.407
204-A	2/18/29	99.827	99.830	9.333	118.829	301.9	132.3	49.3	−1.3	− 6.6	173.7	1.462	−0.009	−0.038	0.008	1.432
Even temperature 100°. $\frac{\Delta H}{\Delta\theta}$ = 4.22 int. joules per gram deg. cent. Z_{100} = 956 int. joules per deg. cent. α_{100} = 4.169 int. joules per gram deg. cent.																
Mean value of β_{100} = 1.409 int. joules per gram																
207-A	2/21/29	110.004	109.997	3.500	69.719	352.0	102.2	18.3	−0.8	15.3	135.0	1.936	0.001	0.004	0.000	1.941
B		110.060	110.047	3.500	77.187	274.8	102.6	18.3	−0.3	27.9	148.5	1.924	0.009	0.038	−0.004	1.958
Even temperature 110°. $\frac{\Delta H}{\Delta\theta}$ = 4.23 int. joules per gram deg. cent. Z_{110} = 967 int. joules per deg. cent. α_{110} = 4.168 int. joules per gram deg. cent.																
Mean value of β_{110} = 1.950 int. joules per gram.																
199-A	2/12/29	120.011	120.016	14.333	130.590	298.1	279.4	74.2	−2.1	−11.1	340.4	2.607	0.003	0.013	−0.001	2.623
B		120.093	120.115	14.333	120.155	303.0	294.2	74.2	0.3	−49.3	319.4	2.658	−0.014	−0.060	−0.007	2.591
203-A	2/16/29	120.044	120.044	9.333	77.736	346.1	154.0	48.3	−0.7	0.0	201.6	2.593	0.012	0.051	−0.004	2.640
B		120.114	120.113	10.333	80.539	265.6	154.9	53.5	−0.4	2.1	210.1	2.609	−0.001	−0.004	−0.009	2.596
Even temperature 120°. $\frac{\Delta H}{\Delta\theta}$ = 4.25 int. joules per gram deg. cent. Z_{120} = 979 int. joules per deg. cent. α_{120} = 4.168 int. joules per gram deg. cent.																
Mean value of β_{120} = 2.613 int. joules per gram																
205-A	2/19/29	129.991	130.006	9.333	57.631	363.8	195.3	47.8	−0.2	−38.6	204.3	3.545	−0.009	−0.038	0.001	3.508
B		130.067	130.059	8.667	80.403	283.4	220.5	44.4	−1.3	18.9	282.5	3.514	0.004	0.017	−0.007	3.524
206-A	2/20/29	130.016	130.050	3.500	75.196	341.1	215.3	17.9	−0.8	25.6	258.0	3.431	0.008	0.034	−0.002	3.463
B		130.071	130.075	3.500	64.235	281.9	215.5	17.9	0.0	− 9.8	223.6	3.481	−0.004	−0.017	−0.007	3.457
Even temperature 130°. $\frac{\Delta H}{\Delta\theta}$ = 4.27 int. joules per gram deg. cent. Z_{130} = 992 int. joules per deg. cent. α_{130} = 4.169 int. joules per gram deg. cent.																
Mean value of β_{130} = 3.488 int. joules per gram																
200-A	2/13/29	139.964	139.969	8.333	58.429	370.5	239.3	42.2	0.3	−12.8	269.0	4.604	0.001	0.004	0.004	4.612
201-A	2/14/29	140.050	140.058	7.333	196.732	228.5	876.5	37.2	−0.3	−15.7	897.7	4.563	−0.004	−0.017	−0.006	4.540
202-A	2/15/29	140.011	140.009	9.333	79.570	339.3	309.4	47.3	−1.1	4.8	360.4	4.529	0.003	0.013	−0.001	4.541
B		140.097	140.105	9.333	73.166	266.1	309.5	47.3	−1.1	−16.9	338.8	4.631	−0.008	−0.034	−0.012	4.585
Even temperature 140°. $\frac{\Delta H}{\Delta\theta}$ = 4.29 int. joules per gram deg. cent. Z_{140} = 1003 int. joules per deg. cent. α_{140} = 4.171 int. joules per gram deg. cent.																
Mean value of β_{140} = 4.570 int. joules per gram																
177-A	12/21/28	150.101	150.106	19.000	152.916	261.7	824.4	95.4	0.0	−10.6	909.2	5.946	0.000	0.000	−0.015	5.931
198-A	2/ 9/29	149.913	149.918	14.333	135.999	285.7	745.8	72.0	−0.8	−11.1	805.9	5.926	−0.003	−0.013	0.013	5.926
209-A	2/25/29	149.986	149.979	5.500	56.233	365.2	285.8	27.6	−0.6	16.5	329.3	5.856	−0.002	−0.009	0.002	5.849
B		150.046	150.051	5.500	58.587	306.6	329.9	27.6	−0.2	−10.6	346.7	5.918	−0.002	−0.009	−0.007	5.902
Even temperature 150°. $\frac{\Delta H}{\Delta\theta}$ = 4.32 int. joules per gram deg. cent. Z_{150} = 1021 int. joules per deg. cent. α_{150} = 4.171 int. joules per gram deg. cent.																
Mean value of β_{150} = 5.902 int. joules per gram																
211-A	2/27/29	159.936	159.938	4.500	66.586	359.7	482.2	22.4	−0.4	− 4.6	499.6	7.503	−0.002	−0.009	−0.012	7.506
B		159.997	159.996	4.500	72.729	287.1	521.9	22.4	−0.4	2.0	545.9	7.506	0.001	0.004	−0.001	7.509
Even temperature 160°. $\frac{\Delta H}{\Delta\theta}$ = 4.35 int. joules per gram deg. cent. Z_{160} = 1037 int. joules per deg. cent. α_{160} = 4.171 int. joules per gram deg. cent.																
Mean value of β_{160} = 7.508 int. joules per gram																
203-B	2/16/29	169.990	169.993	9.333	61.854	356.2	548.4	45.9	−0.6	− 7.6	586.1	9.476	−0.001	−0.004	0.002	9.474
C		170.053	170.041	9.333	71.437	284.7	606.3	45.9	−1.1	26.9	678.0	9.491	0.007	0.031	−0.011	9.511
Even temperature 170°. $\frac{\Delta H}{\Delta\theta}$ = 4.38 int. joules per gram deg. cent. Z_{170} = 1055 int. joules per deg. cent. α_{170} = 4.171 int. joules per gram deg. cent.																
Mean value of β_{170} = 9.492 int. joules per gram																
168 A	11/19/28	200.232	200.233	24.000	128.131	296.4	2184.0	114.5	0.8	− 2.3	2297.0	17.927	−0.004	−0.018	−0.081	17.828
171-A	11/22/28	200.774	200.782	22.000	140.844	285.5	2466.3	104.9	2.1	−18.4	2554.9	18.140	−0.001	−0.005	−0.272	17.863
173-A	12/17/28	200.632	200.641	20.000	121.718	292.9	2128.1	95.4	0.8	−21.1	2203.2	18.101	−0.007	−0.032	−0.222	17.847
176-A	12/20/28	200.215	200.230	19.000	134.847	279.4	2355.7	90.6	4.8	−34.2	2416.9	17.923	0.000	0.000	−0.075	17.848
178-A	12/26/28	200.148	200.157	20.000	132.637	288.2	2282.3	95.4	2.6	−20.8	2359.5	17.789	0.000	0.000	−0.052	17.737
183-A	1/ 2/29	200.172	200.180	18.625	133.674	286.4	2313.6	88.8	−4.1	−18.4	2379.9	17.804	0.005	0.023	−0.060	17.767
184-A	1/ 3/29	200.147	200.150	19.333	133.349	285.6	2299.6	92.2	−1.9	− 6.9	2383.0	17.870	0.003	0.014	−0.052	17.832
190-A	1/23/29	200.152	200.163	19.000	139.456	277.2	2447.7	64.8	−0.1	−26.1	2486.3	17.829	0.007	0.032	−0.053	17.808
191-A	1/24/29	200.250	200.255	19.333	140.607	279.0	2436.5	92.2	0.8	−11.4	2518.1	17.909	0.003	0.014	−0.088	17.835
196-A	2/ 1/29	200.050	200.049	20.000	134.792	287.0	2309.1	95.4	−1.3	2.3	2405.5	17.846	0.003	0.014	−0.018	17.842
Even temperature 200°. $\frac{\Delta H}{\Delta\theta}$ = 4.51 int. joules per gram deg. cent. Z_{200} = 1119 int. joules per deg. cent. α_{200} = 4.159 int. joules per gram deg. cent.																
Mean value of β_{200} = 17.821 int. joules per gram																
204-B	2/18/29	219.920	219.923	14.000	134.034	284.4	3424.8	65.4	0.5	− 7.0	3483.7	25.991	0.001	0.005	0.038	26.034
205-C	2/19/29	220.050	220.050	6.667	141.309	273.6	3647.9	31.1	−0.1	0.7	3679.6	26.039	0.004	0.018	−0.024	26.033
Even temperature 220°. $\frac{\Delta H}{\Delta\theta}$ = 4.62 int. joules per gram deg. cent. Z_{220} = 1169 int. joules per deg. cent. α_{220} = 4.138 int. joules per gram deg. cent.																
Mean value of β_{220} = 26.034 int. joules per gram																
206-C	2/20/29	229.927	229.940	4.000	57.261	359.0	1803.9	18.5	−0.7	−35.9	1785.8	31.187	−0.001	−0.005	0.039	31.221
D		229.992	229.975	3.500	61.843	297.1	1867.7	16.2	0.1	42.2	1926.2	31.147	0.010	0.047	0.004	31.198
Even temperature 230°. $\frac{\Delta H}{\Delta\theta}$ = 4.68 int. joules per gram deg. cent. Z_{230} = 1197 int. joules per deg. cent. α_{230} = 4.134 int. joules per gram deg. cent.																
Mean value of β_{230} = 31.210 int. joules per gram																
192-A	1/25/29	239.986	240.007	15.333	62.817	356.8	2323.1	70.1	1.1	−56.7	2337.6	37.213	−0.007	−0.033	0.009	37.189
B		240.089	240.107	14.333	68.444	288.4	2525.0	65.5	1.9	−45.8	2546.6	37.207	−0.007	−0.033	−0.055	37.119
207-C	2/21/29	240.001	240.005	5.500	62.713	354.9	2319.3	25.1	0.4	−11.8	2333.0	37.201	0.003	0.014	−0.001	37.214
D		240.054	240.063	5.500	65.386	289.5	2425.4	25.1	−0.2	−22.7	2427.6	37.127	0.013	0.062	−0.033	37.156
Even temperature 240°. $\frac{\Delta H}{\Delta\theta}$ = 4.76 int. joules per gram deg. cent. Z_{240} = 1227 int. joules per deg. cent. α_{240} = 4.129 int. joules per gram deg. cent.																
Mean value of β_{240} = 37.170 int. joules per gram																

TABLE 11 PRINCIPAL DATA FROM β EXPERIMENTS—*Continued*

Exp. No.	Date	Initial temp. θ_1	Final temp. θ_2	Total time of exp.	Mass of water removed ΔM	Mass of water remaining M_3	Total electric energy, Q_E	Pump energy Q_P	Heat leak Q_L	Energy correction for final temp. diff. $-\int Z + M_{3\sigma}]_1^2$	Total energy added	Energy added per gram removed	$\theta_1 - \theta_m$	Corr. for temp. variation $\frac{\Delta H}{\Delta\theta}(\theta_1 - \theta_m)$	Corr. to even temp.	β corrected to even temp. per gram
		Degrees centigrade		Minutes	Grams		International joules						Deg. cent.	International joules		
208-A	2/23/29	249.763	249.779	5.500	65.034	355.5	2865.8	24.9	0.6	−44.1	2847.2	43.780	−0.002	−0.010	0.169	43.939
B		249.839	249.831	5.500	67.912	287.6	2928.6	24.9	0.9	19.3	2973.7	43.788	0.009	0.044	0.115	43.947
Even temperature 250°. $\frac{\Delta H}{\Delta\theta}$ = 4.85 int. joules per gram deg. cent. Z_{250} = 1259 int. joules per deg. cent. α_{250} = 4.120 int. joules per gram deg. cent.																
Mean value of β_{250} = 43.943 int. joules per gram																
172-A	11/23/28	270.190	270.211	33.000	140.721	285.4	8523.8	145.9	2.8	−52.5	8620.0	61.256	−0.009	−0.046	−0.189	61.021
175-A	12/19/28	270.232	270.251	25.000	139.687	274.6	8477.8	110.5	8.4	−46.7	8550.0	61.208	0.000	0.000	−0.231	60.977
177-B	12/21/29	270.200	270.192	25.000	137.092	276.2	8235.5	110.5	5.3	19.7	8371.0	61.061	−0.004	−0.020	−0.199	60.842
181-A	12/29/28	269.974	269.983	25.000	142.219	277.4	8564.9	110.5	2.3	−22.2	8655.5	60.860	0.011	0.056	0.026	60.927
185-A	1/ 4/29	270.598	270.597	10.333	34.207	384.7	2045.8	45.7	−0.8	2.9	2093.6	61.204	0.003	0.015	−0.595	60.624
187-A	1/18/29	270.347	270.295	11.000	29.485	385.3	1598.9	48.6	−2.2	151.4	1796.7	60.936	0.034	0.173	−0.345	60.764
B		270.387	270.371	9.000	22.994	362.3	1332.4	39.8	2.2	45.1	1419.5	61.733	0.021	0.108	−0.385	61.456
188-A	1/19/29	270.353	270.351	8.000	32.901	329.4	1984.2	35.4	1.2	5.4	2026.2	61.585	0.003	0.015	−0.351	61.249
B		270.441	270.453	8.000	31.475	298.0	1936.3	35.4	3.1	−30.6	1944.2	61.770	0.003	0.015	−0.439	61.346
189-A	1/22/29	270.077	270.075	25.000	135.841	284.5	8196.7	79.0	10.3	− 5.0	8281.0	60.961	0.005	0.025	−0.077	60.909
195-A	1/31/29	270.009	270.016	25.333	137.263	284.3	8268.4	112.0	1.7	−17.5	8364.6	60.938	−0.002	−0.010	−0.009	60.919
Even temperature 270°. $\frac{\Delta H}{\Delta\theta}$ = 5.09 int. joules per gram deg. cent. Z_{270} = 1328 int. joules per deg. cent. α_{270} = 4.100 int. joules per gram deg. cent.																
Mean value of β_{270} = 61.003 int. joules per gram																

TABLE 12 ASSEMBLY OF OBSERVED CALORIMETRIC DATA AND COMPARISON WITH FORMULATION

Temp., deg. cent.	$\alpha]_0^\theta$ Observed	$\alpha]_0^\theta$ Calculated	$\alpha]_0^\theta$ Obs.−Calc.	β_θ Observed	β_θ Calculated	β_θ Obs.−Calc.	γ_θ Observed	γ_θ Calculated	γ_θ Obs.−Calc.	L_θ Observed	L_θ Calculated	L_θ Obs.−Calc.
	International joules per gram											
0	.....	.....	....	.....	(0.016)[a]	.....	.....	2494.04	....	.....	2494.02	....
10	41.99	42.01	−0.02	....	(0.023)	.....	.....	2472.28	....	.....	2472.26	....
20	83.80	83.80	0.00	....	(0.042)	.....	.....	2450.21	....	.....	2450.17	....
30	125.51	125.53	−0.02	....	(0.074)	.....	.....	2427.80	....	.....	2427.73	....
40	167.22	167.23	−0.01	....	(0.124)	.....	.....	2405.02	....	.....	2404.90	....
50	208.92	208.92	0.00	....	(0.200)	.....	.....	2381.84	....	.....	2381.64	....
60	250.61	250.61	0.00	....	(0.312)	.....	.....	2358.22	....	.....	2357.91	....
70	292.30	292.29	0.01	....	(0.473)	.....	.....	2334.12	....	.....	2333.65	....
80	333.99	333.98	0.01	....	(0.697)	.....	.....	2309.52	....	.....	2308.82	....
90	375.66	375.66	0.00	....	(1.002)	.....	.....	2284.38	....	.....	2283.38	....
100	417.36	417.36	0.00	1.409	1.408	0.001	2258.71	2258.65	0.06	2257.30	2257.24	0.06
110	459.04	459.05	−0.01	1.950	1.939	0.011	.....	2232.29	....	.....	2230.35	....
120	500.72	500.75	−0.03	2.613	2.621	−0.008	.....	2205.27	....	.....	2202.65	....
130	542.39	542.46	−0.07	3.488	3.484	0.004	2177.15	2177.52	−0.37	2173.67	2174.04	−0.37
140	584.09	584.16	−0.07	4.570	4.563	0.007	.....	2149.00	....	.....	2144.44	....
150	625.81	625.87	−0.06	5.902	5.891	0.011	2119.64	2119.65	−0.01	2113.75	2113.76	−0.01
160	667.51	667.57	−0.06	7.508	7.510	−0.002	.....	2089.40	....	.....	2081.89	....
170	709.22	709.21	0.01	9.492	9.464	0.028	2058.78	2058.18	0.60	2049.32	2048.72	0.60
180	750.92	750.94	−0.02	....	11.799	.....	.....	2025.90	....	.....	2014.10	....
190	792.58	792.60	−0.02	....	14.569	.....	.....	1992.46	....	.....	1977.89	....
200	834.23	834.20	0.03	17.821	17.831	−0.010	1957.96	1957.76	0.20	1940.13	1939.93	0.20
210	875.75	875.72	0.03	.....	21.649	.....	.....	1921.65	....	.....	1900.00	....
220	917.16	917.17	−0.01	26.034	26.090	−0.056	1883.42	1883.98	−0.56	1857.33	1857.89	−0.56
230	958.51	958.54	−0.03	31.210	31.228	−0.018	.....	1844.56	....	.....	1813.33	....
240	999.83	999.84	−0.01	37.170	37.144	0.026	.....	1802.16	....	.....	1766.02	....
250	1041.09	1041.06	0.03	43.943	43.925	0.018	1758.88	1759.51	−0.63	1714.95	1715.59	−0.64
260	1082.23	1082.22	0.01	....	51.664	.....	.....	1713.26	....	.....	1661.60	....
270	1123.31	1123.32	−0.01	61.003	61.014	−0.011	1665.21	1664.52	0.69	1604.20	1603.51	0.69

[a] Values in parentheses are calculated from previously published vapor-pressure and specific-volume data.

$$\alpha = 1.240777 \log_{10}\frac{\theta+7}{7} + 415.2412\left(\frac{\theta}{100}\right) + 0.6912\left(\frac{\theta}{100}\right)^2 - 0.044874\left(\frac{\theta}{100}\right)^4. \qquad \text{(Range 0–170°)}$$

$$\alpha = 412.91945\left(\frac{\theta}{100}\right) + 3.844\left(\frac{\theta}{100}\right)^2 - 0.5238\left(\frac{\theta}{100}\right)^4 + 0.0213\left(\frac{\theta}{100}\right)^6. \qquad \text{(Range 170–270°)}$$

$$\frac{100\,\theta}{\theta+273.1} = \left(\frac{\theta}{100}\right)^{10/2}\left[0.304065 + 0.0193\left(\frac{\theta}{100}\right) + 0.00662\left(\frac{\theta}{100}\right)^2\right] + 0.04830. \qquad \text{(Range 100–270°)}$$

$$L = 0.002562(374-\theta)^3 - 5.883(374-\theta) + 249.9(374-\theta)^{\frac{1}{2}} - 113(374-\theta)^{\frac{1}{4}}. \qquad \text{(Range 0–270°)}$$

The calculated values of γ are now obtained from the calculated values of L and β, completing the formulation of the properties measured calorimetrically.

2—Derivation of Heat Content and Entropy From Calorimetric Data

It is now possible to complete the formulation of heat content and entropy, using the smoothed and extended values of the calorimetric data. The results of these derivations are given in Table 13. The values of heat content of saturated liquid water, H, given in column 2 are obtained as the sums of the values of $\alpha]_0^\theta$ and of $\beta]_0^\theta$, calculated, in Table 12. The values of heat content of saturated vapor H' are equal to $H + L$. The calculation of entropy of the saturated liquid has been made using each of the two formulas:

$$\Phi = \int \frac{\alpha d\Theta}{\Theta^2} + \frac{H}{\Theta} = \int \frac{d\alpha}{\Theta} + \frac{\beta}{\Theta}$$

together with the two empirical equations for α, suited to the two ranges of temperature. Thus a check was obtained on the accuracy of the computations. These are rather laborious owing to the complicated form of the α equations. The values of entropy, heat content, α, and β are mutually consistent. The values of entropy of the saturated vapor are obtained by adding L/Θ to Φ, which makes these values consistent with the values of L.

3—Discussion of Accuracy

To the user of experimental data consisting of numerical values of physical quantities, it is important to know how trustworthy the figures are. In compiling a table of the properties of steam, over a large range of conditions, from various experimental sources, the choice of the most truthful figures to take as the basis for formulation must be based on a consideration of accuracy of the experimental results.

It is not a simple problem to make an estimate of the accuracy

TABLE 13 THERMAL PROPERTIES OF SATURATED WATER AND STEAM DERIVED FROM CALORIMETRIC MEASUREMENTS

Bureau of Standards, February, 1930

Temp., deg. cent.	Heat content of liquid H	Latent heat L	Heat content of vapor H'	Entropy of liquid Φ	Entropy of vapor Φ'
	Int. joules per gram			Int. joules per gm. °C.	
0	0	2494.02	2494.02	0	9.132
10	42.02	2472.26	2514.28	0.1511	8.884
20	83.83	2450.17	2534.00	0.2962	8.656
30	125.59	2427.73	2553.32	0.4363	8.446
40	167.34	2404.90	2572.24	0.5719	8.253
50	209.11	2381.64	2590.75	0.7032	8.074
60	250.90	2357.91	2608.81	0.8305	7.909
70	292.75	2333.65	2626.40	0.9543	7.756
80	334.66	2308.82	2643.48	1.0746	7.613
90	376.65	2283.38	2660.03	1.1918	7.480
100	418.75	2257.24	2675.99	1.3064	7.356
110	460.97	2230.35	2691.32	1.4177	7.240
120	503.36	2202.65	2706.01	1.5268	7.130
130	545.93	2174.04	2719.97	1.6335	7.027
140	588.71	2144.44	2733.15	1.7381	6.929
150	631.75	2113.76	2745.51	1.8407	6.837
160	675.06	2081.89	2756.95	1.9416	6.749
170	718.66	2048.72	2767.38	2.0406	6.664
180	762.72	2014.10	2776.82	2.1384	6.584
190	807.15	1977.89	2785.04	2.2348	6.506
200	852.02	1939.93	2791.95	2.3299	6.430
210	897.35	1900.00	2797.35	2.4239	6.357
220	943.24	1857.89	2801.13	2.5169	6.285
230	989.75	1813.33	2803.08	2.6091	6.213
240	1036.97	1766.02	2802.99	2.7007	6.143
250	1084.97	1715.59	2800.56	2.7919	6.072
260	1133.87	1661.60	2795.47	2.8828	6.000
270	1184.32	1603.51	2787.83	2.9746	5.927

of steam data based on calorimetric measurements because so many factors enter, each of which contains an element of uncertainty. The experimenter is probably best acquainted with the facts which should be considered in arriving at such an appraisal. Taken together with the evidence included in a description of the work, and making due allowance for possible bias, an experimenter's own judgment as to accuracy may prove useful to the critical reader.

By careful study of all the factors which enter into the measurements, a figure can be assigned to each which represents an estimate of the magnitude of the outstanding systematic error which remains in each factor after all known corrections for standards and calibrations have been applied. Since the signs of these various outstanding sources of systematic error are unknown, they may be considered as combining fortuitously in the final result.

In addition to these a figure can be deduced, for each of the quantities measured calorimetrically, from the differences actually exhibited by individual results, which denotes an estimate of the magnitude of the error of each final result arising from accidental errors in making the measurements.

Having made these preliminary estimates it is possible, with the aid of the theory of probability, to combine them and finally obtain a figure which represents an estimate of the limit of variability within which it can be expected the true value lies with any certain degree of likelihood.

Such an analysis has been made of the results of the present investigation, by consideration of every source of error both systematic and accidental, which seems to have a significant effect. A final estimate has thus been obtained for each of the three thermal properties, H, L, and H'. In each case limits have been estimated such that there is only one chance in a hundred that the true value lies outside these limits. This is a purely arbitrary way of expressing an index of the accuracy of the results. Such an estimate admits the one chance in a hundred of the error being greater than the index, and also admits a chance that some important source of systematic error has been overlooked or underestimated.

On this basis it is estimated that there is only one chance in a hundred that the values given for H differ from the truth by as much as 1 part in 2000. It is estimated to be equally unlikely that the values given for L and H' are as much as 1.5 joules per gram from the truth in the range of the experiments from 100 to 270 deg. cent.

ACKNOWLEDGMENTS

It is impossible to acknowledge adequately all the aid which has contributed to the success of this endeavor. The initiative can be attributed to Mr. Geo. A. Orrok. His vision and forethought focused attention on the pace at which steam-power development was outstripping physical knowledge of properties of steam, and his active interest enlisted the generous support of the steam industries through The American Society of Mechanical Engineers for the research project of which the present work is a part.

Manufacturers have been glad to undertake special tasks to further the work. The Hider Steel Ball Works showed an especially generous scientific interest in accepting, at a nominal cost, the problem of grinding tiny iridium-alloy balls needed in the construction of the apparatus.

Nearly every division of the Bureau of Standards has cooperated in the work, and individuals of the staff have given valuable assistance. To the skill and handicraft of Mr. William Hausstein, instrument maker, is due much of the excellence of mechanical construction of parts of the apparatus.

Without attempting to do justice individually to all who have cheerfully helped this research along, it can be said that whatever success has been attained should be credited to the hearty cooperation of many contributors.

FSP-52-29

Calorimetry of a Fluid[1]

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A rigorous analysis of the thermal processes involved in the change of state of a saturated fluid shows that four types of calorimetric processes each yield values of a characteristic thermal property of the fluid. This group of calorimetric properties may be used to formulate the thermal behavior of the fluid in terms of familiar and convenient properties within the range of the experimental possibilities.

The principles of a calorimetric method of determining the thermodynamic characteristics of a fluid are outlined. The method has been developed at the Bureau of Standards and used in the determinations of the properties of saturated water and steam, published simultaneously with this paper. This paper supplements that report and furnishes the theoretical groundwork for the experimental work.

The results of the analysis have been used as a guide in developing a systematic experimental project which has resulted in the determination of an important group of properties of saturated steam.

I—INTRODUCTION

THIS paper presents the principles of a systematic method of observing the thermal behavior of fluids. The method is intended primarily for the purpose of obtaining data useful in heat engineering.

Certain characteristic thermodynamic properties of a fluid exhibit its behavior with respect to changes of temperature, pressure, volume, and energy. Knowledge of these properties is indispensable to engineers in analyzing problems which arise from the use of fluids in heat engines. It is customary to collect and arrange in the convenient form of numerical tables and graphical charts the values of these properties for fluids used as media in thermodynamic processes.

The most formidable problem in the preparation of a thermodynamic table or chart for a given substance is to obtain data adequate as to kind, range, and accuracy. With very few exceptions existing thermodynamic tables have been prepared from scanty and incongruous data collected from various sources, and on that account require great dependence on the compiler's appraisal.

The discrepancies between steam tables bear witness to the difficulties of measuring thermodynamic properties and emphasize the need of further development of methods of measurement.

II—CALORIMETRY AS A METHOD OF SURVEYING THERMAL BEHAVIOR OF A FLUID

Measurement of the amount of heat which enters or leaves a mass of fluid when its state changes affords a means of studying thermal behavior appropriate for formulating those thermodynamic properties of a fluid which are important in engineering calculations.

Although calorimetry has been used for obtaining certain indispensable data, the full possibilities and advantages of this mode of experimental study as a systematic basis for a complete thermodynamic formulation have not been utilized. Formerly when measurements of heat were made only in terms of the heat capacity of a chosen standard substance, such as water, when the technique of temperature control and measurement was only beginning to be developed, and when methods for controlling thermal leakage in calorimeters were yet crude, it was expedient to avoid where possible the inherent difficulties of calorimetric methods and use simpler and more easily manipulated means, such as the simultaneous observation of pressure, volume, and temperature, for establishing the thermal behavior of a fluid.

Development of trustworthy standards and methods for electrical measurements brought new possibilities to calorimetry both as to increased accuracy and enlarged scope. The adopting and perfecting of electric heaters, resistance thermometers, and thermocouples, have made it possible to develop calorimetry into a reliable agency for thermal research. Ways to confine heat within a definite region, in opposition to the inherent tendency of heat to dissipate, have been developed, and it is now possible to refine the various elements of a calorimetric apparatus to such a degree that the measurements will be consistently accurate.

Conservation of energy is the keynote of calorimetry. One conception of a calorimeter is a place where we confine measured energy while observing the change in state which it produces in a measured sample of fluid.

If the sample is part liquid and part vapor the definite relation between saturation pressure and temperature makes it possible to keep a definite control on the state of the sample and to make it pass through definite processes which effectively exhibit its thermal characteristics.

The proposed method makes use of a single calorimetric equipment by means of which a system of measurements may be made which will determine some of the most essential thermodynamic properties of a fluid. In this method a sample of the fluid in a closed shell at some chosen temperature is heated electrically from the initial temperature and pressure to some other chosen state, or is withdrawn at any chosen temperature and pressure either as liquid or vapor. The energy input required for each of these processes is measured electrically. The amount of heat added per unit mass in any of these processes is characteristic of the fluid and of the particular experimental process observed.

As the aggregate result of the group of measurements it turns out that the property of a fluid which is of utmost importance as a measure of available energy in a thermal process—that is, the "heat content" or "total heat"—is the one most directly determined by the several calorimetric processes, but that specific heat, entropy, and specific volume also may be determined from this group of measurements. The method has been developed at the Bureau of Standards for determining the properties of steam, and has previously been described in part.[3] The analysis previously given, which was restricted to a saturated fluid, has been revised so as to make it more general, and now includes application of the method to determinations of properties of the superheated vapor.

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[1] Publication approved by the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Physicist, Bureau of Standards.

Presented at the Second International Steam Table Conference, Berlin, Germany, June 23-26, 1930.

[3] *Jl. Opt. Soc. Am. & Rev. Sci. Inst.*, vol. 8, no. 4, April, 1924, p. 519.

In order better to visualize the physical aspect of the method, reference will be made to the actual apparatus which has been developed in collaboration with H. F. Stimson and E. F. Fiock. This apparatus has been completed and has been used for a series of measurements extending over several years. Brief illustrated descriptions of the general form of the apparatus, of some of the important details of construction, and of tentative experimental results obtained have appeared from time to time as progress reports.[4]

A comprehensive account[5] of the construction, method of use, and results already obtained is now in preparation for publication. This account will contain the evidence as to the soundness of the method and the reliability of the results obtained. Only a very concise description of those vital features of the apparatus and procedure which pertain to the analysis of the method will be given here.

III—DESCRIPTION OF EQUIPMENT AND EXPERIMENTS

1—Equipment

The apparatus consists essentially of a calorimeter where a sample of water may be sufficiently isolated from other bodies to enable its amount, state, and energy to be accounted for. The sample may be made to pass through a chosen, accurately determined change of state while the accompanying gain or loss of heat is likewise accurately determined. The design of the apparatus provides for several such experimental processes selected for their physical simplicity and for their fitness and sufficiency to exhibit the thermal behavior of the fluid.

A quantity of water, part liquid and part vapor, is inclosed in a metal shell. The water is circulated rapidly about the interior in such a manner as to distribute heat and promote close approximation to thermal equilibrium. An electric heater continually bathed with flowing water provides a means of adding measured heat, which is speedily distributed throughout the calorimeter system. Outlets with valves provide for the introduction or withdrawal of either liquid or vapor. Detachable receivers suitable for weighing are connected to the outlets to hold the samples of water transferred.

For confining the heat the calorimeter is well insulated from the influence of external sources of heat and cold. In operation, the temperature of an enveloping shell is kept very close to that of the calorimeter shell itself. The heat which passes by leakage to or from the calorimeter system is accounted for as a small correction which is determined. The power consumed in circulating the fluid and added to the system as heat is another small correction which is determined.

Means are provided for observing the following quantities:

a Temperature of the calorimeter and contents.

b Vapor pressure in the calorimeter.

c Mass of fluid contents of the calorimeter.

d Heat added to the system as electric power converted to heat and also the small corrections for thermal leakage and circulation.

The apparatus is designed to permit four special types of experiments to be made. In the ideal case of perfect manipulation and control of experimental conditions these would consist essentially of the following processes:

1 Heating with fixed amount of contents.

2 Isothermal expansion by adding heat, evaporating liquid, and removing *saturated vapor.*

3 Isothermal expansion by adding heat, evaporating liquid, throttling, reheating, and removing *superheated vapor* at the saturation temperature and reduced pressure.

4 Isothermal expansion by adding heat, evaporating liquid, and removing *saturated liquid.*

2—Heat-Capacity Determination

The experiment of the first type is a heat-capacity determination. In this experiment the temperature is raised from an initial to a final equilibrium value by adding heat, keeping the amount of fluid in the calorimeter constant. The pressure, the densities, and the relative proportions of vapor and liquid will change in the manner characteristic of the fluid. The volume of the container will change slightly with temperature and pressure. A single experiment of this type will determine the thermal capacity of the calorimeter and contents over the interval through which the state of the system is changed. This interval may be expressed in terms of either the temperature or the pressure, according to convenience. Two experiments of this kind, each extending over the same interval, but with different amounts of fluid in the calorimeter, will differ in observed thermal capacity by the thermal capacity of the difference in contents. Thus, the thermal capacity of the instrument may be eliminated and a value found for the thermal capacity of unit quantity of a definite combination of saturated liquid and vapor.

This result expresses a definite thermal property of the fluid, for its value is dependent only on the nature of the fluid and on the initial and final states. It bears a definite and simple relation to other more familiar thermal properties, particularly the change in heat content of the saturated liquid which it approximates in value and which is determined by this type of experiment supplemented only by experiments of type 4. An experiment of the first type is thus substantially a determination of the change in heat content of the saturated liquid. The resultant heat quantity differs from the change in heat content by a correction term which is determined by the fourth type of experiment, and is small in amount at low temperatures, increasing with temperature and pressure.

3—Latent-Heat Determination

An experiment of the second type is virtually a latent-heat determination. In this experiment a definite portion of the contents of the calorimeter is withdrawn as saturated vapor at the calorimeter temperature, which is maintained approximately constant during the experiments by adding heat to compensate for the energy absorbed in evaporating the fluid. Such changes in the experimental conditions as do occur will be accidental deviations from the ideal constancy aimed at. These deviations can never be absolutely annulled, but their amount may be controlled by proper refinement of equipment and operation so that the effect is either negligible or else may be evaluated as a small correction.

The result of a single experiment of this type is a determination of the amount of heat which must be supplied when unit quantity of saturated vapor is removed from the calorimeter. This includes not only the quantity of heat required to evaporate the unit mass removed, but also the heat to produce an additional amount of vapor sufficient to fill the space which is vacated by the fluid withdrawn. The observed heat quantity differs from the latent heat of vaporization by this extra amount, and this correction turns out to be the same as that required to evaluate the heat content of the liquid from the result of the first type of experiment.

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[4] *Mech. Eng.*, vol. 45, no. 3, p. 168, 1923; vol. 46, no. 2, pp. 81, 83, 1924; vol. 46, no. 11a, p. 808, 1924; vol. 47, no. 2, p. 106; 1925; vol. 48, no. 2, p. 152, 1926; vol. 49, no. 2, p. 162, 1927; vol. 50, no. 2, p. 152, 1928; vol. 51, no. 2, p. 125, 1929; vol. 52, no. 2, p. 127, 1930.

[5] "A Calorimetric Determination of Thermal Properties of Saturated Water and Steam from 0 Deg. Cent. to 270 Deg. Cent." By N. S. Osborne, H. F. Stimson, and E. F. Fiock. Trans. A.S.M.E., vol. 52 (1930), paper FSP-52-28.

4—Superheat Determination

The third type of experiment is a measurement of the heat content of superheated vapor referred to that of saturated liquid at the same temperature. This experiment differs from the second by throttling the sample of vapor from the state of saturation to a lower pressure, leaving the temperature at which it is withdrawn the same as before. The second type of experiment is the limiting case where the throttling is zero and the change in heat content becomes equal to the latent heat of vaporization. A still more general type of experiment would be the case where fluid is withdrawn at any temperature and pressure which can actually be produced and observed. The special cases are simpler to deal with in actual experiments, although the more general case is useful in developing the theory.

The throttling device consists of a closely fitting threaded metal plug at the entrance to the vapor outflow tube. Throttling occurs as leakage past this thread, and the degree of throttling or drop in pressure may be adjusted by advancing or withdrawing this plug. After passing the throttle the vapor enters a tube which is coiled within the vapor space of the calorimeter and leads to the outlet where the vapor is finally withdrawn from the calorimeter. The coiled tube between the throttle and the outlet is for reheating the vapor after reduction of pressure. The throttle and reheat tube are both situated in the vapor space, where they are always bathed with saturated vapor. Condensation of saturated vapor on the tube supplies the heat necessary to bring the flowing vapor back to the saturation temperature at which the calorimeter is being operated. The heat thus absorbed is a part of the total supplied to the calorimeter.

The whole process is equivalent to isothermal throttling of initially saturated vapor, effected in a place where any cooling which occurs in the process is automatically compensated without disturbing that uniform distribution of temperature which permits control and annulment of thermal leakage.

In this experiment the amount of heat is measured which must be supplied when unit quantity of vapor is produced by evaporation, reduced in pressure, and removed from the calorimeter at the saturation temperature which prevails within, where the evaporation takes place. This quantity of heat includes the latent heat of vaporization of unit mass, the correction for an extra amount evaporated to fill the vacated space, and the heat absorbed by the vapor in the process of isothermal throttling from saturation to the reduced pressure.

5—Determination of Correction

The fourth type of experiment determines the correction which is needed to reduce the results of the other three experiments to simpler terms. This correction is observed directly as the heat added to keep the system in equilibrium when a sample of saturated liquid is withdrawn, and is simply the heat required to vaporize enough liquid to fill the space vacated.

Further discussion of the physical interpretation of the results of the several types of calorimetric experiments will be reserved until after the quantitative relations have been analyzed. It will then become evident that the group of experimental processes each observed quantitatively over a range of experimental conditions, determines a group of thermal properties which, together with general thermodynamic relations and the two additional observed quantities, temperature and pressure, establish the thermal behavior of the fluid in a manner thoroughly consistent and suitable for practical formulation.

In actual experiments, slight departures from the ideal conditions of constant temperature, constant outflow, and constant energy input will be bound to occur even with the most refined technique attainable. In order to take into account these fortuitous variations from the ideal processes, the analysis will be developed first for the general case and then by proper specializations applied to the four particular experimental processes which are expected to be useful in practice.

IV—Analysis of Calorimetric Processes

1—Notation

M = mass of fluid contents of calorimeter
θ = temperature on the thermodynamic scale
π = saturation vapor pressure
V = internal volume or capacity of calorimeter
u = specific volume of saturated liquid
u' = specific volume of saturated vapor
x = fraction of fluid contents of calorimeter which is vapor. Quality factor
Q = net heat added to calorimeter and contents in any experiment including electric power input, thermal leakage, and heat dissipated in work of circulating fluid
W = external work done by the system in any experiment
$-dM$ = any infinitesimal element of mass of fluid transferred from calorimeter
y = fraction which is vapor of any infinitesimal element ($-dM$) transferred
p = pressure at which any infinitesimal element ($-dM$) is transferred
v = specific volume of fluid transferred
E_A = internal energy of calorimeter
E_x = internal energy of contents
ϵ = internal energy of unit mass of saturated liquid
ϵ' = internal energy of unit mass of saturated vapor
e = internal energy per unit mass of fluid transferred
H = heat content per unit mass of saturated liquid referred to the value at some arbitrary temperature as zero. $H = \epsilon + \pi u$
H' = heat content per unit mass of saturated vapor referred to same zero as H. $H' = \epsilon' + \pi u'$
L = heat of vaporization of unit mass of fluid. $L = H' - H$
h = heat content per unit mass of fluid transferred, referred to same zero as H. $h = e + pv$
q = heat added to unit mass of saturated liquid in a reversible process
q' = heat added to unit mass of saturated vapor in a reversible process
Φ = entropy per unit mass of saturated liquid
Φ' = entropy per unit mass of saturated vapor
ϕ = entropy per unit mass of superheated vapor
σ = specific heat of saturated liquid $\frac{dq}{d\theta}$
σ' = specific heat of saturated vapor $\frac{dq'}{d\theta}$

Subscripts 1 and 2 denote initial and final equilibrium values, respectively, of the quantities to which they are attached.

2—Derivation of General Equation

The purpose of the analysis which follows is to formulate the relations between the quantities observed in the experiments and the properties of the fluid which are to be determined, with the ultimate object in mind of making a systematic formulation of the thermal behavior of the fluid in a convenient form. We shall briefly recall, of the things which happen in the experiment of the more comprehensive or general type, those changes or operations which directly concern the analysis.

A measured quantity of heat is added and distributed to the calorimeter and its fluid contents, part liquid and part vapor, and at the same time fluid is gradually withdrawn. These two actions are under the control of the operator. As a result a change occurs in the state of the calorimeter and the fluid within it. Accompanying this change of state, work is done on or by the system against external forces. The change in state is determined by suitable observation of the temperature and pressure, and these data, together with the observed data on the transfer of the fluid, gives a basis of accounting in terms of external work and internal energy for all the measured energy added in the form of heat. This accounting follows the principle of conservation of energy and is the first step in the analysis. The usual direction of the operations in an actual experiment is determined by convenience of electrically measuring heat into the system and of measuring outflow of fluid, and by the direction of pressure change in throttling. Except for these practical limitations these operations are not limited in sense, and the analysis is not restricted to either direction.

We shall define the system to which the principle is to be applied as that space wherein by observation we may account for heat added, amount and state of matter, and work done. This definition is essentially a description of the calorimeter and its contents, a system which, it is true, changes in mass, but for which, nevertheless, the energy account may be kept. In the present analysis we shall disregard as insignificant all changes of energy within the system other than measured heat added, external work done or change of internal energy, thus excluding changes in potential or kinetic energy of the aggregate mass. It then follows from the principle of conservation that whatever net energy is added to the system as heat must appear either as net increase of internal energy of the system in its final state of equilibrium over that in its initial state, as net internal energy of fluid withdrawn, or as net work done against external forces, all these quantities being expressed in the same energy units. We may write this in our chosen notation

$$Q = [E_A]_1^2 + [E_x]_1^2 - \int_1^2 e dM + W \quad \ldots\ldots [1]$$

Since the various elements of internal energy appear in the present analysis only in intermediate steps where numerical evaluation is not required, it is not necessary yet to introduce a reference value, but it is sufficient to specify that all these elements of internal energy are referred to the same state as a datum.

This rudimentary equation will next be transformed into a form adapted to our purpose by introducing specifications as to the mass, volume, quality, and specific properties of the fluid.

In the first place, the total volume V of contained fluid may be expressed in terms of its total mass M, the quality factor or vapor fraction x, and the specific volumes of the saturated vapor and liquid, u' and u, in the equation

$$V = Mxu' + M(1 - x)u \quad \ldots\ldots [2]$$

Likewise the total internal energy, E_x, of the contained fluid may be expressed in terms of M, x, and the internal energies of the saturated vapor and liquid, ϵ' and ϵ, in the equation

$$E_x = Mx\epsilon' + M(1 - x)\epsilon \quad \ldots\ldots [3]$$

Eliminating Mx from [3] by use of [2],

$$E_x = M\epsilon + (V - Mu)\frac{\epsilon' - \epsilon}{u' - u} \quad \ldots\ldots [4]$$

and since by definition $H' = \epsilon' + \pi u'$ and $H = \epsilon + \pi u$, and also since $H' - H = L$, we have

$$L = \epsilon' - \epsilon + \pi(u' - u) \quad \ldots\ldots [5]$$

whence we may eliminate ϵ and ϵ' in Equation [4] and obtain

$$E_x = V\left(\frac{L}{u' - u} - \pi\right) + M\left(H - \frac{u}{u' - u}L\right) \quad \ldots\ldots [6]$$

Before substituting this value for E_x into Equation [1] we will find a similar equivalent for W, the external work done by the system. This consists of two possible parts, one the work done against the external pressure by reason of an increase in volume of the calorimeter shell, and the other the work done by the fluid as it issues from the calorimeter. The first part is the product of the constant and uniform external pressure by the increase of the external volume, and in all practical cases would be insignificant, particularly if the envelope space were evacuated for thermal insulation. The second part is the total work done by all the elements $(-dm)$ removed, each of which in emerging does a bit of work equal to the product of the instantaneous pressure p by the volume $v(-dm)$ transferred. Neglecting the insignificant part, the whole work done by the system during an experiment is therefore

$$W = -\int_1^2 pvdm \quad \ldots\ldots [7]$$

Returning to Equation [1] and substituting for E_x from [6], for W from [7], and h for $e + pv$, we have as the general equation applying to all the processes to be considered—

$$[Q]_1^2 = \left[E_A - V\pi + \frac{V}{u' - u}L\right]_1^2 + \left[M\left(H - \frac{u}{u' - u}L\right)\right]_1^2 - \int_1^2 hdM \quad \ldots\ldots [8]$$

3—Remarks on the Physical Interpretation of the General Equation

Equation [8] is an expression for the heat added in a complete experimental process of the general type which has been considered. It is now in convenient form for physical interpretation and for application to the special cases of the four types of experiment of the simpler kind, for the second member shows how the whole amount of heat added is disposed of, taking into account the quantity, distribution, state, and properties of the fluid, and the characteristics of the container. This equation properly applies to any process where heat is exchanged with a quantity of fluid consisting of part liquid and part vapor, as, for example, the operation of a boiler or a condenser, provided, of course, that we are able to keep the account of energy and material correctly. The main difference between our ideal calorimeter and a boiler consists in the refinements provided for accurate bookkeeping. The scale upon which a calorimeter is constructed, of course, need be only large enough for convenient construction and operation and accurate observation.

If we examine the individual terms of the second member of Equation [8] we shall see that each of the three terms denotes a particular portion of the heat added, or its equivalent, very definitely identified with a particular part of the entire process.

The first term involves the internal volume and internal energy of the container itself but is independent of the amount of fluid contents, excepting that enough fluid must always be present to maintain the saturation condition. This term includes those characteristics of the calorimeter which affect the heat-absorbing capacity of the system. Neither of the other two terms refers to the calorimeter itself.

The second term expresses the heat added by virtue of the combined change in amount and state of the portion of fluid within the calorimeter between the beginning and the end of the experiment. This term is entirely independent of the inter-

mediate process by which this change takes place, being completely determined by the initial and final values of mass and temperature.

The third term completes the account by expressing the heat taken away from the system by the portion of fluid withdrawn. This term depends for its value not only on the initial and final states of the system, but upon the state of the issuing stream at every instant during the course of the experiment.

Inasmuch as the physical operations upon which Equation [8] is based are indifferent as to direction, the equation is not limited in its application to changes in a particular sense. If, instead of an electric heating current, we should employ a current of cooling fluid in a coiled tube so that it could absorb and take out heat, the equation would still apply for a decrease instead of a rise in temperature, although the measurement of the heat would then have to be provided for in a different way. Also, except for the case of the throttling process, the transfer of fluid may occur either to or from the calorimeter without affecting the validity of the equation. Indeed it is possible even to apply it to a case where both directions of flow occur simultaneously, as, for instance, if we should introduce liquid and remove vapor at the same time, thus simulating the action of an ordinary steam boiler, or by reversing this process make the system operate as a condenser.

As far as theory is concerned, Equation [8] is general in its application to the types of thermal processes which may take place in a saturated fluid, considered as an isolated system. We shall consider in detail the four specific types of experiments which are adapted to the determination of the thermal properties of the fluid.

4—Constant-Mass Experiments

In the first type of experiment the amount of fluid in the calorimeter is to be kept constant during a single experiment. For this condition, the last term in Equation [8] vanishes. We shall use the symbol Z to denote the quantity $E_A - V\pi + \frac{V}{u' - u} L$. Let M_a and M_b denote the masses of the contents in two separate experiments in each of which the temperature of the calorimeter and contents is changed from θ_1 to θ_2. Let Q_a and Q_b denote the measured quantities of heat added in the two experiments, thus obtaining the following two equations:

$$Q_a = [Z]_1^2 + M_a \left[H - \frac{u}{u' - u} L \right]_1^2 \quad \ldots\ldots [9]$$

$$Q_b = [Z]_1^2 + M_b \left[H - \frac{u}{u' - u} L \right]_1^2 \quad \ldots\ldots [10]$$

Solving these two simultaneous equations for $\left[H - \frac{u}{u' - u} L \right]_1^2$ and $[Z]_1^2$, we have

$$\left[H - \frac{u}{u' - u} L \right]_1^2 = \frac{Q_b - Q_a}{M_b - M_a} \quad \ldots\ldots [11]$$

$$[Z]_1^2 = \frac{Q_b M_a - Q_a M_b}{M_a - M_b} \quad \ldots\ldots [12]$$

5—Evaporation Experiments

In an experiment of type 2, saturated vapor is withdrawn from the calorimeter and consequently the h in Equation [8] becomes H'. In order to take into account the effect of variations in the state of the vapor, H' may be separated into two parts, one a constant value, H_1', and the other the deviation, $H' - H_1'$, from this constant value. If we substitute for h these two values into Equation [8], the last term, which expresses the energy transferred in the outflowing stream, may be divided, one portion corresponding to the value for outflow at the steady state corresponding to H_1', and the other the effect of the variation of the state of the vapor from this ideal steady state. Making this substitution, also substituting Z for $E_A - V\pi + \frac{V}{u' - u} L$ and expanding the expression $\left[M \left(H - \frac{u}{u' - u} L \right) \right]_1^2$ into the form

$$(M_2 - M_1) \left(H - \frac{u}{u' - u} L \right)_1 + M_2 \left[H - \frac{u}{u' - u} L \right]_1^2$$

we have

$$Q = [Z]_1^2 + M_2 \left[H - \frac{u}{u' - u} L \right]_1^2 - (M_2 - M_1) \left(H - \frac{u}{u' - u} L \right)_1 - (M_2 - M_1) H_1' - \int_1^2 (H' - H_1')\, dM \quad \ldots\ldots [13]$$

Combining terms,

$$Q = [Z]_1^2 + M_2 \left[H - \frac{u}{u' - u} L \right]_1^2 - (M_2 - M_1) \left(L + \frac{u}{u' - u} L \right)_1 - \int_1^2 (H' - H_1')\, dM \quad \ldots\ldots [14]$$

and solving for $\left(L + \frac{u}{u' - u} L \right)_1$,

$$\left(L + \frac{u}{u' - u} L \right)_1 = \frac{1}{M_2 - M_1} \left\{ -Q + [Z]_1^2 + M_2 \left[H - \frac{u}{u' - u} L \right]_1^2 - \int_1^2 (H' - H_1')\, dM \right\} \quad \ldots [15]$$

In Equation [15] the quantities $[Z]_1^2$ and $\left[H - \frac{u}{u' - u} L \right]_1^2$ would be determined from the results of experiments of type 1. As a matter of fact, an experimenter would aim to keep conditions so steady that these two quantities would be negligible.

The quantity $\int_1^2 (H' - H_1')\, dM$ if large enough to be significant would be determined by periodic observations of the state of the outflowing vapor during the experiment. By refinement of equipment and manipulation this correction as well as the two just previously mentioned would all be kept very small. The requirement to make them all zero would be to maintain constant temperature and pressure, and for this ideal experimental condition we should have the simpler form of equation

$$\left(L + \frac{u}{u' - u} L \right)_1 = - \frac{1}{M_2 - M_1} Q \quad \ldots\ldots [16]$$

in which M_2, M_1, and Q are determined by observation, thus allowing the quantity $L + \frac{u}{u' - u} L$ to be determined at a certain chosen temperature in each experiment. This quantity is the property of the fluid, characteristic of this experiment, which was previously mentioned as being approximately the

heat of vaporization at low temperatures and pressures where the ratio $\frac{u}{u'-u}$ is small.

6—Throttling Experiments

In an experiment of type 3 superheated vapor is withdrawn at the temperature θ, and the reduced pressure, p, and for this case the h in Equation [8] may be denoted as $(h)_{\theta p}$. In a manner similar to that followed in the case of evaporation experiments $(h)_{\theta p}$ may be separated into a constant and a variable part; that is,

$$(h)_{\theta_1 p_1}, \text{ and } (h)_{\theta p} - (h)_{\theta_1 p_1}$$

but $(h)_{\theta_1 p_1}$ is equal to $H_1' + [(h)_{\theta 1}]_{\pi_1}^{p_1}$. Substituting $H_1' + [(h)_{\theta 1}]_{\pi_1}^{p_1} + (h)_{\theta p} - (h)_{\theta_1 p_1}$ for h in Equation [8], substituting Z for $E_A - V\pi + \frac{V}{u'-u} L$, and $(M_2 - M_1)\left(H - \frac{u}{u'-u} L\right)_1 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2$ for $\left[M\left(H - \frac{u}{u'-u} L\right)\right]_1^2$ as in the previous derivation of the special equation for evaporation experiments, we have

$$Q = [Z]_1^2 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2 + (M_2 - M_1)\left(H - \frac{u}{u'-u} L\right)_1 - (M_2 - M_1)H_1' - (M_2 - M_1)[(h)_{\theta 1}]_{\pi_1}^{p_1} - \int^2 \{(h)_{\theta p} - (h)_{\theta_1 p_1}\} dM \quad \ldots\ldots [17]$$

Combining terms,

$$Q = [Z]_1^2 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2 - (M_2 - M_1)\left[(h)_{\theta p} - H + \frac{u}{u'-u} L\right]_1 - \int_1^2 \{(h)_{\theta p} - (h)_{\theta_1 p_1}\} dM \quad \ldots\ldots [18]$$

and solving for $\left[(h)_{\theta p} - H + \frac{u}{u'-u} L\right]_1$,

$$\left[(h)_{\theta p} - H + \frac{u}{u'-u} L\right]_1 = \frac{1}{M_2 - M_1}\left\{-Q + [Z]_1^2 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2 - \int^2 [(h)_{\theta p} - (h)_{\theta_1 p_1}] dM\right\} \ldots [19]$$

The previous remarks about determining the values of the correction terms which account for the variability of experimental conditions apply to Equation [19] as well as to Equation [15], the only difference being that the last term in [19] accounts for whatever variation may occur in the state of the superheated vapor during the course of the experiment, and both temperature and pressure need to be specified, while in Equation [15] the relation between temperature and pressure is definitely fixed for the saturated vapor. The simpler form of equation for the ideal experimental condition of constant temperature and pressure in the latter case would be

$$\left[(h)_{\theta p} - H + \frac{u}{u'-u} L\right]_1 = -\frac{1}{M_2 - M_1} Q \quad \ldots\ldots [20]$$

in which M_2, M_1, and Q are determined by observation, allowing the quantity $(h)_{\theta p} - H + \frac{u}{u'-u} L$ to be determined at a certain chosen state. This quantity differs from the characteristic property determined in the preceding experiment merely by the addition of the increase of heat content, $(h)_{\theta p} - H_1'$, which corresponds to the change in state produced by the throttling from the saturation state $\theta_1 \pi_1$ to the state $\theta_1 p_1$ at which the vapor leaves the calorimeter. The latent heat of vaporization, which is the change of heat content, $H_1' - H_1$, for the constant-pressure change of state from saturated liquid to saturated vapor, is included in the result, and also the same factor or correction, $\frac{u}{u'-u} L$, for fluid evaporated which remains in the calorimeter. Except for this correction term the quantity measured with this calorimeter in a flow experiment is the increase in heat content for the entire change in state from the saturated liquid within the calorimeter to the state at which the fluid is withdrawn.

7—Experiments in Which Saturated Liquid Is Withdrawn

In an experiment of type 4, saturated liquid is removed from the calorimeter so that h in Equation [8] becomes H.

In a similar manner as before we substitute $H_1 + H - H_1$ for H,

$$Z \text{ for } E_A - V\pi + \frac{V}{u'-u} L$$

and

$$(M_2 - M_1)\left(H - \frac{u}{u'-u} L\right)_1 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2$$

for

$$\left[M\left(H - \frac{u}{u'-u} L\right)\right]_1^2$$

and have

$$Q = [Z]_1^2 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2 + (M_2 - M_1)\left(H - \frac{u}{u'-u} L\right)_1 - (M_2 - M_1) H_1 - \int_1^2 (H - H_1) dM \quad \ldots\ldots [21]$$

Combining terms,

$$Q = [Z]_1^2 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2 - (M_2 - M_1)\left(\frac{u}{u'-u} L\right)_1 - \int_1^2 (H - H_1)\, dM \quad \ldots\ldots [22]$$

and solving for $\left(\frac{u}{u'-u} L\right)_1$,

$$\left(\frac{u}{u'-u} L\right)_1 = \frac{1}{M_2 - M_1}\left\{-Q + [Z]_1^2 + M_2\left[H - \frac{u}{u'-u} L\right]_1^2 - \int_1^2 (H - H_1) dM\right\} \quad \ldots\ldots [23]$$

Again, as in the two previous cases, we have the three correction terms for variability of experimental conditions, two of which are determined from the results of separate experiments of type 1, and the last by periodic observations of the state of the issuing fluid.

The form of equation for ideally perfect operation of this type of experiment would be

$$\left(\frac{u}{u'-u} L\right)_1 = -\frac{1}{M_2 - M_1} Q \quad \ldots\ldots [24]$$

8—Résumé of Theory of Experiments

As the result of a complete group of the four distinct types of experiments, each carried out as a series extending over the range of temperature and pressure within which the properties of the fluid are desired, we have four calorimetric quantities, each of which is characteristic of the process and of the particular fluid used, and each of which is completely determined with respect to the state of the fluid. We shall find it convenient to designate each of these four properties of the fluid by a symbol for use in formulating the intermediate steps by which other thermal properties are derived from these directly measured quantities. In addition to the four quantities which are calorimetric—that is, measured heat quantities—we may include the observed vapor pressure π, thus making five experimental quantities, four of which are functions of temperature, while the other (λ) is a function of temperature and pressure, as the basis from which to formulate the thermal behavior of the fluid. The definitions of these quantities are

$$\alpha = H - \frac{u}{u' - u} L \quad [25]$$

$$\gamma = L + \frac{u}{u' - u} L \quad [26]$$

$$\lambda = (h)_{\theta p} - H + \frac{u}{u' - u} L \quad [27]$$

$$\beta = \frac{u}{u' - u} L \quad [28]$$

It may be noted that $u/(u' - u)$ is a pure ratio and that L denotes the magnitude of a physical quantity which may be determined by experiment independently of any arbitrary choice of a zero datum. Hence the quantities γ and β may be evaluated at any chosen temperature without regard to any arbitrary datum. The quantity α on the contrary contains the quantity H involving the internal energy of unit mass of the fluid which we are able to evaluate only with respect to some arbitrarily chosen reference value. Consequently the quantity denoted by the symbol α can be determined experimentally only as a difference in value between two states of the liquid. The symbol α is used with reference to the same arbitrary zero which is chosen for the quantity H, so hat the same arbitrary constant is tacitly understood for both.

In the series of experiments the quantity α referred to this zero may be determined either by measurement extending from the zero state to the various chosen temperatures throughout the range; or else by measurements of convenient intermediate increments of α, which must cover the entire range.

The quantity λ, like γ and β, is determined experimentally at a definite state, the arbitrary constant being canceled when the difference between $(h)_{\theta p}$ and H is taken.

These five independently observed quantities, when taken together with the thermodynamic relations which hold, in general, for all fluids, are sufficient to establish completely the thermodynamic properties of a fluid over the range of conditions and to the accuracy determined by the experiments.

It will shortly be shown possible to deduce from these five directly measured thermal properties the other more familiar thermal properties which are generally employed in dealing quantitatively with the thermodynamic processes of the fluid.

The completeness with which the whole system of thermal properties may be formulated from a group of data of this type is in practice limited by the experimental possibilities. The method cannot be expected to be ideal in every particular, and the shortcomings must be watched for, not only in the preparations for experiments but in the formulation of results.

The employment of this group of independent experimental data, together with the necessary thermodynamic general relations, results in a formulation which is self-consistent.

The question of reliability of results is extremely important, and in providing data for engineering purposes no opportunity should be overlooked to check the accuracy of results. This may be done, in the first place, by scrupulous attention to details of apparatus, manipulation, observation, and calculation. In the second place, if possible, experimental results from different methods should be subjected to the test of mutual agreement with the general laws of thermodynamics.

The first four of the group of data—that is, the four calorimetric quantities—are characteristic thermal properties of a fluid, which have been employed here solely for the reason that they are useful in the experimental study of the fluid by the calorimetric method. We shall next outline a method by which the thermal behavior as thus determined may be expressed in terms of the more fundamental properties as usually formulated for engineering calculations.

V—Outline of Method of Formulation of Data

The precise manner of evaluating and expressing as functions of temperature or pressure, or both, the various thermal properties which we may desire to formulate is a detail which may be handled in several ways.

There are three ways of exhibiting physical quantities in terms of independent variables, viz., empirical equations, graphs, and tables. Each of these has some virtue, but none is indispensable. It would probably be advantageous to express the properties of steam finally by all three methods as far as possible, but the process by which the data are thus formulated may depend to a considerable extent on the character of the data themselves and on the possibility of obtaining satisfactory empirical equations to represent the experimental facts. The process of formulation should also provide for the comparison of independent experimental data from different sources, and for the appraisal and adjustment of the numerical values to represent the best estimate of the truth.

We shall present here merely an outline of such a process which may be applied to the group of experimental data which we have recapitulated just above, resulting from the observations which may be made in the calorimeter specified.

By simple algebraic combination of Equations [25], [26], [27], and [28], and without utilizing any further thermodynamic relations whatever, we may deduce the following group of equations

$$H = \alpha + \beta \quad [29]$$

$$H' = \alpha + \gamma \quad [30]$$

$$h = \alpha + \lambda \quad [31]$$

These three equations show how the four directly measured heat quantities, α, β, γ, and λ, determine the heat content of the fluid either as saturated liquid, saturated vapor, or as superheated vapor at any pressure below saturation, in each case referred to the value at a chosen reference state.

It should be noted that these equations do not, in general, yield values of H, etc., as functions of temperature or temperature and pressure. They yield a value of H, for example, for every saturation temperature at which α and β have been determined. Only in case α and β are expressed as functions of temperature will these equations lead to an equation for H as a function of temperature. Each of the quantities H, H', h,

derived in this manner from the experimental data should be considered as denoting a series of values of that particular thermal property corresponding to the various chosen states of the fluid at which the observations have been made. If, therefore, the experiments have been suitably distributed as to temperature and pressure, we have thus the skeleton of a steam table, complete as far as heat content is concerned, which may be elaborated by interpolation, by graphical representation, or by formulation into an equation.

Thus heat content, perhaps the most vital of the thermal properties required in engineering calculations, is directly determined calorimetrically.

Owing to the fact that the type of calorimeter chosen for this method operates at the saturation limit, the range of the determination is continuous along the saturation limit and into the superheat region, thus bridging a gap which would otherwise need to be crossed by extrapolation.

The latent heat of vaporization, equal to $H' - H$, is directly determined by the equation

$$L = \gamma - \beta \quad [32]$$

The ratio u'/u of the specific volumes of the saturated vapor and saturated liquid is determined by the equation

$$\frac{u'}{u} = \frac{\gamma}{\beta} \quad [33]$$

which results directly from Equations [26] and [28].

To proceed further with the derivation of thermal properties from the group of experimental data, it is necessary to use relations which depend on the second law of thermodynamics. We shall use the general relation

$$dh = \theta d\phi + vdp \quad [I]$$

which is a statement combining the substance of the first and second laws for a fluid subjected to a reversible change of state. We shall also use Clapeyron's equation

$$\frac{L}{u' - u} = \theta \frac{d\pi}{d\theta} \quad [II]$$

which is deducible from [I] for the special case of evaporation at a constant temperature.

It is, of course, necessary to have a knowledge of the zero of temperature on the thermodynamic scale relative to the working scale used in the measurements. Wherever temperature θ appears as a factor it is to be referred to this absolute zero of temperature. As a consequence of this last relation, [II], it follows that

$$\frac{u}{u' - u} L = \theta u \frac{d\pi}{d\theta} \quad [III]$$

and therefore from Equations [25], [28], and [26] that

$$\alpha = H - \theta u \frac{d\pi}{d\theta} \quad [34]$$

$$\beta = \theta u \frac{d\pi}{d\theta} \quad [35]$$

$$\gamma = \theta u' \frac{d\pi}{d\theta} \quad [36]$$

It will next be shown that the entropies of the saturated liquid and of the saturated vapor are determined by the calorimetric data α, β, and γ directly through these additional relations.

If we solve Equation [I] for $d\phi$ we have

$$d\phi = \frac{dh}{\theta} - \frac{v}{\theta} dp \quad [IV]$$

and since $\frac{dh}{\theta} = d\left(\frac{h}{\theta}\right) + \frac{h}{\theta^2} d\theta$, we have, by substituting in Equation [IV],

$$d\phi = d\left(\frac{h}{\theta}\right) + \frac{h}{\theta^2} d\theta - \frac{v}{\theta} dp \quad [V]$$

and now applying this equation to saturated liquid, noting that $h = H = \alpha + \beta$, and $vdp = ud\pi$,

$$d\Phi = d\left(\frac{H}{\theta}\right) + \frac{\alpha}{\theta^2} d\theta + \frac{\beta}{\theta^2} d\theta - \frac{u}{\theta} d\pi \quad [37]$$

But by Equation [35] the last two terms cancel, so

$$d\Phi = d\left(\frac{H}{\theta}\right) + \frac{\alpha}{\theta^2} d\theta \quad [38]$$

and by integration of this equation,

$$\Phi = \frac{H}{\theta} + \int \frac{\alpha}{\theta^2} d\theta \quad [39]$$

Applying Equation [V] to the saturated vapor, noting that

$$h = H' = \alpha + \gamma \text{ and } vdp = u'd\pi,$$

$$d\Phi' = d\left(\frac{H'}{\theta}\right) + \frac{\alpha}{\theta^2} d\theta + \frac{\gamma}{\theta^2} d\theta - \frac{u'}{\theta} d\pi \quad [40]$$

But by Equation [36] the last two terms cancel, so

$$d\Phi' = d\left(\frac{H'}{\theta}\right) + \frac{\alpha}{\theta^2} d\theta \quad [41]$$

and by integration,

$$\Phi' = \frac{H'}{\theta} + \int \frac{\alpha}{\theta^2} d\theta \quad [42]$$

Equations [39] and [42] are suitable for calculation of the entropies of the saturated fluid from the calorimetric data. The first step—that is, calculation of H and H'—has already been indicated. The values of $\frac{H}{\theta}$ and $\frac{H'}{\theta}$ may be calculated directly. The term $\int \frac{\alpha}{\theta^2} d\theta$ must be evaluated by taking account of how α varies with temperature. An empirical equation for $\frac{\alpha}{\theta^2}$ as a function of temperature would be a convenient means of evaluating this term, and for this purpose need only conform to the data in the range used and be integrable. The constants of integration depend upon the arbitrary zeros chosen for the various quantities in the equation.

It has not been necessary to use vapor pressure π in any of the reductions this far for the properties of the saturated fluid, except incidentally perhaps as a means of taking account of slight variability in the experimental conditions. But if both temperature and pressure are observed simultaneously, either in this or other apparatus, an independent relation is obtained which, with the calorimetric data, determines the specific volumes of the saturated fluid. For this purpose use is made of Equations [35] and [36], which are corollaries of Clapeyron's equation.

Solving these equations for u and u', the specific volumes of saturated liquid and vapor, respectively, we have

$$u = \frac{\beta}{\theta}\frac{d\theta}{d\pi} \quad [43]$$

$$u' = \frac{\gamma}{\theta}\frac{d\theta}{d\pi} \quad [44]$$

These equations are suitable for calculating the specific volumes within the range where $\frac{\beta}{\theta}$, $\frac{\gamma}{\theta}$, and $\frac{d\theta}{d\pi}$ can be determined with the requisite accuracy.

The method of calculating the ordinary specific heat σ of the saturated liquid and σ' of the saturated vapor is merely another detail of mathematical derivation from the calorimetric data. If we define the specific heat of the saturated liquid as the rate of absorption of heat of the homogeneous saturated liquid, in place, relative to temperature rise—that is, $\frac{dq}{d\theta}$, denoted by the symbol σ—we may obtain the equation for σ' from Equation [38], noting that $dq = \theta d\phi$:

$$dq = \theta d\left(\frac{H}{\theta}\right) + \frac{\alpha}{\theta}d\theta \quad [45]$$

whence

$$\sigma = \frac{dq}{d\theta} = \theta\frac{d}{d\theta}\left(\frac{H}{\theta}\right) + \frac{\alpha}{\theta} \quad [46]$$

and similarly for the vapor,

$$\sigma' = \frac{dq'}{d\theta} = \theta\frac{d}{d\theta}\left(\frac{H'}{\theta}\right) + \frac{\alpha}{\theta} \quad [47]$$

For calculating the entropy of superheated vapor a simple method may be employed if the isothermal throttling experiments can in practice be so carried out that determinations of h are made at temperatures and pressures corresponding to intersections of suitable isothermals and constant-pressure lines on the chart.

From Equation [I] at constant pressure,

$$d\phi = \frac{dh}{\theta} \quad [48]$$

whence by integration

$$\phi = \int \frac{1}{\theta}\left(\frac{dh}{d\theta}\right)_p d\theta \quad [49]$$

For the specific volume in the superheat region, we have from Equation [I] at constant temperature,

$$v = \left(\frac{dh}{dp}\right)_\theta - \theta\left(\frac{d\phi}{dp}\right)_\theta \quad [50]$$

or, at constant entropy,

$$v = \left(\frac{dh}{dp}\right)_\phi \quad [51]$$

It is evident that we may continue and calculate numerous other coefficients and derivatives representing certain thermal properties, such as specific heat at constant pressure, Joule-Thomson effect, etc., for comparison with results of experimental data by other methods. It is not surprising, however, to find that there are practical limitations to the range within which some of the thermal properties may be determined by this method of measurement.

For example, it is obvious that high percentage accuracy would be impossible in determining the specific volume of the liquid at low temperature and pressure where both the measured heat quantity β and the rate of change of pressure with temperature $d\pi/d\theta$ are very small, and the expression for specific volume, u, approaches the indeterminate form $\frac{0}{0}$. But fortunately this is a region where other methods of measurement are relatively easy and capable of great accuracy. On the other hand, as temperature and pressure increase, the magnitudes of the quantities which determine u by the calorimetric method get more favorable for accuracy, while other methods requiring actual measurement of volume are more formidable than at lower temperatures. Furthermore, the state of saturation in the calorimeter is a fundamental condition of the method. This constitutes an advantage in the determination of values of thermal properties at the boundary over methods which rely on extrapolation from observations confined to states in the superheat region.

In the case of approach to the critical state, the fact that the properties of the liquid and vapor phases approach equality might be expected to make it more difficult to maintain perfect separation of the phases, thus giving rise to indeterminateness of the physical processes akin to the mathematical forms which thermal properties are supposed to exhibit in this region. It is certain that for steam the critical state would not be a favorable one for calorimetric measurements on account of the high values of temperature and pressure.

Aside from these limitations there is a range of conditions within which it is possible to determine by the proposed calorimetric method the numerical values of the thermal properties of a given fluid to an accuracy appropriate for engineering purposes. The calculation of these numerical values would not necessitate the setting up of an elaborate system of mutually consistent empirical equations. The values would nevertheless be consistent with thermodynamic criteria to the degree of accuracy fixed by the measurements and the calculations. The thermodynamic criteria will have been already included as independent relations used in the scheme of calculating the several properties from the group of independent experimental quantities.

We have, in what has gone before, the skeleton of a formulation of the thermal behavior of the fluid. The formulation may be exhibited in either of the three ways previously pointed out. The numerical values may be tabulated, interpolating by the usual methods between the determined points. They may be plotted graphically in a variety of ways. They may be used to determine the constants in a series of empirical formulas. Or any combination of the three methods may be used. The details of the precise manner of formulating the data for actual use in thermodynamic calculations is beyond the scope of this paper, which merely aims to establish the basic principles of one method of determining the thermal behavior of a fluid.

FSP-52-30

A Review of Calorimetric Measurements on Thermal Properties of Saturated Water and Steam[1]

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This review has been prepared to assemble the data which are now available on the calorimetric determinations of the thermal properties of saturated water and steam, and to reduce them to a uniform basis for comparison. After a careful analysis of the reports which have appeared in the literature, an attempt is made to point out some possible causes for the variation among the recorded results. Wherever it seemed desirable a thermodynamic interpretation of the experiments is given, and in a number of cases the data have been recalculated. By these means the agreement among the various determinations has been improved in several instances.

It is hoped that this assembly of results presented on a uniform basis may serve as an aid to future International Steam Table Conferences in the choice of base values and tolerances for the International Skeleton Tables.

By means of deviation charts the experimental values for the heat content or enthalpy of the saturated liquid and for the latent heat are exhibited and compared. The agreement is in general quite gratifying.

Several modern steam tables are compared graphically with one another and with the Bureau of Standards experimental results on the thermal properties of the saturated liquid and vapor. A marked improvement in the tables is evident with the increase in scope and the reliability of the basic experimental data.

I—INTRODUCTION

THERE have been numerous attempts in the past to formulate the thermal properties of water and steam, by means of equations, into tables convenient for the use of engineers. These formulations have, through necessity, been based on scattered groups of experimental data, recorded by different investigators in a great variety of units. For the most part the variations in the tabulated values in steam tables may be attributed to three causes: First, the choice of various forms of empirical relations to fit data which in themselves are scattered and somewhat discordant; second, the confusion arising from the use of the different measuring standards; third, the thermodynamic interpretation of the experimental data.

An important advance in the evolution of steam tables was made by the International Steam Table Conference at London in July, 1929. One accomplishment of this Conference[3,4,5] was the establishment of an International Skeleton Table, with tolerances, of the thermal properties of water and steam. It was anticipated that the table there adopted was subject to revision by subsequent conferences with the idea of narrowing the limits and approaching more closely the true values.

The publication of the data from the calorimetric determinations at the Bureau of Standards, presenting new values for the thermal properties of steam over a large portion of the saturation region, has suggested the need of a summary of all available data of a similar kind. This review is given as a supplement, appearing simultaneously with the detailed report of the Bureau of Standards determinations[6] to aid in the comparison of these results with those previously published. By the reduction of all the data to a uniform basis for comparison, it is hoped that the review may serve as an aid in the choice of basic values and tolerances for the International Skeleton Steam Table.

It is often possible, by studying the published descriptions, to reveal important evidence as to reliability, and it has been the aim in this review to bring out such evidence. It is gratifying to find that, in some cases, careful analysis and reduction of published data are rewarded by improvement in the agreement of the results.

The experiments in this field fall logically into two groups: (1) Determinations of the heat content or enthalpy of the liquid; and (2) determinations of the heat of vaporization. The various experiments of each group will be discussed in chronological order as far as is practicable.

II—NOTATION

M = mass of fluid
m = mass of saturated vapor
θ = temperature, centigrade
Θ = temperature, centigrade absolute
π = saturation vapor pressure
P = external pressure
V = internal volume of container for water
u = specific volume of saturated liquid
u' = specific volume of saturated vapor
x = fraction of the water which is vapor
E_s = internal energy of entire water sample
ϵ = internal energy per gram of saturated liquid
ϵ' = internal energy per gram of saturated vapor
H = heat content per gram of saturated liquid $(\epsilon + \pi u)$
H' = heat content per gram of saturated vapor $(\epsilon' + \pi u')$
L = heat of vaporization per gram $= H' - H$

$$\alpha = H - L\frac{u}{u' - u} = H - \beta$$

$$\beta = \frac{Lu}{u' - u}$$

$$\gamma = L + L\frac{u}{u' - u} = L\frac{u'}{u' - u}$$

$$L = \Theta\frac{d\pi}{d\theta}(u' - u) \qquad \text{(Clapeyron equation)}$$

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[1] Publication approved by the Director of the Bureau of Standards of the U. S. Department of Commerce.

[2] Associate Physicist, Bureau of Standards.

[3] *Engineering*, vol. 128 (1929), pp. 751–752.

[4] *Z.V.D.I.*, vol. 73 (1929), no. 52, pp. 1856–1858.

[5] MECHANICAL ENGINEERING, vol. 52 (1930), pp. 120–122.

Presented at the Second International Steam Table Conference, Berlin, Germany, June 23–26, 1930.

[6] "A Calorimetric Determination of Thermal Properties of Saturated Water and Steam from 0 Deg. Cent. to 270 Deg. Cent." By N. S. Osborne, H. F. Stimson, and E. F. Fiock. Trans. A.S.M.E., vol. 52 (1930), paper FSP-52-28.

III—ANALYSIS AND REDUCTION OF SOURCE DATA

1-DETERMINATIONS OF HEAT CONTENT OF THE LIQUID

In the temperature range from 0 deg. cent. to 100 deg. cent. the investigations have been numerous and particularly painstaking because the calorie was used as a primary unit of energy and much effort was expended in determining the precise relation between this and other units of energy. This latter knowledge has been necessary in order that all calorimetric data could be reduced to a common basis for comparison. If, as has already been attempted, a heat unit defined arbitrarily and directly in terms of mechanical units is universally adopted and used, the need for these so-called mechanical equivalents of heat vanishes as far as future work is concerned. The difficulties encountered in the preparation of this review emphasize the advantages of adopting some such arbitrarily defined heat unit.

That the experimental determination of the mechanical equivalent of heat referred to water is a difficult task is evidenced in the literature by the diversity of the results recorded. Even after the most careful attempts to reduce the experimental values to a uniform basis, there still exist divergent values for the mechanical equivalents of the various calories.

Since many of the past determinations of heat capacities and of ratios of these give little important information about the heat content, they are accorded only passing mention in this review. More important among them are determinations by Rowland,[7] Griffiths,[8] Schuster and Gannon,[9] Lüdin,[10] Callendar,[11] Romberg,[12] and Laby and Hercus.[13] The results of Barnes[14] and of Jaeger and von Steinwehr[15] yield, on integration, values of heat content which will be discussed more fully later.

(1) Regnault

For a long time the classical work of Regnault[16] furnished the most reliable information available on the specific and latent heats of water. At the time these researches were carried out the physical sciences were in their infancy, but in spite of the many difficulties Regnault's conclusions were remarkably accurate. In the fifty years that followed no similar experiments on water and steam comparable in accuracy are recorded.

No matter how carefully his experiments were done, they could not have yielded results as near to the truth as can now be obtained with the aid of more advanced theory, more definitely established standards, and improved laboratory apparatus. It is not possible, with any considerable degree of certainty, to interpret the temperatures which he recorded in terms of the present international scale. For these reasons, no attempt will here be made to reduce his results to a comparative basis. They are considered instead as an important milestone which it took years to reach, but which has now been left far behind.

(2) Reynolds and Moorby

In the experiments of Reynolds and Moorby[17] the value of the mean calorie was measured directly in terms of mechanical units with a modified Froude brake. This work was done on an engineering scale, and many precautions were taken to insure the precision of the results. The principal corrections were for thermal leakage and liberation of dissolved air. Some errors were probably introduced by formation of vapor, by variations in the speed at which the driving engine ran, and because thermometry at that time was not on an entirely satisfactory basis. On the average the water entered the brake at a temperature of about 1.3 deg. cent. and a pressure of about 2 atmospheres, and left it at a temperature of about 100 deg. cent. and a pressure of about 1.8 atmospheres. A correction was applied for the energy change due to the pressure drop through the system, so that the value recorded is the mean specific heat of water between 1.3 deg. cent. and 100 deg. cent. at a constant pressure of 2 atmospheres. As the mean of 26 separate determinations Reynolds and Moorby report this mechanical equivalent as 4.1832×10^7 ergs per gram degree centigrade along the 2-atmosphere line. If the figure 1.0004 is taken as the ratio between the international and the absolute joule,[18] this value becomes 4.1815 international joules per gram degree centigrade. It may be further corrected to the saturation path by the following method.

The value for the change in heat content with pressure at constant temperature can be evaluated through the relation

$$\left(\frac{\partial H}{\partial P}\right)_\theta = V - \theta\left(\frac{\partial V}{\partial \theta}\right)_P \quad \ldots\ldots [1]$$

in which the right-hand member can be evaluated with sufficient precision from data on specific volumes and coefficients of thermal expansion given in the International Critical Tables. The corrections desired are then the products $\left(\frac{\partial H}{\partial P}\right)_\theta (1 - \pi)$ when the values $\pi = 0.0066$ atmosphere at 1.3 deg. and $\pi = 1$ atmosphere at 100 deg. are substituted. The change in heat content along the saturation path from 1.3 deg. to 100 deg. is greater by $0.206 - 0.076 = 0.13$ international joule per gram than it is along the 2-atmosphere path.

Reynolds and Moorby's value for the change in heat content from 1.3 deg. to 100 deg. along the saturation path becomes $4.1815 \times 98.7 + 0.13 = 412.84$ international joules per gram. The value for the change in heat content of water between 0 deg. and 100 deg. is obtained by adding to this the Bureau of Standards value for the change in H from 0 deg. to 1.3 deg. = 5.49 international joules per gram. The result thus obtained from the work of Reynolds and Moorby is 418.33 international joules per gram for the change in heat content of water between 0 deg. and 100 deg. along the saturation path.

(3) Dieterici

Dieterici's[19] experiments consisted in the introduction of quartz tubes at known temperatures, first empty and then containing known masses of water, into a Bunsen ice calorimeter and observing the contraction in volume due to the fusion of ice. This contraction, which he expressed in milligrams of mercury, is a measure of the heat lost by the tube or the tube and contents when its temperature changed from the initial value θ to 0 deg. cent. By difference the contraction due to the cooling of the water sample alone was obtained. Expressing by $Q]_0^\theta$ this heat lost by the water and by C_m an integrated mean specific heat which is to be experimentally determined, Dieterici's results were evaluated through the equation

$$Q]_0^\theta = m\rho]_0^\theta + MC_m\theta]_0^\theta \quad \ldots\ldots [2]$$

[7] Proc. Am. Acad. of Arts and Sciences, vol. 15 (1879), pp. 75–200.
[8] Phil. Trans. Roy. Soc. London, vol. A184 (1893), pp. 361–504.
[9] Phil. Trans. Roy. Soc. London, vol. A186 (1895), part I, pp. 415–467.
[10] "Die Abhängigkeit der specifischen Wärme des Wassers von der Temperatur." Inaugural Dissertation, Zurich (1895).
[11] Phil. Trans. Roy. Soc. London, vol. A199 (1902), pp. 55–148.
[12] Proc. Am. Acad. of Arts and Sciences, vol. 57 (1921–22), pp. 377–387.
[13] Phil. Trans. Roy. Soc. London, vol. A227 (1927), pp. 63–92.
[14] Phil. Trans. Roy. Soc. London, vol. A199 (1902), pp. 149–263
[15] *Ann. Phys.*, vol. 369 (1921), pp. 305–366.
[16] "Relations des Expériences," vol. 1 (1847), pp. 635–748.
[17] Phil. Trans. Roy. Soc. London, vol. A190 (1897), pp. 301–422.
[18] Bureau of Standards Technical News Bulletin, April, 1930.
[19] *Ann. Phys.*, vol. 321 (1905), pp. 593–620.

in which

$$m\rho]_0^\theta = \left[(V - Mu) \left(\Theta \frac{d\pi}{d\theta} - \pi \right) \right]_0^\theta \quad [3]$$

This correction $m\rho]_0^\theta$ takes into account the energy changes in the ever-present vapor phase. Since the volume change of the retaining tubes due to changes in temperature and pressure is negligible, the heat exchange of the contents becomes identical with the loss of internal energy.

For any state,

$$E_x = Mx\epsilon' + M(1 - x)\epsilon = M\epsilon + Mx(\epsilon' - \epsilon) \quad [4]$$

and

$$V = Mxu' + M(1 - x)u = Mu + Mx(u' - u) \quad [5]$$

Solving,

$$x = \frac{V - Mu}{M(u' - u)} \quad [6]$$

Hence

$$E_x = M\epsilon + (V - Mu) \frac{\epsilon' - \epsilon}{u' - u} \quad [7]$$

But

$$L = \epsilon' - \epsilon + \pi(u' - u) = \Theta \frac{d\pi}{d\theta} (u' - u) \quad [8]$$

Hence

$$\frac{\epsilon' - \epsilon}{u' - u} = \Theta \frac{d\pi}{d\theta} - \pi \quad [9]$$

and

$$E_x = M\epsilon + (V - Mu) \left(\Theta \frac{d\pi}{d\theta} - \pi \right) \quad [10]$$

But

$$Q]_0^\theta = E_x]_0^\theta = MC_m\theta]_0^\theta + \left[(V - Mu) \left(\Theta \frac{d\pi}{d\theta} - \pi \right) \right]_0^\theta \quad [11]$$

From which

$$[C_m\theta]_0^\theta = \epsilon]_0^\theta \quad [12]$$

Dieterici records experimental values of C_m and θ, and, as shown, the product of these is the specific internal-energy change of the saturated liquid. His values of the correction term $m\rho]_0^\theta = \left[(V - Mu) \left(\Theta \frac{d\pi}{d\theta} - \pi \right) \right]_0^\theta$ were calculated from the best values of u, π, and $\frac{d\pi}{d\theta}$ then available. In order that the values of the contractions which he observed in milligrams of mercury could be converted into more convenient units, he made very careful direct determinations of the volume changes produced by the addition of measured quantities of electrical energy.

With the experimental data now available it is possible to interpret his results directly in international joules without reference to his experimentally determined electrical equivalent of the calorimetrically observed quantity. His results at 100 deg. cent. furnish, in effect, a direct determination of the equivalent of the mean calorie in terms of milligrams of mercury. This value in milligrams of mercury corresponds to the value of the mean calorie in international joules, which is now known to a greater precision than he could have obtained with his observations on electrical equivalent. When this method is used to calculate the results of his measurements, all difficulties arising from the evaluation of his electrical standards and from the changes which electrical units have since undergone are avoided.

From recent experiments it is possible to evaluate the correction term $m\rho]_0^\theta$ more exactly than before. At the same time a term of the magnitude πu can be introduced to convert the internal-energy changes into changes of heat content.

As above

$$Q]_0^\theta = M\epsilon]_0^\theta + \left[(V - Mu) \left(\Theta \frac{d\pi}{d\theta} - \pi \right) \right]_0^\theta \quad [13]$$

But

$$\Theta \frac{d\pi}{d\theta} = \frac{L}{u' - u} = \frac{\beta}{u} \quad [14]$$

Hence

$$Q]_0^\theta = M\epsilon]_0^\theta + \left[(V - Mu) \left(\frac{\beta}{u} - \pi \right) \right]_0^\theta \quad [15]$$

Adding $M\pi u]_0^\theta$ to both sides

$$Q]_0^\theta + M\pi u]_0^\theta = M\epsilon]_0^\theta + M\pi u]_0^\theta + \left[(V - Mu) \left(\frac{\beta}{u} - \pi \right) \right]_0^\theta \quad [16]$$

Whence

$$Q]_0^\theta + M\pi u]_0^\theta = MH]_0^\theta + \left[(V - Mu) \left(\frac{\beta}{u} - \pi \right) \right]_0^\theta \quad [17]$$

Expanding and collecting

$$Q]_0^\theta = MH]_0^\theta + \left[V \left(\frac{\beta}{u} - \pi \right) \right]_0^\theta - M\beta]_0^\theta \quad [18]$$

But

$$H]_0^\theta = [\alpha + \beta]_0^\theta \quad [19]$$

Hence

$$Q]_0^\theta = M\alpha]_0^\theta + \left[V \left(\frac{\beta}{u} - \pi \right) \right]_0^\theta \quad [20]$$

Considering now the experiments at 100 deg. cent., if $Q]_0^\theta$ is expressed in milligrams of mercury and both terms in the right-hand member in international joules, we may write

$$KQ]_0^\theta = M\alpha]_0^\theta + \left[V \left(\frac{\beta}{u} - \pi \right) \right]_0^\theta \quad [21]$$

from which the value of the conversion factor K can be determined. Values of α and β are obtainable from the previous paper,[9] and values of π and u, which enter only in the relatively small correction term, from the best sources now available. There are 13 experiments in the neighborhood of 100 deg., each of which yields a value of K.

The 13 values of K thus obtained yield a mean value of 0.27024 international joules per milligram of mercury when the experiments are weighted according to Dieterici's estimate of their reliability. They are very consistent, the maximum deviation from the mean being about 1 in 720 and the average 1 in 1380. The accuracy of the chosen mean value of K is doubtless greater than the experimental accuracy at temperatures other than 100 deg., because of the greater number of experiments performed at that temperature, and because of the comparative ease of maintaining and determining the steam point.

To transform Equation [20] given above into a form convenient for the calculation of heat content, add $M\beta]_0^\theta$ to both sides and collect.

$$M\alpha]_0^\theta + M\beta]_0^\theta = 0.27024\, Q]_0^\theta - \left[V \left(\frac{\beta}{u} - \pi \right) \right]_0^\theta + M\beta]_0^\theta = MH]_0^\theta \quad [22]$$

$$H]_0^\theta = \frac{0.27024\ Q]_0^\theta}{M} - \frac{V}{M}\left[\frac{\beta}{u} - \pi\right]_0^\theta + \beta]_0^\theta \ldots\ldots [23]$$

At any temperature the terms in the right-hand member of this equation can be evaluated from the same sources as at 100 deg. Dieterici's data give the values of $Q]_0^\theta$ corrected for heat leak, the values of M corrected for buoyancy, and the measured values of V. Hence $H]_0^\theta$ can be evaluated at each temperature for which data are given.

Making use of the Bureau of Standards data his values of H have been reduced to even temperatures for convenience in making comparisons with others. Table 1 shows the results of Dieterici's experiments as obtained by this method of calculation.

TABLE 1 HEAT CONTENT OF SATURATED LIQUID WATER, CALCULATED FROM DIETERICI'S EXPERIMENTS

Temperature θ deg. cent.	Heat content H Int. joules per gram	Temperature θ deg. cent.	Heat content H Int. joules per gram
15	63.10	100	418.75[a]
20	83.82	110	461.1
25	104.6	130	546.6
30	125.6	156	660.6
35	146.2	183	776.5
46	192.2	200	852.6
56	234.2	221	946.2
65	271.8	240	1034.7
78	326.4	259	1126.7
88	368.2	303	1348.9

[a] This value has been made to correspond exactly with the Bureau of Standards value by the method of computation.

The accuracy of the temperature measurements, which were made with mercury in glass thermometers calibrated at the Reichsanstalt, was probably so good that the thermometry did not limit the overall accuracy of the experiments. How nearly the water containers assumed the same temperature as that observed for the bath is less certain. Heat-leak corrections were small, and though there is little discussion of how they were obtained, it is believed that they furnished only a minor portion of the error in the final results. At the high temperatures the values of V become more significant, and higher precision in the determination of these would have increased the accuracy of the results. As a whole the research seems to have been carried out with great care and skill, and claims the admiration of the careful reader.

(4) Barnes

Barnes[14] gives a very detailed account of his researches on the heat capacity of water between 5 deg. and 95 deg., using a flow calorimeter of the so-called "continuous electric" type designed in cooperation with Callendar.[11]

Water was admitted into a vacuum-jacketed glass calorimeter at a steady but controllable rate, and its temperature was raised by means of an electric heater located in the stream between a pair of differential resistance thermometers which indicated the temperature rise. The amount of water which had passed through the calorimeter was determined by weighing. The experiments were performed at a constant pressure differing from the atmospheric pressure by the small pressure head used to maintain the flow.

For the electrical equivalent of the mean specific heat between 5 deg. and 95 deg., Barnes gives 4.1888 joules, referred to the international ohm and the volt defined as 1/1.4342 of the e.m.f. of the Clark cell at 15 deg. In a recomputation of his results in 1909[20] he expressed the belief that the voltage of the Clark cells at 15 deg. as used in 1902 was 1.4333. This reduces the results previously published by 1 part in 798. Thus recomputed, the value of 4.1888 joules per gram for the mean specific heat between 5 deg. and 95 deg. becomes 4.1835 joules per gram. Correcting for the end intervals he obtained the value 4.1849 international joules per gram for the equivalent of the mean calorie along the atmospheric-pressure path. In 1910 the value of the international volt was reassigned, using as a standard the Weston normal cell having an e.m.f. at 20 deg. of 1.0183 international volts. In terms of this standard of 1910, Clark cells made up according to the old specification have an e.m.f. at 15 deg. of 1.4328 volts.[21] This may be considered at least a probable value for the Clark cells used by Barnes in terms of the present system. This value further reduces his mechanical equivalent of the mean calorie along the atmospheric-pressure path by 1 part in 1433, to 4.1820 international joules per gram. This value, corrected to saturation pressure, becomes 4.1830 international joules per gram.

In Laby's[22] critical discussion of the mechanical equivalent of heat, the results of Barnes, among others, are reduced to absolute units. Since the values are desired here in international units, and since Laby discusses only the 20-deg. calorie, no comparison with the results given in his review is included.

Without reference to the report of Barnes in 1909, his values for the specific heats can be transformed into present international units as follows. In his original computations the value assigned to the Clark cell at 15 deg. was 1.4342 volts. As stated above, the e.m.f. of Clark cells made up according to the same specifications is 1.4328 international volts. Therefore the values of specific heat as originally reported can be transformed into present international joules by reducing them by 1 part in 512.

Barnes represents his data as finally reduced by three equations, appropriate over the temperature intervals indicated.

$$(5°\text{–}37.5°)\quad C_p = 0.99722 + 0.0000035\ (37.5 - \theta)^2 + 0.00000010(37.5 - \theta)^3 \ldots\ldots [24]$$

$$(37.5°\text{–}55°)\quad C_p = 0.99722 + 0.0000035\ (\theta - 37.5)^2 + 0.00000010(\theta - 37.5)^3 \ldots\ldots [25]$$

$$(50°\text{–}100°)\quad C_p = 0.99839 + 0.000120\ (\theta - 55) + 0.00000025(\theta - 55)^2 \ldots\ldots [26]$$

These are in terms of his own mean calorie, and yield, on integration with respect to temperature, values for the changes in heat content in mean calories. The integrated equations are

$$H]_0^\theta = 0.99722 - 0.00000117(37.5 - \theta)^3 - 0.000000025(37.5 - \theta)^4 \ldots\ldots [27]$$

$$H]_{37.5}^\theta = 0.99722 + 0.00000117(\theta - 37.5)^3 + 0.000000025(\theta - 37.5)^4 \ldots\ldots [28]$$

$$H]_{55}^\theta = 0.99839 + 0.000060(\theta - 55)^2 + 0.000000083(\theta - 55)^3 \ldots\ldots [29]$$

To transform the values calculated from these equations into a form convenient for comparison, two further corrections are applied. Since the first equation is valid only down to 5 deg., the value of $H]_0^5 = 5.02$ mean calories is supplied from the Bureau of Standards data. Further correction is necessary to reduce the results to the saturation pressure for each temperature recorded. The method of calculation of these corrections has already been described. Table 2 gives the values of heat content calculated from Barnes's equations, the corrections to reduce them to the saturation pressures and, in the last column, the values along the saturation path. The figure 418.30 inter-

[20] Proc. Roy. Soc. London, vol. 82 (1909), pp. 390–395.

[21] Bureau of Standards Circular No. 60 (1920), p. 41.

[22] Proc. Phys. Soc. London, vol. 38 (1925–26), pp. 169–175.

national joules per gram as derived above from Barnes's own measurements is used for the conversion of mean calories to international joules.

TABLE 2 HEAT CONTENT FROM BARNES'S SPECIFIC-HEAT EQUATIONS

Temperature θ	Heat content (P = 1 atm.) $H_{1atm.}$	$\left(\frac{\partial H}{\partial P}\right)_\theta (1-\pi)$	Heat content ($P = \pi$) H	Heat content ($P = \pi$) H
deg. cent.	Mean calories per gram			Int. joules per gram
10	10.03	0.02	10.05	42.04
20	20.04	0.02	20.06	83.91
30	30.02	0.02	30.04	125.66
40	39.99	0.02	40.01	167.36
50	49.97	0.02	49.99	209.11
60	59.95	0.02	59.97	250.85
70	69.95	0.01	69.96	292.64
80	79.96	0.01	79.97	334.51
90	89.98	0.00	89.98	376.39
100	100.01	0.00	100.01	418.30

The chief uncertainty in the value of the heat content of water at 100 deg. expressed in international joules per gram from the experiments of Barnes lies in the interpretation of the e.m.f. of the Clark cells used. The value which seems most probable to the reviewer is 418.30 international joules per gram.

In a glass apparatus of the type used by Barnes, the evaluation of the heat leak (4 per cent of the total energy supplied in the extreme case), including the heat effects observed and interpreted as due to adsorption or liberation of residual gas in the evacuated jacket, is a difficult task.

The temperature scale used in these experiments was substantially the same as the present international scale. Observations of temperature differences with the differential resistance thermometers were probably sufficiently precise. The precision with which the electrical measurements were made appears to have been ample. The assumption that the walls of the weighing vessel always retained the same mass of liquid may have been a source of error. The possibility of formation of vapor and liberation of dissolved gas within the calorimeter are inherent in the method and the effects are difficult to evaluate.

To obtain an accuracy of better than 1 part in 1000 in this type of experiment requires a degree of refinement in the calorimetry itself which is not easily attained. It appears that there may have been possible sources of systematic error, some of which have been mentioned and several of which may have exceeded 1 part in 10,000. In the absence of positive supporting evidence it seems that Barnes's estimate of accuracy of 1 part in 10,000 may have been too optimistic.

The reader of Barnes's report is impressed with the remarkable care and skill with which the experimental program was planned and executed, and this piece of work remains one of the outstanding contributions to our knowledge of the heat capacity of water.

(5) Callendar

Callendar spent many years in the investigation and formulation of the properties of steam. In his work on the liquid he is responsible for the development of two types of calorimetry which he has described as the continuous electric method and the continuous mixture method. The former method was used by Barnes in the researches already described, and the results of this investigation are correctly attributed to Callendar and Barnes. At a later date Callendar[23] alone performed experiments by the second method to test the ratios of the specific heats reported by Barnes and found good agreement between the methods. He stated that the method of mixtures was more dependable than the electric method because the heat leak was reduced from about 4 per cent to about $^1/_2$ per cent of the total energy changes involved.

[23] Phil. Trans. Roy. Soc. London, vol. A212 (1912–13), pp. 1–32.

In not only this work of 1912, but also in all the succeeding reports which Callendar has published, the reviewer has little opportunity to appraise the experimental work as such, because practically no experimental data are given. Instead, Callendar has chosen always to formulate his experimental data through equations into tables of the properties most desired. These tables must be taken as representing the best values indicated by his own experiments, but the data of other observers are included in the reductions in such a way that it is impossible to separate them.

Beyond the knowledge that Callendar was an exceptionally brilliant, skilful, and painstaking experimenter who devoted much time to investigations in his chosen field, the lack of published experimental data precludes an estimation of the absolute accuracy of his experimental results, or the agreement between these results and his tabulated values.

Therefore, for the purposes of this comparison, the values taken from his latest steam tables[24] are used as best representative of his entire group of measurements.

Callendar's equation for the heat content of the liquid is

$$H = k(\theta - c) + \frac{Lu}{u' - u} \quad [30]$$

in which k and c are constants. The last term in this equation is the quantity β measured experimentally at the Bureau of Standards, and from the relation

$$H = \alpha + \beta \quad [31]$$

it follows directly that, according to the Callendar formula,

$$\alpha = k(\theta - c) \quad [32]$$

and that

$$\frac{d\alpha}{d\theta} = k \quad [33]$$

The quantity α has been very carefully determined by direct calorimetric measurements at this Bureau at each 10-deg. interval from 0 deg. to 270 deg. These results indicate that the value of $d\alpha/d\theta$ is practically constant in the range 50 deg. to 170 deg., but outside this range it varies considerably. Callendar recognized that his equation was inadequate in the region from 0 deg. to the temperature of the minimum specific heat of water, but accepted it nevertheless, because of its simplicity, its degree of approximation to the truth, and the relative unimportance to the engineers of the values calculated from it in the range near the freezing point. His own experimental evidence confirming the validity of the equations at high temperatures is not at hand. According to the Bureau of Standards data the value of $d\alpha/d\theta$ decreases from a practically constant value of 4.17 international joules per gram degree centigrade in the range 50 deg. to 170 deg. to about 4.10 international joules per gram degree centigrade at 270 deg.

Since the quantity $\beta = Lu/(u' - u)$ is small in comparison with α, the discrepancy in the values of H at the higher temperatures is due largely to the difference in the evaluation of the variation of α with temperature.

Callendar's values of H' doubtless have as a basis his numerous experiments in the superheat region. His values of H are from the equation discussed above, which was based on the work of Callendar, Barnes, and probably Dieterici in the high-temperature range.

In Table 3 are shown, in international joules, the values of H and H' from the most recently published Extended Steam Tables.[24]

[24] Proc. Inst. Mech. Eng., no. 3 (1929), pp. 507–527.

TABLE 3 CALLENDAR'S VALUES OF HEAT CONTENT

Temperature θ deg. cent.	Heat content: Of liquid H	Heat content: Of vapor H'
	International joules per gram	
0	0	2488.3
50	208.77	2584.2
100	418.54	2675.7
150	631.62	2751.8
200	852.14	2801.7
250	1087.36	2818.4
270	1187.81	2811.3

(6) Jaeger and von Steinwehr

The experiments of Jaeger and von Steinwehr on the heat capacity of water from 5 deg. to 50 deg. cent., performed in 1911–12 and published in 1915[25] and more completely in 1921,[15] have been the basis for the value of the 15-deg. calorie used by the Reichsanstalt. Their calorimeter containing 50 liters of water at atmospheric pressure was heated electrically, in steps of about 1.4 deg., and the rise in temperature was observed with resistance thermometers. A stirrer aided in the distribution of the heat added, and itself added mechanical energy at the rate of over 60 joules per minute. The heat capacity of the calorimeter, stirrer, and accessories was calculated from the masses of the parts and their specific heats, and was considered constant over the temperature range covered. The time of electrical heating was automatically recorded with sufficient precision by a chronograph. The temperature scale and the electrical units employed were substantially those now in use by international agreement.

The evaluation of the correction which includes thermal leakage and pump energy was carried to considerable lengths theoretically, but there is some doubt that the values so obtained were sufficiently accurate. No statement was made concerning the constancy of the pump speed.

In an effort to gain sensitivity, the measuring current through the thermometer was increased to 0.08 ampere. Such a procedure throws doubt upon the accurate indication of temperature changes to 0.0001 deg., because of the heating of the thermometer itself. This suspicion is supported by the fact that a considerable rate of stirring was required in order that the temperature change of the calorimeter would appear regular during the periods when no energy was being supplied in the heating coil. The temperature intervals were so small that the assumption of the constancy of the specific heat in any interval was entirely justified.

The capacity of the calorimeter was chosen so that the mass of the water sample was large compared to the water equivalent. The calorimeter was not tightly sealed and there was the possibility of the loss of water vapor. Great care was taken with the electrical measurements, and the only questionable point here is the heating of the standard 0.1-ohm resistor due to the current in it, during observations of the current through the heater.

From a total of 111 experiments, 37 were rejected because of known faults in manipulation, etc. An empirical equation of specific heat as a function of temperature was fitted to the remaining 74 points by the method of least squares. The consistency of the experimental points is indicated by the fact that they lie so well on a smooth curve, the average deviation from the curve being 1 part in 2350.

Their equation

$$C_p = 4.20477 - 0.001768\ \theta + 0.000026447\ \theta^2 \ldots [34]$$

yields on integration with respect to temperature

$$\int_1^2 C_p d\theta = H]_1^2 = 4.20477\theta]_1^2 - 0.000884\ \theta^2]_1^2 + 0.000008816\ \theta^3]_1^2 \ldots\ldots [35]$$

[25] Sitzungber. d. Berl. Akad. (1915), p. 424.

The values of changes in heat content along the atmospheric pressure path can be computed from this equation. Corrections to the saturation path are the same as those given in the discussion of the work of Barnes.

Table 4 shows in column 2 the values of H for water under a constant pressure of 1 atmosphere from the above equation, in column 3 the corrections necessary to reduce these to saturation pressures, and in column 4 the values thus reduced.

TABLE 4 HEAT CONTENT FROM JAEGER AND VON STEINWEHR'S SPECIFIC-HEAT EQUATION

Temp. θ deg. cent.	Heat content (P = 1 atm.) H_1 atm.	$\left(\frac{\partial H}{\partial P}\right)_\theta (1 - \pi)$	Heat content ($P = \pi$) H
	International joules per gram		
10	41.97	0.10	42.07
20	83.82	0.10	83.92
30	125.59	0.09	125.68
40	167.34	0.08	167.42
50	209.13	0.08	209.21

2—DETERMINATIONS OF LATENT HEAT

(1) Dieterici

Using a Bunsen ice calorimeter, Dieterici[26] made twenty determinations of the latent heat of water at 0 deg. cent. Considering the early date of this work and the comparative simplicity of his apparatus, the results seem remarkably good. He vaporized the water at pressures slightly below the vapor pressure at 0 deg. cent. from small containers of glass or of platinum surrounded by a considerable quantity of mercury, and observed the increase in volume of the ice-water mixture in the calorimeter around the mercury. Several errors could have affected the final results. Some water could have been lost as vapor from the capsule between the time it was weighed and the time it was connected to the lead-in tube in the calorimeter. Some was doubtless lost as vapor to fill the tube system itself. The adiabatic expansion of the air in the system could have produced some cooling, though Dieterici states that this was negligible. His data for the evaluation of the heat leaks must be taken as given because of the lack of the details of their computation. There is an uncertainty as to the amount of energy involved in the expansion of the vapor after all the liquid had evaporated. Uncertainties in units can be avoided by using the conversion factor $K = 0.27024$ international joules per milligram of mercury as derived above and based on his own determinations of the mean calorie and the Bureau of Standards value of the electrical equivalent of the mean calorie. The twenty experiments thus reduced yield a mean value of 2490.6 international joules per gram as the latent heat at 0 deg. cent. The average deviation of the individual experiments from this mean is 1 part in 847.

(2) Griffiths

Among the earlier determinations of latent heat are those of Griffiths.[27] He reported determinations by a vaporization method under reduced pressures at 30 deg. and 40 deg. That part of his apparatus in which the thermal changes were isolated and observed, consisted essentially of a small oil bath, mechanically stirred and electrically heated, from which water was vaporized. This unit was located inside an iron box containing a large quantity of mercury for thermal ballast and for activating the thermoregulator which automatically maintained the temperature of the surrounding water bath. Temperatures were observed with mercury thermometers carefully calibrated and corrected. Temperature differences between the calorimeter and its surroundings were measured with differential platinum resistance thermometers.

[26] *Ann. Phys.*, vol. 273 (1889), pp. 494–508.

[27] Phil. Trans. Roy. Soc. London, vol. A186 (1895), part I, 261–342.

Because of the method of operation adopted, namely, the evaporation of the entire water sample in the calorimeter, the end effects were rather large and troublesome. The energy supplied by the stirrer amounted to about 1 per cent of the total and could be determined to within 1 part in 50 of itself. Griffiths considered this the limiting error in his experiments. The heat leak was on the average about 0.1 per cent of the total energy change involved, a fact which increases the confidence in the final results.

By passing the vapor through 18 ft. of silver tube in the oil of the calorimeter proper, the danger of priming was lessened. The possibility that the vapor was superheated was small, and the likelihood of error from this source is not large.

The greatest uncertainty in the reduction of these data to a comparative basis lies in the evaluation of the Clark cells used, in terms of present international volts. Assuming that his cells were made up according to the specifications of the Chicago Congress of 1893, their e.m.f. on the present basis becomes 1.4328 international volts.[21] Since Griffiths assigned the value 1.434 volts to his cells at 15 deg., his results as given must be reduced by 1 part in 600 for conversion to present units. The individual experiments have also been corrected to even temperatures with the aid of the Bureau of Standards values of $dL/d\theta$. The mean results from 7 experiments near 30 deg. and 11 near 40 deg. are

$$L_{30^\circ} = 2426.0 \text{ international joules per gram}$$
$$L_{40^\circ} = 2400.5 \text{ international joules per gram}$$

At 30 deg. the individual experiments are in very good agreement, the average deviation from the mean being 1 part in 5170. At 40 deg. the average deviation from the mean is 1 part in 1500. In spite of the greater number of experiments at 40 deg., it appears that the mean value at 30 deg. is probably the more accurate of the two.

(3) A. W. Smith

A. W. Smith[28,29] has made determinations of the heat of evaporation of water at five temperatures below 100 deg. He lays considerable stress on the advantages of the method which he used of allowing quiet evaporation to proceed in a stream of air passing over the surface of the evaporating fluid. In the earlier work the stream of air was allowed to bubble through the water, and the stirring was accomplished in this manner. In these experiments the corrections for thermal leakage were considered negligibly small. In the later experiments which constituted a determination of the latent heat at 100 deg cent., the stirring was accomplished mechanically and the air did not bubble through the liquid. Experiments were run at various rates and the thermal leakage and pump energy effects were thus eliminated, provided they remained the same in any two experiments which were combined. No correction was applied for the heat effect involved in the expansion of the vapor from saturation pressure down to the partial pressure which it exerted in the mixture of air and vapor. In the reports mentioned above there are not sufficient data on the rate of passage of air to permit calculation of this correction. Professor Smith's privately communicated estimate is that, in the experiments at 100 deg. cent., the amount of air was not greater than 1 liter for 5 grams of water evaporated. If this maximum amount flowed in all the experiments at 100 deg., the value of the latent heat from each should be decreased by about 1 joule per gram. For all the determinations below 100 deg. where the specific volume of the vapor is relatively large, this correction is doubtless negligibly small.

The measurements of temperature and of power supplied to the calorimeter were sufficiently precise, so that they do not limit the accuracy of the final results.

The masses of the water evaporated were carefully determined, but where the samples were as small as 2 or 3 grams, as they were in all the earlier experiments, the absolute accuracy becomes more important and is somewhat open to question. No mention is made of the means employed to keep the pump power constant in the stirred calorimeter. A comparison of the results of the individual experiments does not indicate a high degree of reproducibility. For instance, at 100 deg. cent., where the consistency is best, the average deviation from the mean is 1 part in 1000.

A series of four determinations at 100 deg. was made with active boiling and very small flows of air. The average result of this series is 16.7 joules per gram lower than the average of a large series in which the evaporation took place quietly. This low result is explained as being due to the carrying over of unevaporated droplets of water in the air-water stream.

To the reviewer it appears that the danger of priming has often been greatly overestimated. The experimental values for the latent heat are, in nearly every case at hand, independent of the rate of withdrawal of vapor. Mathews[30] records test experiments which indicate the absence of unevaporated droplets in a region above the surface of actively boiling water.

The details of the latter series of experiments by Smith seem too meager to justify his explanation of the low results. For the same reason the formulation of an alternative hypothesis is precluded.

In Table 5 are given the results of Smith's experiments.

TABLE 5 LATENT HEATS FROM SMITH'S EXPERIMENTS

Temperature, deg. cent.	Latent heat, Int. joules per gram
14	2463.5
21	2447.6
28	2431.1
40	2401.4
100	2261.6

The data recorded in the report of 1907 have been converted into present international joules by decreasing the values as recorded in terms of the Clark cell at 15 deg. = 1.434 volts by 1 part in 600 in the latent heats, to conform to the value 1.4328 international volts as previously used in this paper. Corrections to the temperatures indicated have been made using the Bureau of Standards values for $dL/d\theta$.

Application of the above-mentioned correction of one joule per gram for the superheating of the vapor due to its drop in pressure would reduce the latent heat at 100 deg. to 2260.6 international joules per gram. Since one joule per gram is the maximum to which this correction could have amounted, it can be stated only that according to the best information from Smith's experiments the latent heat at 100 deg. is not less than 2260.6 international joules per gram.

(4) Richards and Mathews

Determinations of latent heat at 100 deg. by Richards and Mathews[31] and by Mathews,[30] using a condensation method, were carefully performed. This method possesses the obvious disadvantages that accurate knowledge of the heat capacity of the calorimeter and of the heat content of water is necessary for the reduction of the data. Mathews points out an uncertainty in the heat capacity of his calorimeter equivalent to 1 part in 500 in the latent heat, and calls attention to the fact that the error from this source in the earlier experiments was probably 2 parts in 500. Both papers assume a mean

[28] *Phys. Rev.*, vol. 25 (1907), pp. 145–170.
[29] *Phys. Rev.*, vol. 33 (1911), pp. 173–183.
[30] *Jl. Phys. Chem.*, vol. 21 (1917), pp. 536–569.
[31] *Jl. Am. Chem. Soc.*, vol. 33, (1911), pp. 863–888.

specific heat for water between 20 deg. and 100 deg. of 1.0012 times the specific heat at 21.4 deg.

Because of so-called "premature condensation" it was found necessary to extrapolate linearly the values of latent heat plotted against rate of condensation to an infinite rate. Consideration of heat leak alone, which disappears at the infinite rate, forms a more logical basis for such a procedure. It is assumed in the following that the same method of operation and reduction of the data was used by Mathews as that employed in the earlier work. Heat leak was measured experimentally with no flow of steam through the tube connecting the evaporator and the condenser. The value so determined is greater than the true heat leak during an experiment because of the change in gradient in the connecting tube due to the passage of hot steam through it. On account of the warming of the lower part of the tube thus produced, there is less heat flow down it from the evaporator to the calorimeter.

Let—

- Q = heat transferred to the calorimeter by the condensing steam corrected for everything except the heat leak during the flow period
- A = heat leak per minute as determined experimentally without flow and as actually used in their reduction of the data
- B = the true heat leak per minute with steam flowing
- L_c = latent heat as Richards and Mathews calculated it
- L = value of latent heat as here recalculated
- M = mass of vapor condensed, and
- t = time during which steam was flowing

Then

$$L_c = \frac{Q - At}{M} \quad \text{and} \quad L = \frac{Q - Bt}{M} \qquad [36]$$

Subtracting,

$$L - L_c = \frac{(A - B)t}{M} \qquad [37]$$

or

$$L = L_c + \frac{(A - B)t}{M} \qquad [38]$$

As suggested above $(A - B)$, is a positive quantity and, over the moderate range of flows used in the experiments, is probably constant. Therefore the values of L_c as calculated by Richards and Mathews are always lower than the true experimental values by the amount $(A - B)t/M$. The quantity t/M is the time required for condensation of one gram and is the abscissa of the curve used by them in the extrapolation to infinite flow. As the value of t/M increases, that is, the slower the rate of condensation, the more does L_c deviate linearly from L. It follows that the method of correction to the infinite rate was correct, but that the theory of "premature condensation" was superfluous. The question of the deviations from the straight line, which might appear if the flows could be made to approach infinity, does not enter, because the slope only is of importance in the calculation of the results.

The data from the 1911 paper have been recomputed, using the correction to the water equivalent which Mathews later suggested. The data from both sets of experiments have been calculated, using the Bureau of Standards values for the heat content of water.

Because two different vaporizers were employed in the earlier work, the data must be treated in two parts. The results of the recalculation are given in Table 6.

TABLE 6 LATENT HEATS FROM EXPERIMENTS OF RICHARDS AND MATHEWS

Year	Latent heat at 100 deg. L Int. joules per gram
1911	2256.5
1911	2253.5
1917	2255.2

It appears that it is the systematic errors, especially the uncertainties in the heat capacities of the calorimeters, which limit the accuracy of the final results of these experiments. Professor Mathews expressed the opinion, in a private communication, that the values of the heat capacities of the calorimeters as used in these calculations are slightly low, but he did not wish to suggest definite numerical changes in them. As a heavier vaporizer was used in the second series of 1911, it may be that the water equivalent in this set should have been corrected by more than the two grams originally suggested by Mathews.

(5) Henning

Among the more important determinations of the latent heat of water are those of Henning.[32,33] His method consisted essentially of vaporizing the water from a calorimeter in a thermostatted bath and collecting the distillate in a condenser vessel in which the pressure was maintained as nearly as possible equal to the pressure in the calorimeter by means of an atmosphere of inert gas. Determinations at eleven different temperatures between 30 deg. cent. and 180 deg. cent. are recorded. His own reduction of these values is summarized in terms of present international units in a brief report published in 1919.[34] This summary gives values of latent heats at 10-deg. intervals from 30 deg. to 180 deg., obtained by smoothing the experimental values and interpolating with the aid of empirical equations at intermediate temperatures. His own choice of equations to best fit his data is considered adequate, and probably better than one made by one less familiar with the work. Among the individual experiments at any given temperature there is rather large variation, but in general the experiments were so numerous that the effect of accidental errors on the final results has been reduced. At the higher temperatures the number of experiments was not so large and the results are consequently less certain. In the calculation of the mass of vapor which is evaporated but remains in the calorimeter to fill the space left vacant by the removal of water, an approximation is involved, but the error from this source is negligibly small below 180 deg.

Table 7 shows the values of latent heat which Henning recorded in 1919.

TABLE 7 HENNING'S VALUES OF LATENT HEAT

Temperature, θ deg. cent.	Latent heat, L Int. joules per gram	Temperature, θ deg. cent.	Latent heat, L Int. joules per gram
30	2426.0	110	2228.5
40	2403.8	120	2200.1
50	2380.8	130	2170.3
60	2357.4	140	2139.8
70	2333.1	150	2109.7
80	2308.0	160	2079.6
90	2282.5	170	2049.4
100	2255.7	180	2019.7

(6) Jakob

Using a method almost identical with that of Henning, but with a new apparatus designed for use at higher temperatures, Jakob,[35] and Jakob and Fritz[36] have continued the latent-heat determinations from 180 deg. These reports give values through 250 deg., but the experiments have since been extended to about

[33] *Ann. Phys.*, vol. 326 (1906), pp. 849–878.
[33] *Ibid.*, vol. 334 (1909), pp. 441–465.
[34] *Ibid.*, vol. 363 (1919), pp. 759–760.
[35] Forschungsarbeiten V.D.I., vol. 310 (1928), pp. 9–19.
[36] *Z.V.D.I.*, vol. 73 (1929), pp. 629–636.

310 deg. cent. This is an unusually comprehensive undertaking, the final results of which are not yet available. Therefore no attempt is made to include a comparison of such preliminary results of this work as are at hand with those of other experimenters.

(7) Carlton-Sutton

The determinations by Carlton-Sutton[37] of the latent heat at 100 deg. deserve special prominence among the other investigations of this important constant. Using a modification of Joly's classical apparatus, he measured the heat of condensation of water directly in terms of the mean calorie without any reference to electrical standards. He weighed in place the steam which was condensed on a glass bulb to change its temperature from 0 deg. cent. to 100 deg. cent. By performing experiments, first with the bulb nearly filled with pure water and then with the bulb containing a much smaller amount of water, he was able to eliminate its heat capacity as well as many other small factors which would otherwise have appeared as corrections. The difference in the masses of steam condensed in two such experiments is that amount which would change the temperature of a mass of water corresponding to the difference of the two bulb fillings from 0 deg. to 100 deg.

If the thermal conditions during the two experiments were alike, as was doubtless very nearly the case, no corrections for heat leak are required. Corrections were applied for thermal changes occurring within the bulb and for various other small elements involved in the method. The correction for the thermal changes in the bulb has been independently recomputed and found valid. Although the water in the bulb was heated under a total pressure greater than its vapor pressure, the method of calculation of the correction takes this into account and reduces the data to mean calories defined by the saturation path.

To obtain an accuracy of 1 part in 5000 requires that the masses be accurate to 0.1 milligram. In this factor the author's estimate seems somewhat optimistic. He considers that the formation of a vapor cloud upon admitting the steam into the condensing chamber at 0 deg. is the limiting error. It seems probable that the error in determining the masses is at least as great.

The deviations among the individual determinations indicate a high degree of experimental precision.

The final reported value of 538.88 mean calories, when interpreted in terms of the Bureau of Standards mechanical equivalent of 4.1875 international joules per gram, yields for the latent heat of water at 100 deg. the value 2256.6 international joules per gram.

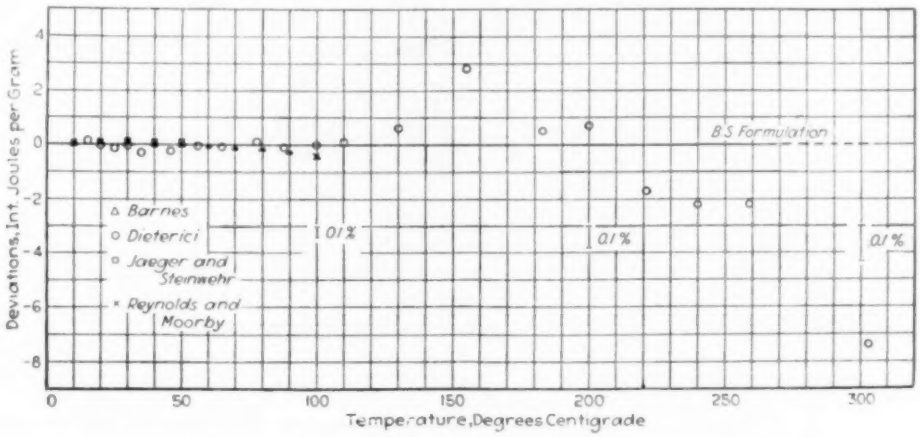


Fig. 1 Deviation Chart—Heat Content of Liquid (Experimental)

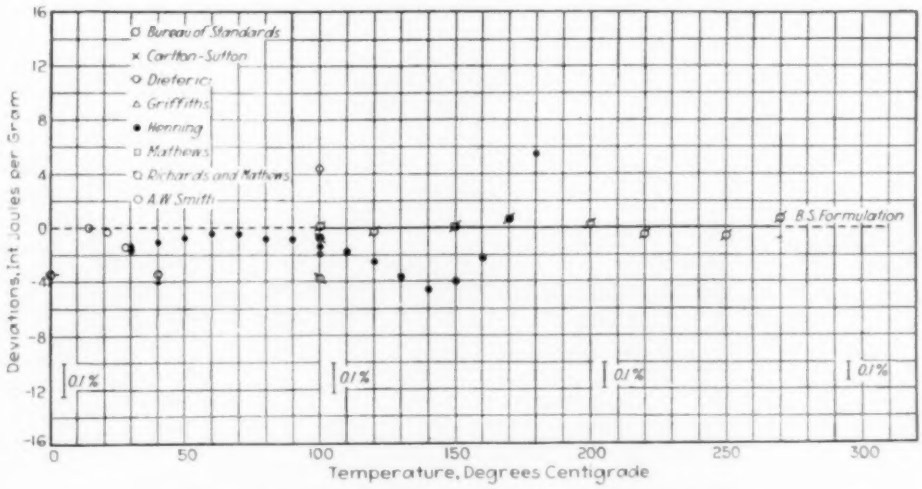


Fig. 2 Deviation Chart—Latent Heat (Experimental)

[37] Proc. Roy. Soc. London, vol. A93 (1916–17), pp. 155–176.

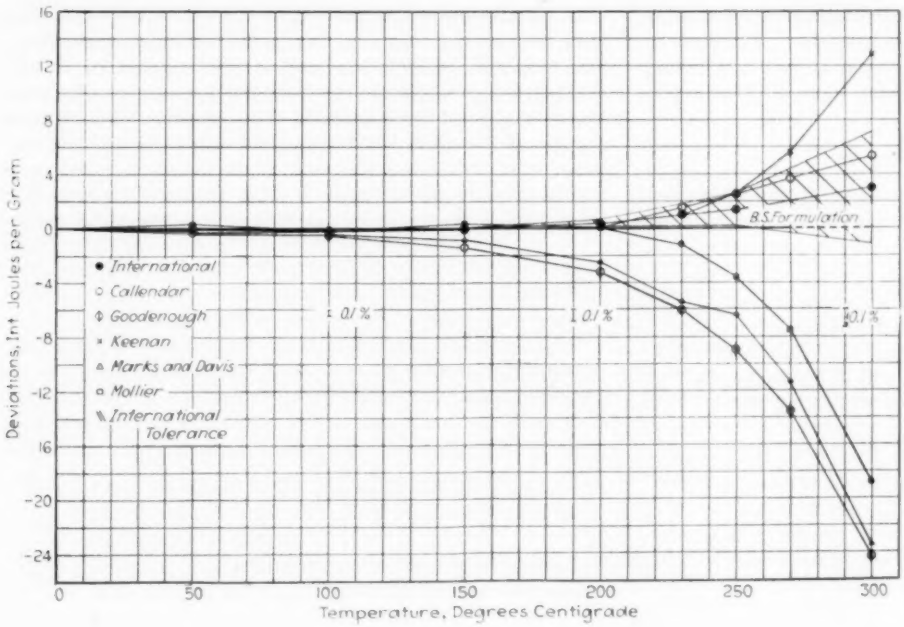


FIG. 3 DEVIATION CHART—HEAT CONTENT OF LIQUID

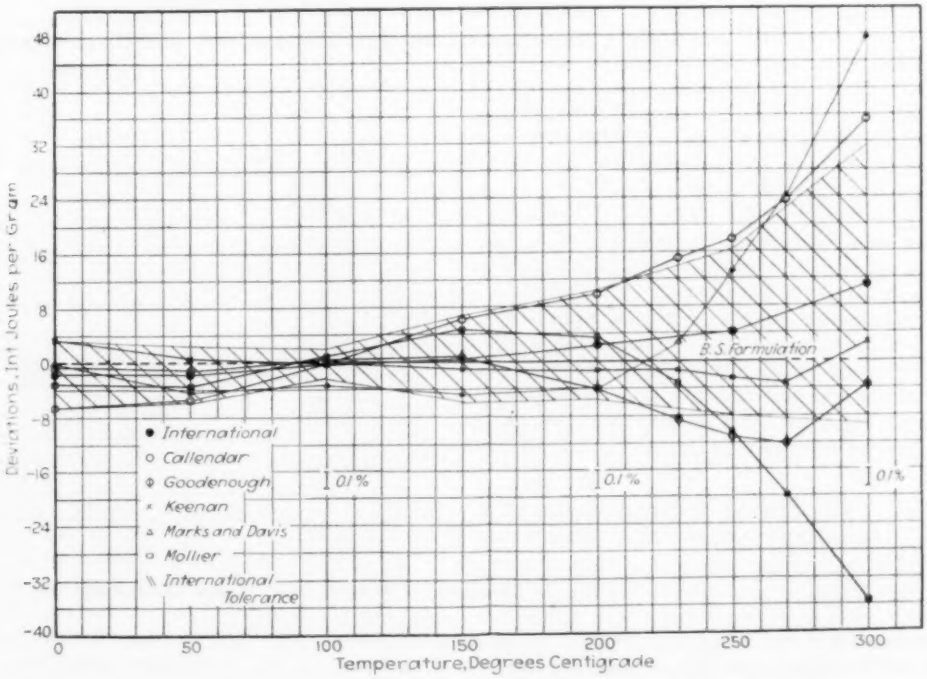


FIG. 4 DEVIATION CHART—HEAT CONTENT OF VAPOR

IV—INTERCOMPARISON OF EXPERIMENTAL DATA

To aid in the comparison of the data they are assembled in tables and deviation charts. Deviations from the Bureau of Standards values as bases are plotted to facilitate the comparison of these more recent data with those published previously.

In Table 8 are shown the values of heat content of the liquid, assembled from the previous sections. By the method of computation, the result of Dieterici is made to agree exactly with that of the Bureau of Standards at 100 deg. Because of the uncertainty in the interpretation of the e.m.f. of the Clark cells used by Barnes, his results, expressed in international joules, involve an uncertainty of considerable magnitude. The results of Callendar are omitted in this table, but appear later in the comparisons of the steam tables. The value in the Bureau of Standards column at 303 deg. is extrapolated and is therefore put in parentheses.

Fig. 1 shows the deviations of the previously published values of heat content of the liquid from the Bureau of Standards values.

In Table 9 are assembled the values of latent heat. Henning's values and those of the Bureau of Standards are calculated from empirical equations which were fitted to the data. The rest of the values are experimental points not smoothed by equations or graphically. The values in parentheses are calcu-

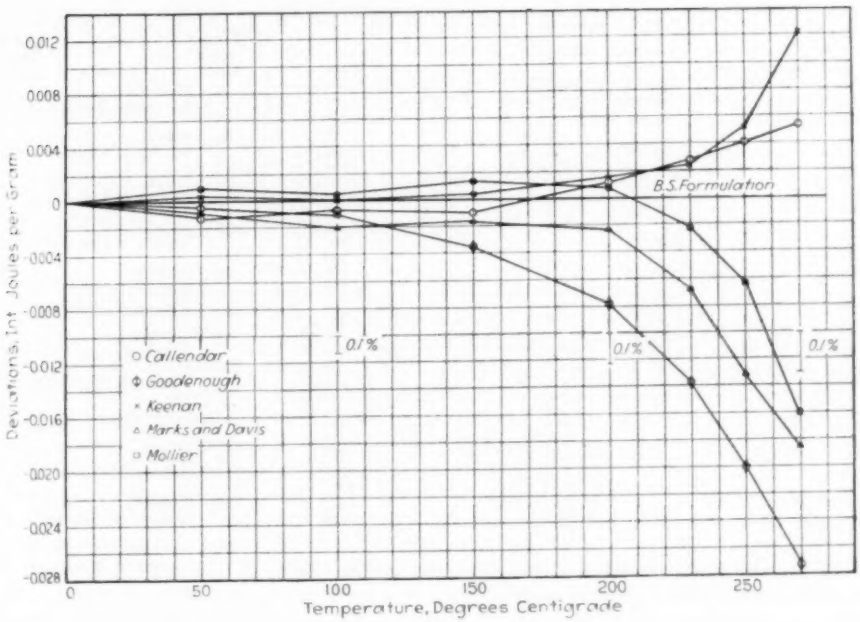


FIG. 5 DEVIATION CHART—ENTROPY OF LIQUID

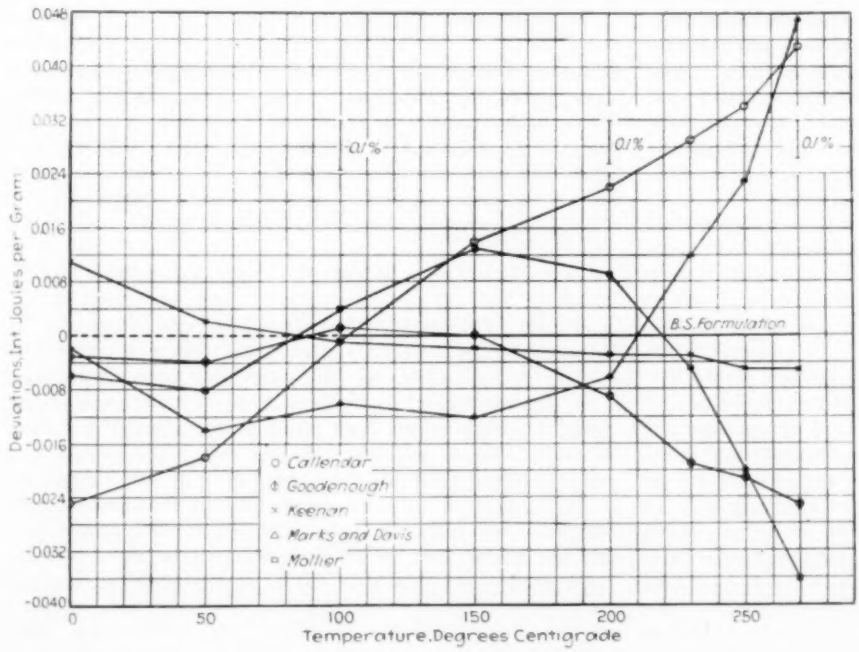


FIG. 6 DEVIATION CHART—ENTROPY OF VAPOR

lated from the Bureau of Standards latent-heat equation outside the range of the experiments.

Fig. 2 shows the deviations of the experimentally determined values of latent heat from the Bureau of Standards latent-heat equation.

V—COMPARISON OF STEAM TABLES WITH BUREAU OF STANDARDS RESULTS

For the purposes of this comparison, deviation charts alone are considered adequate. In Figs. 3 to 6, which are largely self-explanatory, the base lines are broken in the regions not covered by the experiments. The entropy charts are included because values of entropy can be calculated directly from the Bureau of Standards calorimetric measurements.

VI—CONCLUSIONS

For the most part the reviewer believes that the agreement among the experimental results which have been compared is within the limits of the total error involved in the determinations themselves and the conversions to present units.

Referring to the heat-content determinations, the deviation of the Bureau of Standards values from those of Barnes and of Jaeger and von Steinwehr and also of the latter two from each other are, in places, somewhat greater than can be embraced by the estimates which these authors make of their own precision.

The comparisons indicate that the Bureau of Standards latent-heat equation can be used with considerable confidence in the range 0 deg. to 100 deg., though it was derived solely on the basis of the experiments in the range 100 deg. to 270 deg.,

TABLE 8 SUMMARY OF RECALCULATED RESULTS ON HEAT CONTENT OF THE LIQUID

Temp., deg. cent.	Observer: Reynolds and Moorby	Barnes	Dieterici	Jaeger and von Steinwehr	Bureau of Standards
	International joules per gram				
10	....	42.04	.....	42.07	42.02
15	....	....	63.10	....	62.94
20	....	83.91	83.82	83.92	83.83
25	....	....	104.6	....	104.72
30	....	125.66	125.6	125.68	125.59
35	....	....	146.2	....	146.48
40	....	167.36	.....	167.42	167.34
46	....	....	192.2	....	192.41
50	....	209.11	.....	209.21	209.11
56	....	....	234.2	....	234.19
60	....	250.85	.....	....	250.90
65	....	....	271.8	....	271.84
70	....	292.64	.....	....	292.75
78	....	....	326.4	....	326.29
80	....	334.51	.....	....	334.66
88	....	....	368.2	....	368.27
90	....	376.39	.....	....	376.65
100	418.33	418.30	418.75	....	418.75
110	....	....	461.1	....	461.0
130	....	....	546.6	....	545.9
156	....	....	660.6	....	657.8
183	....	....	776.5	....	776.0
200	....	....	852.6	....	852.0
221	....	....	946.1	....	947.8
240	....	....	1034.7	....	1037.0
259	....	....	1126.7	....	1128.9
303	....	....	1348.9	....	(1356.3)

TABLE 9 SUMMARY OF RESULTS ON LATENT HEAT

Temp, deg. cent.	Observer	Latent heat, Int. joules per gram	Temp., deg. cent.	Observer	Latent heat Int. joules per gram
0	Dieterici	2490.6	120	Henning	2200.1
	Bureau of Standards	(2494.0)		Bureau of Standards	2202.7
14	Smith	2463.5	130	Henning	2170.3
	Bureau of Standards	(2463.5)		Bureau of Standards	2174.0
21	Smith	2447.6	140	Henning	2139.8
	Bureau of Standards	(2447.9)		Bureau of Standards	2144.4
28	Smith	2431.1	150	Henning	2109.7
	Bureau of Standards	(2432.2)		Bureau of Standards	2113.8
30	Griffiths	2426.0	160	Henning	2079.6
	Henning	2426.0		Bureau of Standards	2081.9
	Bureau of Standards	(2427.7)	170	Henning	2049.4
40	Griffiths	2400.5		Bureau of Standards	2048.7
	Henning	2403.8			
	Smith	2401.4	180	Henning	2019.7
	Bureau of Standards	(2404.9)		Bureau of Standards	2014.1
50	Henning	2380.8	200	Bureau of Standards	1939.9
	Bureau of Standards	(2381.6)	220	Bureau of Standards	1857.9
60	Henning	2357.4			
	Bureau of Standards	(2357.9)	250	Bureau of Standards	1715.6
80	Henning	2308.0	270	Bureau of Standards	1603.5
	Bureau of Standards	(2308.8)	...	.....	....
			...	.....	....
90	Henning	2282.5	...	.....	....
	Bureau of Standards	(2283.4)	...	.....	....
			...	.....	....
100	Henning	2255.7	...	.....	....
	Smith	2261.6	...	.....	....
	Richards & Mathews	2256.5	...	.....	....
	Richards & Mathews	2253.5	...	.....	....
	Mathews	2255.2	...	.....	....
	Carlton-Sutton	2256.6	...	.....	....
	Bureau of Standards	2257.2	...	.....	....

and on the classical conception of the vanishing of the latent heat at an infinite rate at the critical point.

The comparison of the steam tables indicates that the divergence in the tabulated properties of saturated water and steam is considerable. None of these tables comes completely within the limits of tolerance of the International Skeleton Steam Table of 1929, and therefore none can be called an "international table" as specified by the London Conference. A marked improvement is evident in the tables as more and more data of a precise nature became available for use in the formulations. It is believed that, with the progress in experimental equipment and technique, and the mutual understanding of the investigators, the evolution of accurate international tables of the thermal properties of water and steam has progressed beyond the stage of idealism, and that such tables will be a reality before very many years have passed.

FSP-52-31

Effect of Large Boilers at High Capacities on Operating Characteristics and Investment in Boiler Plants

By FRANK S. CLARK,[1] BOSTON, MASS.

This paper treats of the various factors that enter into the design of boiler plants, their effect on investment costs and on operating characteristics. It traces the development of the boiler plant from a number of small units operated at moderate capacities to the present plant of a few large units run at high rates of evaporation. It gives the conditions that limit the sizes and output and shows how, as the demand for larger units increases, these conditions have been overcome. The paper shows the effect of large units in reducing the sizes of buildings to house them, and the number of auxiliary equipment to serve them. It shows the effect of reliability of equipment on increased service factor, with consequent reduced necessity for spare equipment. It states that the governing factors in investment costs for a given condition are the size of the unit and the output obtainable, the former depending on the ability to manufacture, and the latter on the fuel that can be burned in the furnace and the ability of the unit to absorb the heat. Further, it discusses the proportioning of heat-absorbing surfaces between boiler, economizer, and air preheater to obtain the most economic result at minimum cost, and also the effect on operation of large units run at high capacity, and in turn their effect on labor and maintenance cost.

IN THESE days when attention is being concentrated on anything and everything that affects the cost of design and construction of a steam power station or a boiler plant, with a view to keeping investments down to an economical minimum, it is proper and timely that the steam-generating plant should be made the subject of analysis. The chief item of equipment in such a plant is, of course, the boiler or steam-generating unit, but of importance also, and affecting the size, capacity, and type of the unit, are the fuel-burning equipment and the other heat-absorbing units installed.

The trend of steam-generating equipment, like that of electric generators, has been toward increased size and output, and has followed the development of the industry. Limitations other than the ability of the boiler to generate steam held the sizes down, and at about the time of the introduction of the steam turbine there were a number of boilers serving one unit. This frequently necessitated the placing of the boiler-firing aisles at right angles to the axis of the electric-generating room, thus dividing the boiler-operating room into a number of separate operating aisles. In congested city districts it was necessary even to resort to double decking or placing boilers on two floors, one above the other, and in at least one station of this period the boilers were arranged on three different floors. The congestion of equipment, complications of installation, and difficulties of operation due to such conditions are apparent.

This situation was brought about largely by conditions other than the ability of boilers to produce steam at a high rate of generation. Inability of combustion facilities to burn the fuel in sufficient quantities, and of the furnace walls to withstand the punishment due to high rates of combustion, limited the capacity of boilers. Impure feedwater and the maintenance of grates and walls required frequent outages for cleaning and repairs, thus increasing the number of boilers necessary to insure reliability of operation.

The advent of the mechanical stoker for coal, and of the mechanical-atomizing oil burner, bringing with them the use of forced draft, increased the fuel-burning ability, and placed the limitation of capacity on the ability of solid refractory furnace walls to withstand the punishment incident to high rates of combustion. To offset this limitation, first the air-cooled and then the water-cooled furnace was developed. Added momentum to increased rates of evaporation has been given by the use of pulverized coal. Greater service reliability has accompanied these developments, increasing the period of continuous high-capacity operation until service factors exceeding 90 per cent have been attained by a number of large boiler installations, which is comparable to the service obtained from large turbine generators. This has reduced the necessity for spare steam-generating equipment to a point where it is possible to design a plant with a single boiler serving one turbine-generating unit, or with no greater reserve in boiler plant than in turbine plant.

With the removal of the restrictions other than those imposed by the steam generator itself, and with the increasing capacities of plants and of individual units, there have followed demands for boilers of larger and larger steam-generating capacities, until at the present time there are in operation in this country a number of boilers with steaming capacities of over 400,000 lb. per hr. each, with some few on order or being erected whose output will be upward of 800,000 lb. per hr. This increase in steam capacities has been accompanied by increases in steam pressure and temperature which, with greater efficiency of steam utilization, have decreased greatly the amount of steam required per kilowatt-hour of electrical energy produced.

For a given condition investment costs for a boiler plant are determined largely by two considerations: size of individual steam-generating units, and output obtainable from each unit. The first has as its maximum limit the ability of the manufacturer to produce the unit, and as a governing limit the relation of the individual units to the plant as a whole, to the characteristics of load, and to the spare units deemed necessary to provide reliable and continuous operation. The second is governed by the fuel that can be burned in the furnace, and the ability of the unit to absorb heat for the production of the steam.

The size and number of boilers affect the cost, not only of these units themselves, but of the equipment that goes with them and of the building that houses them. For a given output the percentage of building space devoted to aisles and firing floors de-

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creases appreciably with increase of size of boilers, affecting materially the area of the building and its cubic contents, with consequent influence on the cost of the structure. Supporting steel and its foundations likewise become less costly as sizes of boilers increase and their numbers decrease. Piping for both steam and water becomes larger in size, but of less number of parts, resulting in simplification in arrangement and reduction in cost. Equipment for fuel handling and burning is similarly affected. Setting walls have less area per unit of furnace volume as the sizes of combustion spaces increase, and this has a marked effect on the cost of the installation. Boilers themselves are less expensive per unit of output as their sizes increase. Fans, flues, economizers, air heaters, and other auxiliary equipment are reduced in unit cost with increase in capacity and decrease in number.

To illustrate these statements Table 1 has been prepared which gives a comparison of building areas and investment costs of boiler plants designed for a given steam output and pressure, but with different numbers of steam-generating units installed. The table also gives the proportional costs of main items of construction and equipment, indicating the relation of each item to the total cost. In this comparison are prorated so-called overhead costs, which include in each case an item to cover possible contingencies. The estimates are comparable as to scope and hold good for the case assumed, but many vary somewhat for a similar analysis of a different set of conditions.

TABLE 1 BUILDING AREAS AND INVESTMENT COSTS OF BOILER PLANTS

Number of boilers	4	6	8	12
Sections per boiler	60	40	30	20
Relative building area, per cent	100.00	112.00	127.00	133.00
Relative investment cost per cent	100.00	112.00	124.00	142.00
Proportional cost of main items of construction and equipment:				
Building and foundations	16.10	16.50	16.70	17.10
Boilers, superheaters, settings, etc.	32.00	33.00	34.60	38.00
Draft and feedwater systems	22.40	22.40	22.00	19.80
Piping	9.60	9.50	9.20	9.00
Fuel and ash handling	7.60	7.40	7.20	6.70
Fuel preparation	9.60	8.60	7.90	7.00
Auxiliaries	2.70	2.60	2.40	2.40
Totals	100.00	100.00	100.00	100.00

With small boiler units, low construction costs, low steam pressure, operation at moderate rates of output, and with fuel not so important an item of production cost as it is at present, the boiler plant of fifteen years ago consisted usually of the boiler as the only piece of heat-absorbing equipment. The required draft was produced by a stack of reasonable height. No other equipment except possibly a superheater to give a moderate steam temperature was necessary to gain the desired efficiency. With present-day sizes and capacities, the situation becomes much more complex. The problem now is not solely one of absorbing the heat of combustion, but of the manner in which this is to be done and by what media. Higher steam pressures have raised the temperatures of gases leaving the boiler, as have the higher outputs, and additional equipment has become necessary to recover the heat in the gases, which has brought about extensive use of the economizer and the air heater for these purposes. Within the limits imposed by their operating characteristics, the selection of a size and arrangement of heating surface for each type of equipment must be such as will give the desired overall results for a minimum of cost.

Limitations to size of individual units would appear to be the length of drum or drums which it is possible to manufacture and to support properly. For the riveted type, which can be used for pressures up to 700 lb., it is possible to obtain drums accommodating sixty 4-in. tubes spaced $7^3/_4$ in. between centers with only end suspensions necessary, giving a furnace width of about 39 ft. For pressures involving the use of forged drums, the limit is the size of the ingot from which the drum is forged. At present this would allow a furnace width of about 43 ft. On account of the thickness of the drum shell the suspension is not a serious problem.

With these sizes and the ability to burn the necessary amounts of fuel under them, outputs of more than 500,000 lb. of steam per hour are obtainable from the so-called standard types of boiler. By combining two of these units with one furnace much greater capacities can be obtained, with consequent saving in space occupied and in investment cost. Boilers are now in operation and in process of installation that will develop more than 800,000 lb. of steam per hour.

The limits to the capacity that is obtainable are the fuels it is possible to burn in the furnace, and the ability to maintain proper circulation of water and to deliver the steam without accompaniment of excessive moisture. In general, it may be said that surfaces can be provided to absorb all the heat that can be liberated in a given furnace, and that present designs can recover economically. It therefore becomes a question of method of fuel burning. Fuel burned in suspension as gas, oil, or pulverized coal permits the use of highly preheated air, of rapid and turbulent mixture of air and fuel, of ready and easy regulation of combustion through control of air and fuel to each burner, and efficient utilization of the whole furnace volume. The burning of coal on a grate involves the problem of supplying and regulating a fuel bed of large proportions, together with an air supply at proper pressures and in sufficient quantities to all parts of the bed. This type of combustion equipment also affects the proportioning of heat-absorbing surfaces by the necessity of limiting the preheating of the air to a temperature which will not cause binding of moving parts of the grate or burning of the parts themselves. With both these methods of firing there arises the serious problem of limitation and loss of capacity at high rates of combustion due to clogging of the passages around the tubes by molten ash carried by the gases freezing to the tubes. This becomes more and more serious as the fusing temperature of the ash becomes less, and must be met by the arrangement of tubes that will provide wide spaces between them with consequent slower velocities of entering gases.

It has been stated that the availability for service of large boiler units has become as high as that of the turbine units which they serve. For purposes of reliability and also to take care of variations in load, auxiliary equipment serving the turbine is practically always in duplicate. For the same reasons and also because of practical limitations of sizes obtainable, mechanical-draft equipment is being duplicated in the case of large boilers.

Decrease in number of units means less equipment to operate and maintain, consequently labor and maintenance costs decrease correspondingly. Introduction of automatic-control equipment, and the use of an adequate number of indicating and recording instruments properly placed with their dials and charts congregated so as to give at one point an indication of the performance of a boiler unit and its auxiliaries, are a further aid to decreased operating labor and increased efficiency of operation. While the use of automatic control can be overdone, in general some degree of such control is essential to the realization of the full operating possibilities of large boiler units, and to the reduction of operating labor to a minimum.

Discussion

J. W. ANDREE.[2] The author's statement that it is no longer necessary to have spare steam-generating equipment in modern plants or that the reserve in the boiler plant need not exceed the

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capacity of the turbine plant is borne out by experience in the Long Beach No. 3 Plant, which went into operation June 20, 1928, and has operated almost continuously from that time to date. The 94,000-kw. turbine in this plant is served steam from three boilers with practically no spare steam-generating capacity. This unit is operated as a base-load unit with a net output of 100,000 kw. continuously.

Excluding preliminary operation during its first month of service, this unit has operated a total of 10,592 hours out of a possible 11,688 hours up to December 1, 1929, or a total out-of-service time of 1096 hour. Of this total out-of-service time, 809 hours was forced out-of-service time, all due to turbine or its auxiliaries and none due to the boiler plant. In addition to the total outage there was 791 hours of reduced-load operation due to boiler or boiler-auxiliary trouble and 1252 hours of reduced-load operation due to turbine or turbine-auxiliary trouble.

Practically all of the outage in the boiler room was caused by draft-fan trouble which could have been greatly reduced with duplicate equipment. There were several cases of split tubes which would have caused a total outage in a plant of one boiler only.

That partial outage time due to turbine or turbine auxiliaries consists of 840 hours of operation at reduced load during a period of blading trouble; the remainder is made up largely of condenser trouble.

J. B. Crane.[3] The paper brings up a subject that is most timely, now that boilers which will deliver 1,000,000 lb. of steam each are in satisfactory operation.

The author, in Table 1, has prepared a summary showing the relative investment cost for four to twelve boilers. The writer gives the same table arranged in a slightly different form (see Table 2), which indicates that the main increase for small boilers is in building, foundations, boilers, superheaters, and settings; and it will be noticed that on these two items increased cost for twelve boilers over four is roughly 50 per cent. This indicates how misleading it is to base capital charges on the horsepower basis or 1000-lb. steam basis and how necessary it is to secure reliable figures on cost in making any comparison of different sizes of equipment. The same thing applies to different pressures.

TABLE 2

Number of boilers	4	6	8	12
Relative investment cost, per cent	100	112	124	142
Buildings and foundations	16.10	18.4	20.4	24.3
Boilers, superheaters, settings, etc	32.00	37.0	42.9	54.0
Draft and feedwater systems	22.40	25.2	27.4	28.2
Piping	9.60	10.7	11.7	12.7
Fuel and ash handling	7.60	8.3	8.9	9.5
Fuel preparation	9.60	9.6	9.8	9.9
Auxiliaries	2.70	2.8	2.9	3.4
	100.00	112.0	124.0	142.0

Higher pressure boilers are less efficient than lower pressures; for example, at 200 lb. pressure the temperature of the water in the boiler is 388 deg. fahr. and the exit temperature from the boiler may be 500 deg. fahr. If the same boiler is built for 450 lb. pressure the temperature of the water in the boiler will be 460 deg. and the exit temperature at the same rating will be 572 deg. The advantage of going to the higher pressure is in the savings effected in the turbine room, and it is necessary to install additional heat-absorbing surface in the way of economizers or air heaters to reduce the higher temperature.

In the design of steam-generating units it is found that the simplest and most economical unit is obtained by determining the amount of fuel to be burned and by designing a furnace that will most satisfactorily burn the fuel. Between the furnace and the superheater there should be installed sufficient heating surface to reduce the temperature of the gases to that required for the proper steam temperature to be delivered from the superheater. After determining the efficiency for which the unit is to be built and the exit temperature necessary to secure this efficiency, an air heater is designed for the maximum size with standard materials and the temperature determined at the entrance to the air preheater. This leaves a difference in temperature between the outlet of the superheater and the entrance to the air preheater; and sufficient absorbing surface, in the additional convection surface in the boiler or in the economizer surface, or in a combination of the two, is installed to give this temperature difference. With this design a study is then made to see that there is sufficient volume in the drums to obtain dry steam and that proper circulation will result with the heat-absorbing surfaces laid out, and a check is made to see if some other combination of economizer and air preheater is more economical.

Considerable study has been given to this type of equipment in the past few years, with the result that it has been possible to reduce the total heating surface required at least 50 per cent in order to give the same efficiency.

In an effort to reduce the amount of high-pressure surface it has been necessary to increase the draft loss, and the addition of economizers and air preheaters has meant the installation of induced-draft fans instead of natural draft.

We find the following methods are in use in comparing the fan cost on any particular job:

1 Consider the fan power as costing the same per kilowatt-hour as the cost of manufacturing current at the station, with no capital charges.

2 Consider fan power as costing the same per kilowatt-hour as sales price of the current to other customers using the same amount of current at the same load factor.

3 Consider that additional equipment must be installed in the station to take care of the fan power and that this equipment will cost the same per kilowatt as the main equipment in the station.

4 Assume that there is enough excess capacity so that it is not necessary to capitalize the power necessary for the fans.

5 Assume that this excess capacity must earn the same as equivalent power-station capacity.

It will be seen that, under the foregoing methods of figuring the cost of fan power, widely varying results are obtained as to the effect of increased efficiency as against increased fan power necessary to secure higher capacities or higher efficiencies. Otherwise, most consulting engineers use similar methods in comparing costs.

Ollison Craig.[4] The author has discussed the economy of installing large boiler units and operating them at high capacities. There is no doubt that the trend to larger sizes and greater capacities will continue. The author has pointed out present limitations to both size and operating capacities. If the trend is to continue, these limits must of necessity be removed. The whole history of engineering has consisted of a continuing series of boundaries and limitations which have been continuously passed, new limitations being erected.

It might be interesting to speculate as to the means that will extend the present limitations. This writer would like to make a guess that the boiler limitations will be extended by the use of small tubes and forced circulation, permitting more heat-absorbing surface in a given space, providing means for greater heat absorption per unit of area, and making possible design of gas passages which will maintain most nearly the ideal velocity of gases.

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[4] Consulting Engineer, Riley Stoker Corporation, Worcester, Mass. Mem. A.S.M.E.

Limitation of fuel-burning capacity in the case of fuel burned on stokers is a function of two dimensions. In the case of powdered-coal firing the limitation is a function of three dimensions. For this reason the limit of coal burning with powdered-coal firing will exceed the limit with stoker firing. Powdered-coal firing can take advantage of any furnace area that can be used with stoker firing. In addition, increasing furnace height permits an increase in coal-burning capacity with powdered-coal firing approximately proportioned to increase in height. With stoker firing, coal-burning capacity is increased only slightly with increased furnace height and that only with certain limits.

This writer will then make another guess, and probably less of a guess than in the first case, that powdered-coal firing will extend fuel-burning capacities beyond anything possible with stokers and will develop to meet demand for increased capacities for a considerable period of time in the future.

A. R. MUMFORD.[5] Table 1 of the paper indicates that if a plant of given capacity is designed for twelve units instead of four, the cost of boilers, superheaters, settings, etc., will increase approximately 70 per cent. It is assumed that the large units differ from the small units only in the amount of heating surface per unit inasmuch as the boilers of the four-unit plant contained 60 sections each, whereas the boilers of the twelve-unit plant contained 20 sections each. Under such conditions the cost of this item, although the principal factor in the increase in investment costs as shown, is not as overshadowing as it might be in a plant which altered the unit design itself as the capacity was increased. The reduction in boiler cost can be said to be entirely in the drum cost—namely, the substitution of one drum for three. If it is assumed that the setting cost is roughly proportional to the perimeter of the unit, then a 50 per cent reduction in setting cost is obtained by substituting one large unit for three small units. The saving in setting cost undoubtedly constitutes a large part of the total saving in the boiler, superheating, setting, etc., item in Table 1. It seems that a boiler unit designed for high capacities should be especially designed and not necessarily be a small boiler grown up, and, further, such changes in design of boiler will permit more capital economy in the use of surface combinations.

A recent study of preliminary proposals from boiler manufacturers indicated that about 75 per cent of the cost of a boiler alone was represented by drums in a bent-tube boiler, and presumably by drums and headers in a straight-tube boiler. Any attempt to reduce the cost of a boiler by designing it for higher steam generation is usually met by an increase in drum sizes which, because of their proportion of the total cost, causes the boiler cost of a high-rating unit to be nearly the same as that of a low-rating unit. In other words, the drum sizes would be proportioned to the steam output without a definite relation to the amount of heating surface. The general reason for the relation of drum sizes to rates of generation seems to be that a certain area of water surface is required for a unit of generation. It seems that the area of the drum at the water level is usually taken as the steam-releasing surface. Experiments made during the moisture investigation at Kips Bay Station cast grave doubts on the existence of any definite water surface in high-capacity boilers. Further, if it is assumed that the maximum rate of generation at which dry steam was produced with no auxiliary equipment indicated the proper drum size, then the drums ought to have been about four times the diameter they are if releasing surface was to control. The boilers were made to produce dry steam at four or five times the original dry point by the use of auxiliary moisture equipment and at comparatively low cost. From the standpoint of moisture in steam the writer believes that the cost of large units can be reduced by substituting effective auxiliary separating equipment for large drums.

Small units operating at low rates of generation usually have a large volume of water below the water level, so that a cessation of water supply is not as dangerous as it is in large units operating at high rates of generation. The drum sizes of the large units are often increased to increase the water reserve and to steady the water level. Chemical concentration studies of a large high-rating unit indicate that approximately 50 per cent of the volume below the water level is filled with steam. If the supply of heat to the boiler surface is suddenly and drastically reduced when the unit has only 50 per cent of its volume filled with water, it is apparent that a change of water level will take place which will be of such a magnitude that reserve water capacity in practical drum sizes will be insufficient to maintain a visible water level. The writer believes that this problem can best be answered by increasing the feed-pump capacity to the extent which will result in a minimum drop in water pressure when a control valve opens wide. If the pressure is permitted to drop too far, the other units on the line will be adversely affected. The increased investment in pumping machinery will offset to some extent the saving resulting from the removal of water storage from the drums, but it is probable that a balance can be reached which will result in a net decrease in investment.

If a boiler is designed so that all of its surface will be working as hard as does the first pass of existing boilers, and if the indications and their interpretations, which the writer has found in one case, are found to apply generally, then large units will be even more economical than the author has shown them to be. The investment in economizers, superheaters, and air heaters is more nearly proportional to surface than that of boilers, in the writer's experience, so the assignment of duty to each type of surface can be readily made for a given set of load conditions and fuel cost. Until a real high-rating boiler is developed, the maximum reduction in investment possible as a result of increased unit capacity cannot be realized.

In properly assigning duty to the various types of heating surface, a consideration of changes in surface spacing will indicate the possibilities of investment reduction under the heading of draft systems. In one instance a new unit was designed to have a 75 per cent increase in steam capacity with only a 12 per cent increase in fan-drive power over another unit, by opening up the gas passages. The necessity for such balancing of heat absorption and draft loss is evident to any one who has faced the problem of materially increasing the capacity of a given unit.

In any consideration of hard driving the effect of the precipitation of dissolved solids in the boiler water becomes of great importance, as the author has indicated. Because of the greater temperature differences at high absorbing rates, scales which have not been dangerous in the past become dangerous, and unless the boiler water is properly conditioned many of the advantages of large units will be lost by the necessity of frequent outages for internal cleaning. The writer tends to overemphasize this phase of operation because most of his experience has been in plants using 90 per cent make-up. However, a 5 per cent make-up plant will, with as pure water as New York City drinking water, develop dangerous scales in from 36 to 54 weeks in high-rating boilers unless the water is carefully conditioned.

In general it is felt that, although large units are now economically justified under suitable loads, as the author has said, high-rating units will become even more economical from an investment standpoint as their uses increases and provides a breath of experience necessary for alteration to design.

H. B. LAWRENCE.[6] The author brings out some of the features

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[6] Engineer, H. L. Doherty & Co., New York, N. Y. Mem. A.S.M.E.

that have been instrumental in the demand for increasing the size of boiler units. This demand has gone hand in hand with the increasing production of kilowatt-hours by steam power, the decreasing amount of coal used per kilowatt-hour, the increasing size of steam-turbine units, the increase in steam pressure and temperature.

On such items with which all are familiar, there is no especial need of further emphasis or to draw attention to them at this time.

When one turns to the matter of investment, time of construction, and period of getting into service, however, one finds more difficulty in obtaining for discussion figures that are representative; and while there has been an improvement fully worth while on the matter of investment, so that the overall operating maintenance and fixed charges show a correct tendency, there remains a need to give the matter of cost further study toward the end of more reduction.

The operating ratios of representative groups of utilities have certainly not kept pace with the tremendous gains in economy, whether they are gains in production or distribution. Increases in fuel prices, wages, and rate reductions are responsible in a large measure for the rather limited betterment of the operating ratio.

A boiler plant containing four 350,000 lb. per hour boiler units can be put in for about $2,500,000 when the fuel is Illinois bituminous. An oil-burning plant of the same size would cost about $2,360,000, and a natural-gas plant would cost about $1,860,000. These figures are probably not correct, but they are relative and represent the boiler plant.

Obviously there need be nothing in a plant of such size for burning coal or oil or gas that would fail for lack of reliability or any other feature, and the writer believes that it can be shown, as the author aptly points out, that if we decrease the number of boilers, the cost of the boiler plant will come down.

However, if one satisfactorily answers the questions of an engineering and thermal nature, such, for instance, as the effect of size on reliability, simplicity, spare or reserve capacity, ease of operation, time of construction, auxiliary power, there remains a number of matters of as great or possibly greater importance to consider in the question of size, such as:

1 To forecast as closely as possible the growth and rate of growth of the plant.

2 To have available some ideas as to obsolescence, whether from physical limitations or from advancement of the art.

3 To know the effect on manufacturing cost.

4 To study commodity prices, wage levels, fuel prices, and cost of money.

A casual glance at curves of cost of money, bond yields, and commodity prices indicates that a plant built in the period, say, from 1917 to 1921 would suffer from relatively high bond yields and commodity prices.

Over a long period the general business curves can be divided into sections representing cycles, and if one is to construct plants that embody good engineering in the full sense of the word, he would have to inject into the question something more than the physical characteristics. Such a study will often determine size.

After all, size per se does not give the thing that is wanted, and the writer feels that all situations must be studied in the light of the conditions that exist and are likely to exist, paying especial attention to the economic, geographic, and social factors and governing the question of capacity and increments of capacity by all the means at command.

Author's Closure

Mr. Crane in his discussion emphasizes the futility of expressing cost comparisons on a horsepower basis. This is timely and true. Horsepower and per cent of rating are obsolete terms so far as modern practice is concerned.

Mr. Craig's prophecy as to methods that will be adopted to increase outputs is interesting. A few forced circulation boilers are being used abroad, but to the author's knowledge nothing is being done commercially in this country.

It is recognized that the types of boiler to be used for different capacities will not necessarily be the same, as is pointed out by Mr. Mumford. The object of the paper was to bring out certain relations between number and size of boilers and their effect on costs. To have assumed a different type of boiler for each case would have introduced another variable, and the value of the comparison would have been lessened correspondingly.

FSP-52-32

Flow Characteristics of Some Lead Cable Sheath at Temperatures Above Atmospheric

By C. L. CLARK[1] and C. UPTHEGROVE,[2] ANN ARBOR, MICH.

This paper gives results of an investigation on three types of lead cable sheath differing only in the antimony content, which was, respectively, 0.06, 0.6, and 1.2 per cent.

At room temperature all three compositions elongated at more or less fixed rates with loads as low as 245 lb. per sq. in. The results would indicate that a similar behavior would be obtained with still lower loads and that the behavior of lead at 70 deg. fahr. should be similar to that of plain carbon steel at a temperature above its equicohesive range.

Increasing antimony content does not result in a decrease in the rate of flow at the lower temperatures, but in a lowered resistance to creep.

At a temperature of 80 to 120 deg. cent. the higher antimony content results in a decrease in the rate of creep, while the lower antimony results in a comparatively high rate of creep.

ALTHOUGH lead is a material which finds wide commercial application, relatively little has been published concerning its physical properties. Especially is this true of its flow characteristics, or its strength, at temperatures slightly above normal. Contradictory statements may be found as to the elastic-limit values and the conditions under which lead will flow under a constant load. Claims have been made that commercial lead will sustain indefinitely, in tension, loads well over the 200 lb. per sq. in. elastic limit as reported by Shoji. As power lead cable sheathing is subjected to varying pressures and to temperatures which may be well above normal, it appeared desirable to investigate its flow characteristics.

In addition to this practical reason for desiring to know the behavior of lead at elevated temperatures, there is also one of theoretical interest. At the present time considerable attention is being given to the behavior of steel at elevated temperatures. It has been claimed by some that the critical temperature in this work is the lowest temperature of recrystallization or the equicohesive temperature. This temperature has been reported to be about 800 deg. fahr. for plain carbon steel, and experiments have shown fairly conclusively that at any temperatures above this, for example, at 1000 deg. fahr., steel behaves very differently than it does at 600 deg. fahr. The equicohesive temperature for lead has been reported as below 70 deg. fahr., and accordingly lead at room temperature should behave similarly to plain carbon steel when at temperatures above its equicohesive range. If such is the case, lead would be expected to flow under any given load, no matter how small, at room temperature which may be assumed to be approximately 70 deg. fahr. This should not be considered to mean, however, that lead cannot safely be used at room or slightly elevated temperatures, as the rate of flow with the smaller loads is sufficiently low so as not to interfere with its commercial usefulness.

Material

Three types of conductor paper-and-lead cables of 350,000 circular units were furnished by The Detroit Edison Company as the source of lead for this investigation. The three types differed from each other only as to the antimony content. The cable sheathing was used in preference to regular sheet lead in that it was desired to retain any possible effects that might be due to the method of fabricating.

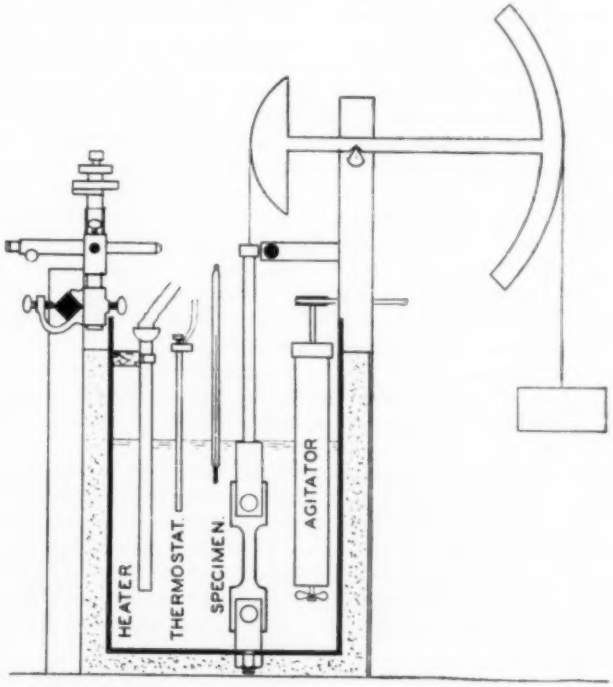


Fig. 1 Apparatus for Flow Tests on Lead

The antimony contents of the three types selected were:

Type of cable sheath	A	B	C
Antimony content, per cent	0.6	1.2	0.06

Arsenic and bismuth contents were also determined, but were found to show no appreciable variation. The three samples of cable sheath may be considered as representative of the compositions generally encountered in this field.

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[2] Associate Professor, Department of Metallurgical Engineering, University of Michigan.

Contributed by the Joint Research Committee on the Effect of Temperature on the Properties of Metals and presented at the Semi-Annual Meeting, Detroit, Mich., June 9 to 12, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

Procedure

The temperatures selected for the creep tests were 25 deg. cent. (77 deg. fahr.), 40 deg. cent. (104 deg. fahr.), 60 deg. cent. (140 deg. fahr.), 80 deg. cent. (176 deg. fahr.), 100 deg. cent. (212 deg. fahr.), and 120 deg. cent. (248 deg. fahr.). Loads of 245 and 368 lb. per sq. in. were used. These loads correspond to pressures within the cables of 45 and 75 lb. per sq. in. and may be assumed to approach the extremes of the pressures normally encountered in commercial practice. The loads were maintained for a period of 360 hours (15 days), unless failure occurred.

Transverse specimens were used for all temperatures and load conditions rather than longitudinal specimens in that the investigation was primarily concerned not with a longitudinal stretching of the cable sheath but rather with a possible expansion in diameter due to internal pressure.

In order that a comparison between the flow of longitudinal and transverse specimens might be made, longitudinal specimens of all three compositions were run at 80 deg. cent. Longitudinal specimens of the 0.06 per cent antimony lead were also run at 25, 40, and 100 deg. cent.

Creep Apparatus

The apparatus used in the creep tests is shown in Fig. 1. The load is applied by means of a lever system, the arms of which are in a ratio of two to one. Arcs whose radii are in the ratio of two to one are fastened to each end of the lever, and the load and the specimen are in turn fastened to the arcs. The specimen is suspended in an oil tank between two holders or adapters which are of a saddle-like arrangement and which permit the specimen to adjust itself to the applied load. The lower holder or adapter is fastened to the bottom of the tank, while the upper adapter is connected by a wire to the arc on the short arm of the lever. The arrangement of the lever and attached arcs permits the load to be applied in a constant direction and also keeps the ratio of the lever arms constant regardless of the position of the lever. The apparatus is constructed to permit the simultaneous testing of four specimens.

Flat specimens, which were approximately $5^1/_2$ in. long and 0.125 in. in thickness were used. The reduced section or gage length was 0.5 in. wide as compared to $1^1/_2$ in. in the unreduced sections.

An oil bath of the proper viscosity to be fluid at the lower temperatures and which did not fume at the upper temperatures was used as the heating medium. Two knife-blade immersion heaters were used for heating the bath, and uniformity of temperature was maintained by circulation of the oil with a motor-driven pump. The temperature was maintained within a fraction of a degree centigrade by a Central Scientific Company thermostat.

The elongation or flow of the specimen was determined by means of a micrometer telescope by measuring the relative change in position of a mark or line on the upper specimen holder and a fixed point of reference on the apparatus. The micrometer telescope which permitted reading accurately to 0.01 mm. or 0.00039 in. was mounted so that it could be readily moved from one specimen to another.

It is recognized that the method of measuring the elongation does not record only that occurring in the gage section, but that occurring over the entire test specimen. In spite of the enlarged sections at each end of the specimen and the method of supporting, some flow necessarily occurred outside of the gage sections. As the purpose of the investigation was, however, primarily to determine the comparative flow characteristics, it was not considered necessary to attempt to limit the elongation measured to the 2-in. gage length.

Short-Time Tensile Tests

Preliminary to the creep tests, short-time tensile tests were made on the three samples of lead cable sheath at room temperatures. These tests were made for the purpose of determining the effect of the antimony content on the breaking strength and also to obtain a comparison of the transverse and longitudinal properties of the three materials. The results are given in Table 1.

TABLE 1 SHORT-TIME TENSILE TESTS ON THREE CABLE SHEATHING LEADS

Specimen number	Composition	Direction of specimen	Breaking stress, lb. per sq. in.	Elongation, per cent in 2 in.
CT	0.06 Sb	Transverse	2013	29.2
CL	0.06 Sb	Longitudinal	2158	33.0
AT	0.60 Sb	Transverse	2000	19.8
AL	0.60 Sb	Longitudinal	2118	33.0
BT	1.20 Sb	Transverse	3234	29.0
BL	1.20 Sb	Longitudinal	3096	28.2

For the particular samples of lead cable sheath the antimony up to 0.6 per cent was found to have no effect on the breaking strength. Increasing the antimony to 1.2 per cent resulted in a very appreciable increase in strength. Differences between longitudinal and transverse breaking strengths are relatively small, the order being reversed in the case of the higher antimony content.

Long-Time Creep Tests

Long-time creep tests were conducted on transverse specimens of the 0.6 and 1.2 per cent antimony leads at 25, 40, 60, 80, 100 and 120 deg. cent. (77, 104, 140, 176, 212, and 248 deg. fahr.) and on longitudinal specimens of these two leads at 80 deg. cent. (176 deg. fahr.). Similar tests were conducted on both transverse and longitudinal specimens of the 0.06 per cent antimony lead at 25, 40, 80, and 100 deg. cent. (77, 104, 176, and 212 deg. fahr.). Two tests were conducted on each type of lead specimen at each temperature, one with a load of 245 lb. per sq. in. and the other with a load of 368 lb. In the performance of these tests, the specimen was brought to heat and the load applied after sufficient time had been allowed to permit a state of equilibrium to be reached. Elongation readings were taken immediately before and after the application of the load. The results obtained from the transverse specimens of the three leads are shown in Figs. 2 through 7, while those from the transverse and longitudinal specimens at 80 deg. cent. (176 deg. fahr.) are compared in Figs. 8 and 9.

As seen from these curves, the tests in each case were continued for 360 hours, unless rupture was obtained before that length of time had elapsed. Elongation readings were taken at intervals and plotted against time. The slope of these lines therefore represents the rate of elongation of the specimen under the given load.

The slope, or rate of elongation, of each curve has been computed and the results are given in Table 2 and Fig. 10. The rate of elongation is expressed in inches per inch per hour. An examination of the values in either the table or figure will best reveal the behavior of these three leads under the conditions to which they were subjected.

At either 100 or 120 deg. cent. (212 or 248 deg. fahr.) the particular specimens of the 1.2 per cent antimony lead cable sheathing examined are seen to offer the greatest resistance to creep of the three leads tested. The remaining two leads are of about the same strength at 100 deg. cent. in so far as the small loading is concerned, but on the basis of the larger load the 0.06 per cent antimony lead cable sheathing flows at the more rapid rate of flow. At 80, 60, 40, and 25 deg. cent. (176, 140, 104, and 77 deg. fahr.) the 1.2 per cent antimony lead elongates at a more rapid rate than does the 0.6 per cent anti-

mony lead. The 0.06 per cent antimony lead was not tested at all of these temperatures. At 80 deg., on the basis of the larger load, it is the weakest of the three leads studied, while at 40 and 25 deg., it offers about the same resistance to deformation under fixed load as does the 0.6 per cent antimony lead and consequently is stronger than the 1.2 per cent.

In regard to creep tests on longitudinal and transverse specimens at 80 deg. cent. (176 deg. fahr.) shown in Figs. 7 and 8, the results can again best be compared by considering the numerical values of the rates of creep. These, together with

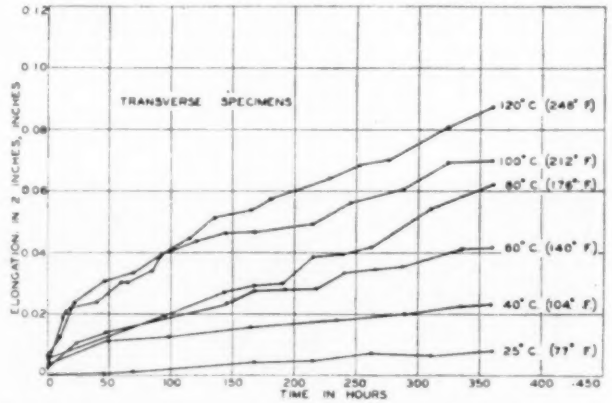


FIG. 2 FLOW TESTS ON 1.2 PER CENT SB LEAD WITH CONSTANT LOAD OF 245 LB. PER SQ. IN. AND AT INDICATED TEMPERATURES

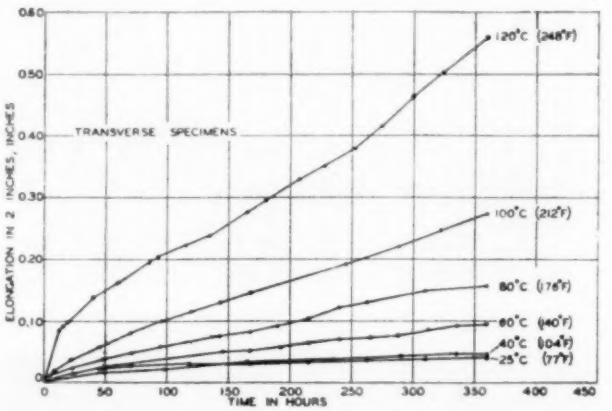


FIG. 3 (SAME AS FIG. 2 EXCEPT 368 LB.)

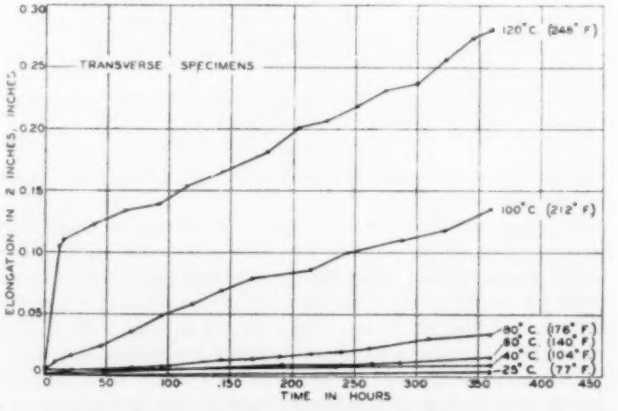


FIG. 4 FLOW TESTS ON 0.6 PER CENT SB LEAD WITH CONSTANT LOAD OF 245 LB. PER SQ. IN. AND AT INDICATED TEMPERATURES

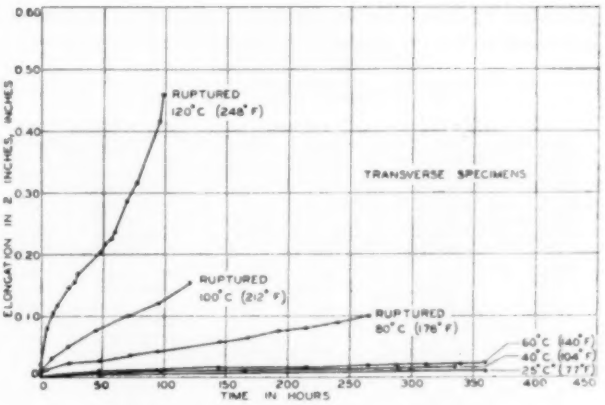


FIG. 5 (SAME AS FIG. 4 EXCEPT 368 LB.)

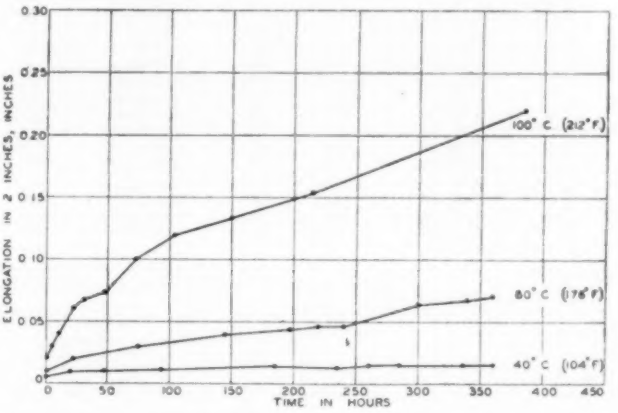


FIG. 6 FLOW TESTS ON 0.06 PER CENT SB LEAD WITH CONSTANT LOAD OF 245 LB. PER SQ. IN. AND AT INDICATED TEMPERATURES

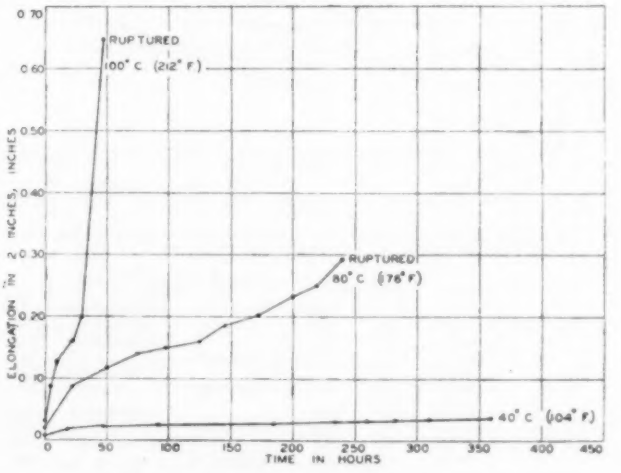


FIG. 7 (SAME AS FIG. 6 EXCEPT 368 LB.)

corresponding values obtained with 0.06 antimony lead at 25, 40, and 100 deg. cent., are given in Table 3.

The results show no consistently superior values to be given by either type of specimen. At 40 and 100 deg. cent. the longitudinal specimens of the particular specimens of the 0.06 per cent antimony lead cable sheathing elongated at a faster rate than did the corresponding transverse ones, while at 25 and 80 deg. cent. the opposite was true. Corresponding tests conducted on

the other two leads at 80 deg. cent. showed the longitudinal specimens to be the stronger in all cases except one.

TABLE 2 VARIATIONS IN RATE OF FLOW OF LEAD WITH COMPOSITIONS AND TEMPERATURE

Designation	Type of lead, per cent	Temperature, Deg. cent.	Deg. fahr.	Rate of flow, inches per inch per hr.	Stress in specimen per sq. in.
A 45	0.6 Sb	25	77	0.0000038	245
A 75	0.6 Sb	25	77	0.0000063	368
B 45	1.2 Sb	25	77	0.0000095	245
B 75	1.2 Sb	25	77	0.0000250	368
C 45	0.06 Sb	25	77	0.0000057	245
C 75	0.06 Sb	25	77	0.00000673	368
A 45	0.6 Sb	40	104	0.0000065	245
A 75	0.6 Sb	40	104	0.0000190	368
B 45	1.2 Sb	40	104	0.0000185	245
B 75	1.2 Sb	40	104	0.0000530	368
C 45	0.06 Sb	40	104	0.0000070	245
C 75	0.06 Sb	40	104	0.0000170	368
A 45	0.6 Sb	60	140	0.0000229	245
A 75	0.6 Sb	60	140	0.0000372	368
B 45	1.2 Sb	60	140	0.0000453	245
B 75	1.2 Sb	60	140	0.0001072	368
A 45	0.6 Sb	80	176	0.0000500	245
A 75	0.6 Sb	80	176	0.0001663	368
B 45	1.2 Sb	80	176	0.0000864	245
B 75	1.2 Sb	80	176	0.0002146	368
C 45	0.06 Sb	80	176	0.0000700	245
C 75	0.06 Sb	80	176	0.0004050	368
A 45	0.6 Sb	100	212	0.0001630	245
A 75	0.6 Sb	100	212	0.0005000	368
B 45	1.2 Sb	100	212	0.0000950	245
B 75	1.2 Sb	100	212	0.0003150	368
C 45	0.06 Sb	100	212	0.0001550	245
C 75	0.06 Sb	100	212	0.0040500	368
A 45	0.6 Sb	120	248	0.0002235	245
A 75	0.6 Sb	120	248	0.0012046	368
B 45	1.2 Sb	120	248	0.0001004	245
B 75	1.2 Sb	120	248	0.0005697	368

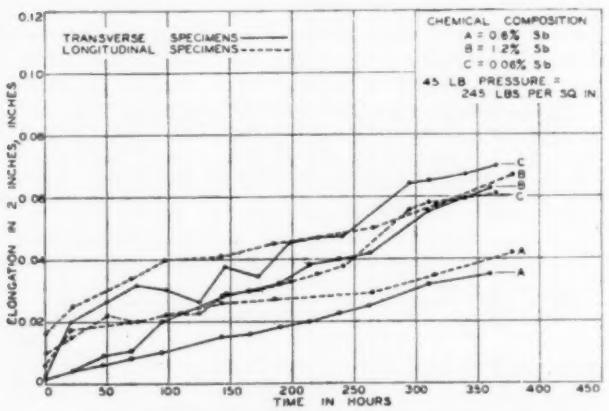


FIG. 8 FLOW TESTS ON LEAD AT 80 DEG. CENT. (176 DEG. FAHR.)

TABLE 3 EFFECT OF FABRICATION UPON THE RATE OF FLOW OF LEAD AT VARIOUS TEMPERATURES

Designation	Type of lead, per cent	Temperature Deg. cent.	Deg. fahr.	Rate of flow, inches per inch per hour Transverse specimens	Longitudinal specimens	Load, lb. per sq. in.
CL 45	0.06 Sb	25	77	.....	0.00000235	245
CT 45	0.06 Sb	25	77	0.00000579	.....	245
CL 75	0.06 Sb	25	77	.....	0.00000428	368
CT 75	0.06 Sb	25	77	0.00000673	.....	368
CL 45	0.06 Sb	40	104	.....	0.0000105	245
CT 45	0.06 Sb	40	104	0.0000070	.....	245
CL 75	0.06 Sb	40	104	.....	0.0000250	368
CT 75	0.06 Sb	40	104	0.0000170	.....	368
CL 45	0.06 Sb	80	176	.....	0.0000650	245
CT 45	0.06 Sb	80	176	0.0000700	.....	245
CL 75	0.06 Sb	80	176	.....	0.0003550	368
CT 75	0.06 Sb	80	176	0.0004050	.....	368
AL 45	0.60 Sb	80	176	.....	0.0000360	245
AT 45	0.60 Sb	80	176	0.0000500	.....	245
AL 75	0.60 Sb	80	176	.....	0.0001200	368
AT 75	0.60 Sb	80	176	0.0001663	.....	368
BL 45	1.20 Sb	80	176	.....	0.0000560	245
BT 45	1.20 Sb	80	176	0.0000864	.....	245
BL 75	1.20 Sb	80	176	.....	0.0002660	368
BT 75	1.20 Sb	80	176	0.0002146	.....	368
CL 45	0.06 Sb	100	212	.....	0.000185	245
CT 45	0.06 Sb	100	212	0.000155	.....	245
CL 75	0.06 Sb	100	212	.....	0.025000	368
CT 75	0.06 Sb	100	212	0.004050	.....	368

CONCLUSIONS

The results obtained allow certain conclusions to be drawn regarding the effect of antimony content, temperature, and magnitude of stress upon the creep characteristics of certain specimens of lead cable sheath. Because of the extreme instability of the lead-antimony alloys, care should be used in extending these conclusions to other alloys of similar composition

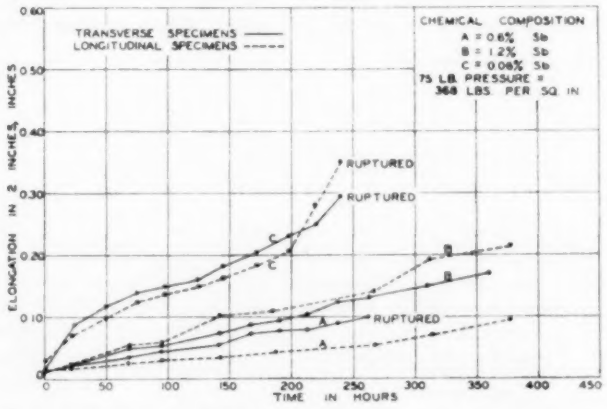


FIG. 9 FLOW TESTS ON LEAD AT 80 DEG. CENT. (176 DEG. FAHR.)

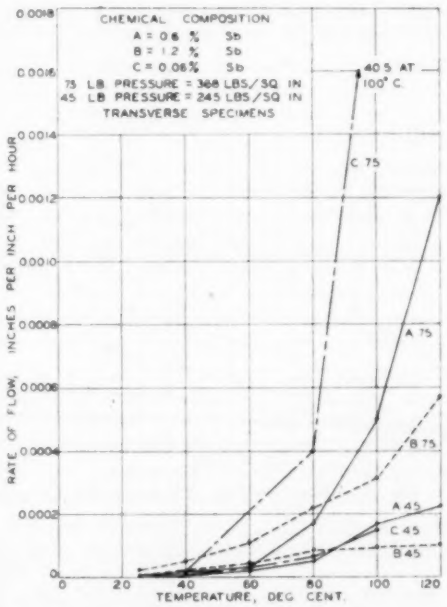


FIG. 10 VARIATION IN RATE OF CREEP OF LEAD WITH COMPOSITION AND TEMPERATURE

or to alloys of the same composition but which have been fabricated differently.

At room temperature all three compositions of lead cable sheath elongated at more or less fixed rates with loads as low as 245 lb. per sq. in. The results would indicate that a similar behavior would be obtained with still lower loads and that the behavior of lead at 70 deg. fahr. should be similar to that of plain carbon steel at a temperature above its equicohesive range.

Increasing antimony content does not result in a decrease in the rate of flow at the lower temperatures, but results in a lowered resistance to creep.

At a temperature of 80 to 120 deg. cent., the higher antimony

content results in a decrease in the rate of creep, while the lower antimony results in a comparatively high rate of creep.

Exceptions may be found to these findings at intermediate temperatures, compositions, or loads due to several reasons, chief of which are variations in grain size of the lead, the mutual solubility of the lead and antimony, and the amount, size, and distribution of the excess antimony particles. No explanation is offered at this time.

The authors wish to express their appreciation to The Detroit Edison Company for the financial aid given to this work, and to C. F. Hirshfeld, Chief of Research for The Detroit Edison Company, for his kindly interest and advice.

Discussion

R. L. Peek, Jr.[3] As an approximation to the type of analysis suggested by Dr. Nadai it should be possible, if determinations of creep are made under several loads at each temperature, to compute from the elongation the approximate average cross-section at any instant, and hence, approximately, the instantaneous average stress intensity. From such data there could then be obtained the rate of creep as a function of elongation for a constant stress intensity. When, as in some of the results given in this paper, the rate of creep for a constant load (increasing stress intensity) is constant through a considerable range of elongation, it would seem that the rate for a constant stress intensity must be decreasing and perhaps approaching zero for some limiting value of the elongation, i.e., that here there is also some strain hardening, or some similar effect.

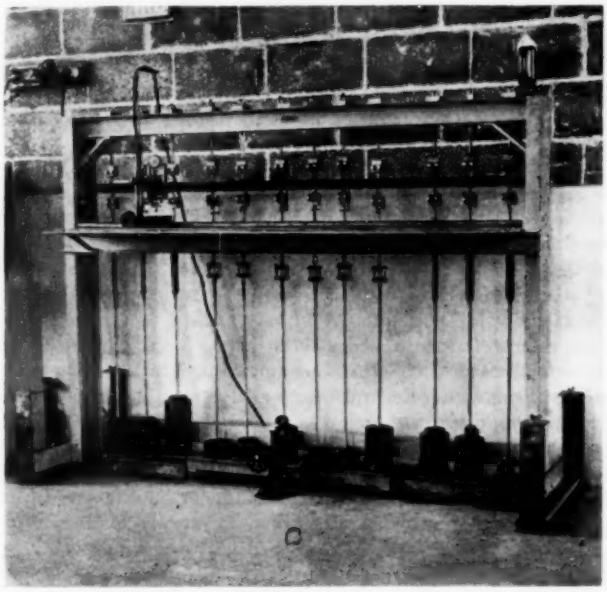


Fig. 11 Test Racks Arranged for Room-Temperature Tests

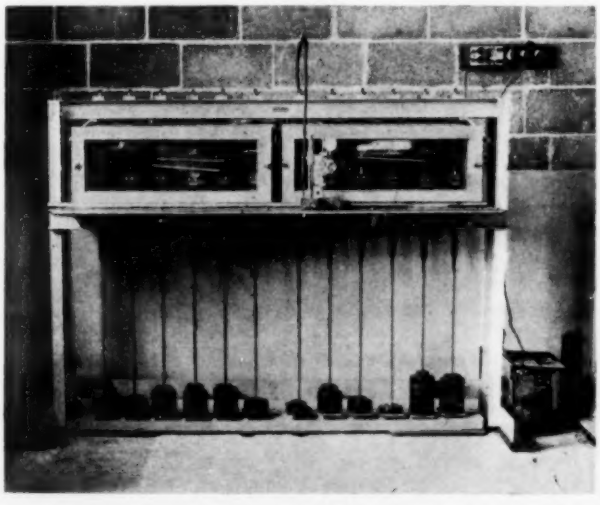


Fig. 12 Test Racks With Specimens Enclosed in a Heat-Insulating Chamber

[3] Bell Telephone Laboratories, New York, N. Y.

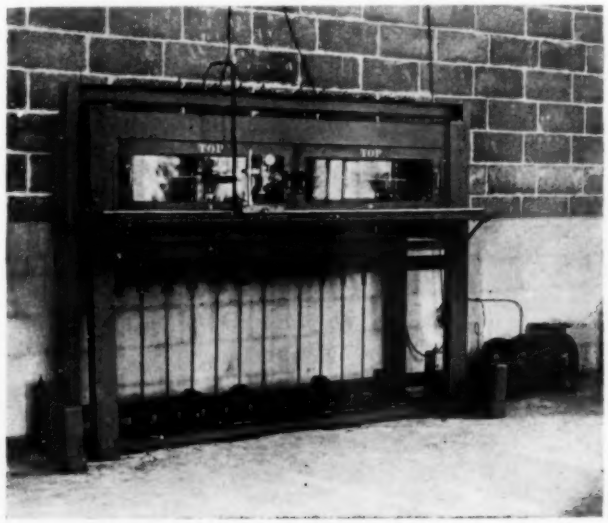


Fig. 13 Test Racks With Air-Cooling Equipment

H. F. Moore.[4] This paper presents interesting and valuable results and is of especial interest in emphasizing the importance of flow at room temperatures of certain of the softer metals. A series of flow tests of lead cable sheathing is in progress at the University of Illinois under the auspices of the Utilities Research Commission, and it is thought that some comment on these tests may be of interest in connection with this paper.

The tests at Illinois are being made on unalloyed lead, lead alloyed with antimony, and lead alloyed with tin. At present most of the tests have been on unalloyed lead. The general arrangement of the test racks are shown in Figs. 11, 12, and 13. Lead specimens with a minimum section 1/4 in. wide are subjected to the direct load of dead weights. The elongation of the specimen is measured by means of a microscope with micrometer-controlled travel. To each specimen are clamped two pieces

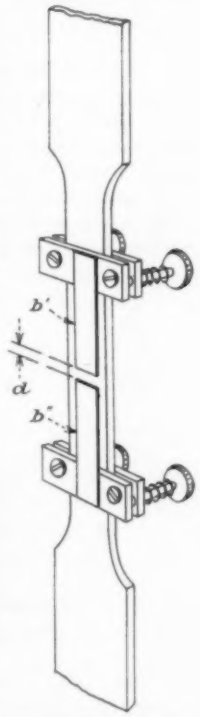


Fig. 14 Methods of Clamping Brass Strips to Specimens

[4] Research Professor of Engineering Materials, University of Illinois, Urbana, Illinois.

of brass (b' and b'') as shown in Fig. 14, and the distance between the edges of the brass pieces (b in Fig. 14) is measured by the traveling microscope shown in Fig. 13, and in somewhat greater detail in Fig. 15. Tests are run at (1) room temperature, (2) at a temperature approximately 32 deg. fahr., and (3) at a temperature approximately 150 deg. fahr. Fig. 11 shows the rig for room temperature tests. The six specimens in the center of this test rack are arranged so that they may be readily loaded and unloaded, and are tested in cycles of eight hours under load followed by 16 hours with load released.

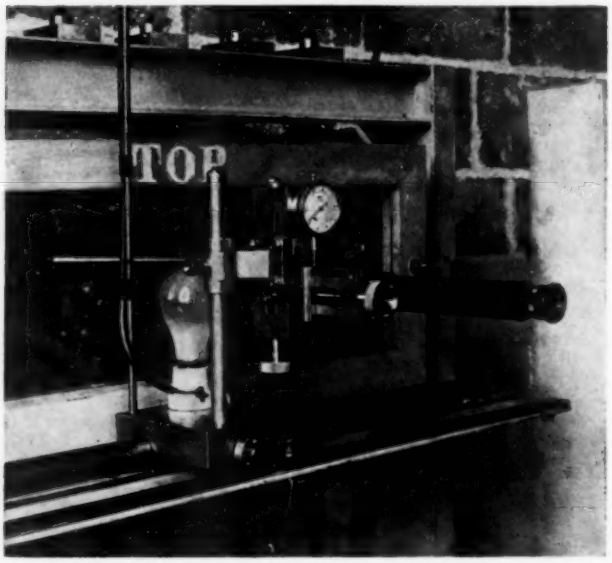


FIG. 15 TRAVELING MICROSCOPE FOR MAKING MEASUREMENTS

Fig. 12 shows the rig for 150 deg. fahr. The specimens are enclosed in a heat-insulating chamber, which is heated by a small electric heater, and the temperature is controlled by a thermostat circulation. A good degree of uniformity of temperature is secured by baffles. Fig. 13 shows the rig at 32 deg. fahr. The air is cooled by means of a Frigidaire unit. All three test racks are suspended from springs so adjusted that the racks rest lightly on sponge-rubber pads. This insures a minimum of vibration in the specimens.

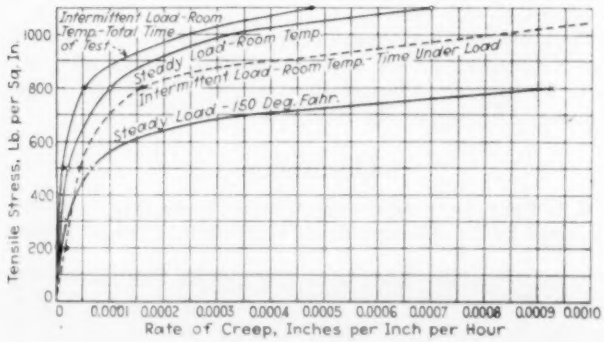


FIG. 16 TEST RESULTS WITH AN UNALLOYED-LEAD TRANSVERSE SPECIMEN—INTERMITTENT LOAD, EIGHT HOURS ON AND SIXTEEN HOURS OFF

Fig. 16 shows some results which have been obtained with unalloyed lead. Test data for tests at 32 deg. fahr. are not yet available. In general these tests are in agreement with those of the authors, but the unalloyed lead seems to have a somewhat greater rate of flow than does the antimony-lead alloys reported by the authors. Attention is called to the relatively high rate of flow at 150 deg. fahr. as compared with that at room temperature, and also to a comparison of rate of flow under steady load with rate under intermittent loading. As might be expected, if the rate of flow is measured by the entire elapsed time of test it is somewhat less for intermittent loading than for steady load. However, if the rate of flow is measured by the time the specimens were actually under load it was to be greater for intermittent loading than for steady load.

FSP-52-33

The Heat-Balance Test as a Means of Obtaining Evaporative Efficiencies of Steam-Generating Equipment

By B. J. CROSS,[1] NEW YORK, N. Y.

The efficiency of any energy-converting apparatus is determined by a division of the energy output by the energy input. In the case of boiler and furnace this is attained by weighing the coal or other fuel and weighing the water evaporated. The paper attempts to show that the efficiency as determined by the heat-balance method should have equal value with an efficiency as obtained by the test of weighing the coal and water.

THE fundamental operation for the determination of the efficiency of any energy-converting apparatus is the division of the energy output by the energy input. This is the direct method and should always be employed when these two quantities can be accurately determined. For the determination of the efficiency of a boiler and furnace, this method entails the weighing of the coal or other fuel delivered to the furnace and the weighing of the water evaporated in the boiler. The steam represents the operator's energy requirement and the coal represents the price he must pay for it. It is quite logical therefore to seek out these fundamental quantities.

There is often, however, some difficulty in the direct weighing of the coal and water. Water, especially, is often difficult to weigh under the conditions that exist in the boiler room. In small plants where the water is fed cold to the boiler or economizer or to closed heaters it is usually not a difficult matter to install tanks ahead of the boiler-feed pumps and weigh or measure all of the water fed to the boiler. In many cases, however, the weighing of water is very difficult and often practically impossible. The practice of feedwater heating by stage bleeding of turbines has introduced many difficulties. The water cannot be handled in open tanks at temperatures approaching the boiling point. The water as it comes from the condensers does not represent the total amount fed to the boilers, as a considerable quantity is bypassed through the heaters.

The greatest difficulty is in the quantities to be handled. Boilers that evaporate 300,000 to 500,000 lb. of water an hour are now fairly common, and a few are operating or have been projected to operate at capacities up to 800,000 lb. per hour. Testing facilities have not kept pace with the development in size and capacities of boilers, and it is improbable that any attempt will be made to weigh water to boilers at these rates. The expense of making complete evaporative tests on such large units would be very great and would usually be considered prohibitive.

The weighing of the coal used does not present so great a problem as does the water. In the first place its amount is only about one-tenth of that of the water. Even when it is to be used in powdered form, it can be weighed without difficulty, although special tanks or bins must be used. The greatest possibility for error is in judging the quantity of coal on the grates or in the pulverized-coal bins at the start and end of the test. This error is reduced in per cent by making the test of long duration. However, if it is impractical or impossible to weigh the water, it is of little avail to weigh the coal. Whether the water can be weighed is usually the deciding question as to the possibility of making the evaporative test.

If it is not possible to weigh the water evaporated, the only alternative for the purpose of determining the efficiency of a steam-generating unit is the heat-balance method. This method consists of the measurements of losses and subtracting their sum from 100. It would more accurately be called the method of losses. It is the purpose of this paper to present a discussion of this method.

The Heat Balance

The heat balance is a part of all complete evaporative tests where fuel and water are weighed. The heat absorbed as determined by the weight of water evaporated and its increase in heat content plus the sum of all the losses should equal 100 per cent. While all of the losses cannot be measured, those that can be measured can be determined quite accurately, while those that cannot be readily measured are small and are of the order of less than 0.1 per cent. The item "unaccounted-for losses" in the heat balance, therefore, consists almost entirely of errors. These errors are commonly considered to lie in the measured losses. The following is a discussion of these losses and of the degree of precision with which they can be measured. The losses that can be measured are as follows:

1 The loss due to the sensible heat above room temperature in the dry flue gases.

2 The loss due to steam in the flue gases. This steam comes from three sources: from evaporation of the moisture in the fuel, from the water formed when the hydrogen of the fuel combines with oxygen, and from the water vapor present in the air used for combustion.

3 The loss due to incomplete combustion of carbon. This loss is evidenced by the presence of carbon monoxide in the flue gases.

4 The loss due to unburned solid carbon. This loss is represented by the carbon or combustible in the refuse from the ashpit and various settling chambers and the carbon in the refuse that is carried up the stack with the gases.

5 The loss due to the heat radiated from the surfaces of the unit. This loss cannot be directly measured, but can be estimated and a figure set as a high limit.

Fig. 1 shows a heat-flow diagram of a steam-generating unit. The quantities are all to scale. This diagram is to show the relative magnitude of the various quantities. The individual losses are small compared to the heat absorbed, and a comparatively large error in their determination will have a proportionally small effect on the efficiency.

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[1] Combustion Engineering Corporation, New York, N. Y.

Presented at a meeting, New York, N. Y., April 11, 1929, of the Metropolitan Section, The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The Dry-Gas Loss

The loss due to heat in the dry flue gases is the largest item of the heat balance. It is also the item that is susceptible to the largest error. The greatest precautions should, therefore, be used in its determination.

The formula for determining the weight of dry flue gases per pound of coal is usually written

$$\left[\frac{11\ CO_2 + 8\ O_2 + 7\ N_2 + 7\ CO}{3\ (CO_2 + CO)}\right] \times C_b$$

in which the various gases are expressed in per cent by volume and C_b is the carbon burned per pound of coal.

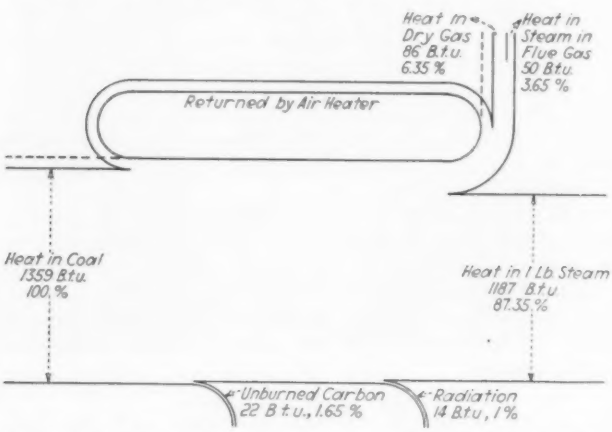


Fig. 1 Heat-Flow Diagram of a Steam-Generating Unit

This may be changed to

$$\left[\frac{4\ CO_2 + O_2 + 7\ (CO_2 + O_2 + N_2 + CO)}{3\ (CO_2 + CO)}\right] \times C_b$$

But as $(CO_2 + O_2 + N_2 + CO)$ constitute 100 per cent, the formula in its simplest form becomes

$$\left[\frac{4\ CO_2 + O_2 + 700}{3\ (CO_2 + CO)}\right] \times C_b$$

The entire expression for the heat lost in the dry gases is

$$\left[\frac{4\ CO_2 + O_2 + 700}{3\ (CO_2 + CO)}\right] \times C_b \times \left[0.24\ (t_2 - t_1)\right]$$

The items that are measured are CO_2, O_2, and CO in per cent by volume, t_2 the flue-gas temperature, and t_1 the air temperature. C_b, the carbon burned, is obtained by subtracting the unburned carbon per pound of coal from the carbon as reported by analysis.

Flue-Gas Analysis

The most important measurements in the determination of the dry-gas loss are the CO_2 and the flue-gas temperature. The oxygen and CO are small quantities, and considerable error in their determination causes only a small error in the result. C_b, the carbon burned, will vary only slightly with an error in the unconsumed carbon. It is improbable that any considerable error will be made in room air temperature.

The possible error in measuring CO_2 is influenced almost entirely by the location of the sampler. Once the sample is in the Orsat, its analysis can be made with an accuracy of plus or minus 0.2 per cent. The difficulty lies in obtaining a true sample. A great many devices have been tried in the attempt to obtain a single sample that is representative of the composition of the gas. Probably the best way is to use simple open-end pipes, taking as many samples in as many locations as the resources at hand permit. In wide settings as many as six samplers distributed across the setting may be necessary. A preliminary traverse of the breeching may be made and the variation in composition of the gases determined. The greater the variation the larger will be the number of sampling points necessary. In modern units where often both economizers and air heaters are employed, the gases are apt to be rather thoroughly mixed by the time they leave the system and the sampling thereby be simplified. Air leakage between the various units increases the difficulty of sampling and should be kept at a minimum.

The error in the per cent of CO_2 due to sampling and analysis should with proper care not exceed 0.5 per cent.

Temperature Measurement

As in the sampling and analysis of gases, the possible error in the determination of temperature of the flue gases is due almost entirely to the location of the instrument used. The solution similarly is to use as many points as possible. A temperature traverse may be made and the number of points to be used and their locations be thus determined. The use of thermocouples in this measurement permits a large number of temperature measurements to be made in a section of the breeching with a minimum of effort and expense. In a breeching, for example, of 5 ft. by 15 ft., five couples may be located at 3-ft. intervals. Each couple may then be moved to five positions. There can

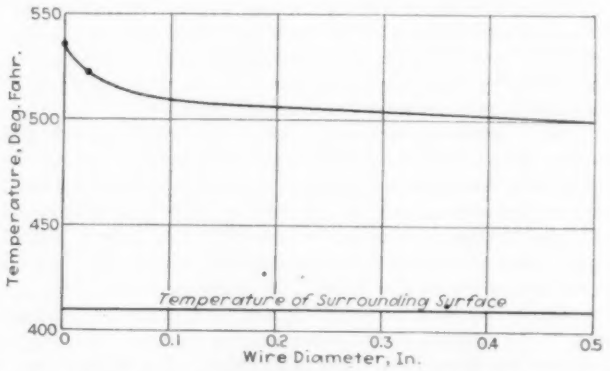


Fig. 2 Effect of Thermocouple-Wire Diameter on Radiation Error

thus be made 25 readings at as many points in the section. If a little care has been taken in wiring the couples to a convenient switch, these 25 measurements can be made in not more than five minutes. A potentiometer should preferably be used in measurements, as there is then no error due to the resistance of the leads.

The thermocouple used for measurement of gas temperature should be made of wire as small in diameter as possible consistent with durability. Theoretically, in a stream of hot gas inclosed by walls at a lower temperature than the gas, the junction of the couple can never reach the gas temperature. It is receiving heat from the gases and radiating heat to the walls of the duct. The temperature it indicates is therefore an equilibrium temperature and lies somewhere between the gas temperature and the wall temperature. The smaller the diameter of the wire the closer will be the indicated temperature to the true temperature.

Fig. 2 shows the effect of wire diameter on the indicated temperature. This chart is taken from Bulletin 145 of the U. S. Bureau of Mines and is based on experimental data by Kreisinger and Barkley. In practice No. 22 B. & S. gage wire, which is 0.025 in. in diameter, has been found to be well suited. It is

large enough to be fairly durable and is not too large to cause a serious error due to radiation. In gas ducts, the walls of which are insulated and where the gas velocity is fairly high, the error due to radiation should not be large. In boiler settings where the gas velocities are low and the couple can "see" the comparatively cooler boiler surfaces the radiation error may be considerable.

The junction of a thermocouple is best made by welding. Copper constantan couples can be welded in an ordinary bunsen burner or even in the flame of a small gasoline blow torch. Iron-constantan couples require a higher temperature, and an oxygen-gas flame or an electric arc must be used. The wires should be butt welded, the junction being but little if any larger in diameter than the wire. Fig. 3 shows some examples of good and bad thermocouple junctions. The hot junction of a couple is shown at *a*. The wire of the couple should project well beyond its protection tube. The error due to the conduction of heat along the wire is thus minimized. A good butt weld is shown at *b*. A couple with a bead is shown at *c*. The wire has been held in the flame too long, and the surface tension of the molten metal has drawn the melted metal from the ends of the wires, causing the thin sections near the head. Such a weld is fragile, and in addition there is apt to be a large error due to radiation. At *d* is shown a couple with too short an exposure to the gas. The outside supporting pipe and the insulation are always at a lower temperature than the wire of the couple, and heat is conducted from the wire. All that can be said in favor of *e* and *f* is that they are easy to make. The greatest advantage in the use of a thermocouple in measuring flue-gas temperatures lies in the small mass of the active part of the instrument. In examples *e* and *f* this advantage is sacrificed.

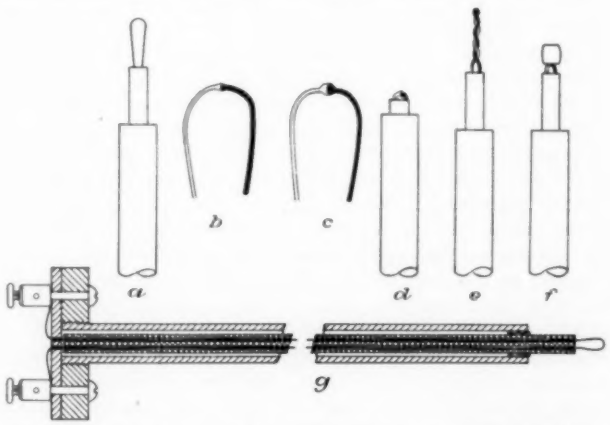


FIG. 3 EXAMPLES OF GOOD AND BAD THERMOCOUPLE WELDS

A completed couple is shown at *g*. It consists of a $^1/_4$-in. iron pipe which supports the wire in a suitable insulation. This insulation may consist of porcelain or glass tubing. The head holding the binding posts may be made of fiber or wood. The cost of such a couple should not exceed $1, labor included, and the wire can be renewed at the cost of a few cents.

A good portable potentiometer should be part of the equipment of the boiler room. Thermocouples can replace mercury thermometers in most temperature measurements. While the first cost of the potentiometer is higher than that of the thermometers, the upkeep is apt to be considerably lower. Bulletin 145 of the U. S. Bureau of Mines is recommended as a text on the use of thermocouples.

The calibration error of a couple over the range of temperatures found in flue gases should not exceed 2 or 3 deg. The largest chance for error is due to the location of the couple and the radiation of heat to cooler surrounding surfaces. In ducts, the walls of which are well insulated, the temperature of the walls will not be far below that of the gases. The error due to radiation should not exceed 10 deg. fahr. The combined error due to radiation and location should not exceed 15 deg. fahr.

Fig. 4 shows the loss due to heat in the dry flue gases at various temperatures plotted against CO_2. Thus at 15 per cent CO_2 and 500 deg. temperature, an error of 0.5 per cent in CO_2 and 15 deg. in temperature will cause an error of 85 B.t.u. or 6 per cent of the total quantity of heat lost. But this represents an error of only 0.5 per cent in the efficiency. At a temperature of 300 deg. and 15 per cent CO_2, the same errors in measurement will cause an error of 65 B.t.u., or 9 per cent of the total loss. But again this represents an error of only 0.55 per cent in the efficiency.

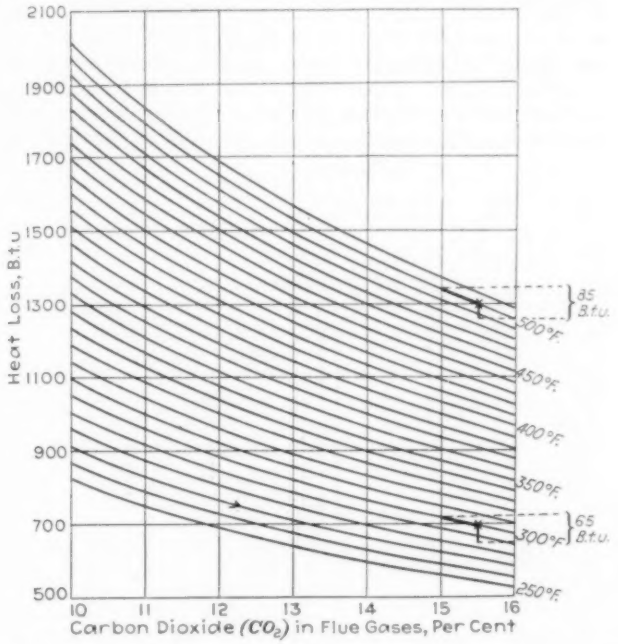


FIG. 4 VARIATION OF DRY-GAS LOSS WITH CO_2 AND TEMPERATURE

STEAM IN FLUE GASES

The loss second in importance in size is that due to moisture in the flue gases. This moisture comes from three sources: from the moisture in the fuel, from the water vapor present in the air used for combustion, and from the moisture formed by the combustion of hydrogen in the fuel.

Where the moisture W_m must first be evaporated, as is the case with the moisture in the fuel and that formed by burning hydrogen, the loss is equal to $W_m = [1050 + 0.48\ (t_2 - t_1)]$. As the latent heat of evaporation is large in comparison with the second term, the loss is almost constant. It would take a considerable error in t_2, the flue-gas temperature, to appreciably affect the final result. An error of 15 deg. in the temperature affects the efficiency about 0.25 per cent.

The equation for the heat in steam in flue gases is sometimes written

$$\text{loss} = W_m\ [(212 - t) + 970.4 + 0.48\ (t_2 - 212)]$$

This implies that the moisture is evaporated at a temperature of 212 and superheated to the temperature of the stack gases. Actually the water is evaporated not at atmospheric pressure but at its own partial pressure. A more nearly correct expression, which also has the advantage of being simpler is

$$\text{loss} = W_m\,[1050 + 0.48\,(t_2 - t_1)]$$

The actual difference in results by the two methods is, however, small.

As the loss due to moisture in the air used for combustion is very small, usually less than 0.1 per cent, any error in its determination need not be considered.

Loss Due to Unburned Carbon

The loss due to unburned carbon is difficult to measure. If this loss occurred entirely in the ashpit, its determination, while apt to be rather laborious, would be comparatively simple. A large part of the unburned carbon, however, passes out of the stack with the gases. This is true both of stokers and of powdered-coal-fired furnaces. The ashpit loss can be determined by the weight of the refuse and its analysis. The loss of carbon in the

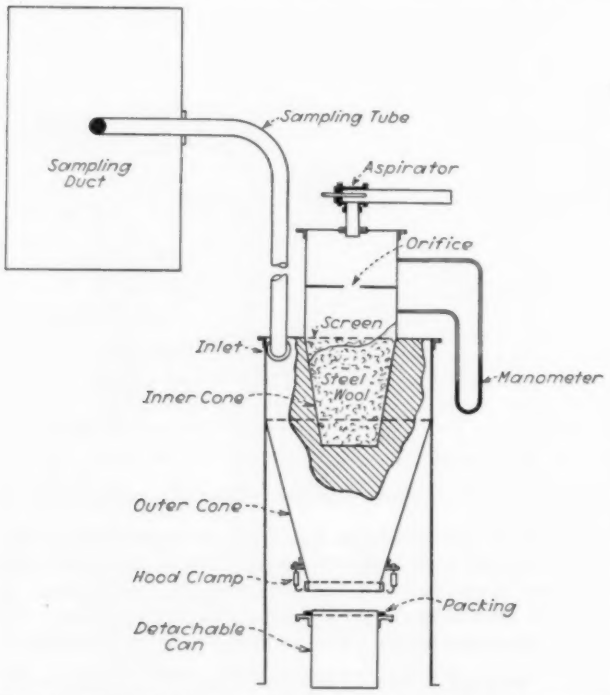


Fig. 5 Arrangement of Collector for Determination of Dust in Flue Gases

stack gases can be determined with a fair degree of accuracy by aspirating a known weight of gases through a collector and filter. Such a device is shown in Fig. 5. The purpose of the flow measurement by means of the orifice is to obtain the correct velocity at the sampler intake and also to obtain a quantitative measure of the total amount of refuse carried up the stack. The fraction sampler area over total area of the duct is the proportion of the total gases that must be drawn through the apparatus. The assumption is made that the velocity at the point of sampling is the average velocity in the duct. The error in quantity of refuse so determined may be as great as 10 per cent. The error in quality (i.e., carbon content) is due chiefly to the loss of fine particles and is usually on the high side. The combined error of sampling in quantity of refuse and quality of the sample obtained should not exceed 20 per cent. If the total loss due to unburned solid carbon is 1.5 per cent, the maximum error in the efficiency would then be 0.3 per cent.

The error in the chemical analysis of the refuse is probably on the high side. The method for determining the combustible consists of burning the sample in a muffle furnace and calling the loss in weight carbon, which is assumed to have a heat value of 14,600 B.t.u. per lb. Actual calorimetric determinations of the combustible have shown a heat value averaging about 13,800 B.t.u. per lb. The combustible as determined by the loss in weight on burning is apparently not pure carbon.

The Radiation Loss

The loss due to radiation of heat from the boiler setting cannot be directly measured. The rate of radiation from such surfaces has, however, been the subject of much study and is of the order of about 250 B.t.u. per sq. ft. per hour. Boiler and furnace walls are usually well insulated, and there are few parts of a setting where the hand cannot be comfortably held.

As in most plants boilers are set in battery or adjacent to each other, the radiation amounts in part to a mutual exchange and the loss is reduced. The rate of 250 B.t.u. per sq. ft. per hour can be safely said to be a maximum figure. The surface exposed need not be accurately measured but only approximated.

The actual loss due to radiation is practically constant regardless of rating. It can be estimated for normal rating of the boiler, and the loss at any rating may then be computed by dividing by the per cent rating. The per cent loss due to radiation will thus vary considerably with rating. At moderate ratings it will be in the neighborhood of 1 per cent. The error in computing this loss will not exceed 25 per cent of the loss and would cause an error of 0.25 per cent in efficiency.

Sampling and Analysis of Fuel

The error in the heat value of the fuel may result from two causes: the sampling of the coal at the plant and the analysis of the sample in the laboratory. The error of sampling, even when the most careful methods are used, may be considerable. If the coal is moist, as is often the case, the time consumed in sampling must be kept at a minimum in order that the loss of moisture may be kept low. On the other hand, time is required to reduce the sample in size and to mix it thoroughly so that the samples taken may be representative. The best way out of this dilemma is to take separate samples for the moisture content and for the heat-value determination. Investigations are now in progress to determine the accuracy with which coal may be sampled, the preliminary results of which indicate that an error of 150 to 200 B.t.u. may be expected. The error of analysis of coal is smaller. A chemist is expected to check his own determination within 40 to 50 B.t.u. Different laboratories should check within 75 B.t.u. The combined error due to sampling and analysis may easily be 200 B.t.u.

For the purpose of a heat-balance test an error of 200 B.t.u. in the heat value of the fuel is, fortunately, not serious. The equation for efficiency by the heat balance method is

$$\frac{\text{B.t.u. of coal—losses}}{\text{B.t.u. of coal}}$$

As the term "B.t.u. of coal" appears in both the numerator and the denominator, the error is largely compensated.

Loss Due to Incomplete Combustion of Carbon to CO

In a comparison of heat balances of tests made 10 or 15 years ago with those made more recently, one of the noticeable differences is in the CO reported in the flue gases. Where it was once common to report percentages of CO from 0.5 to 1, in the more recent tests this item is usually zero. This difference is due to the increase in size of combustion chambers and the use of air over the fuel bed. As the amount of CO is small—usually it is not measurable—and as the same per cent of error in its determination applies as for CO_2 determination, the error in the determination of this loss can safely be neglected.

Unaccounted-For Losses

If radiation has been accounted for by the method previously described, the unaccounted-for losses should consist only of the undetermined items of the gas analysis and sensible heat in refuse. The undetermined constituents of the stable gases consist of unburned hydrogen and hydrocarbons. The loss due to these constituents can almost invariably be neglected because they usually exist only in traces.

The amount of heat lost due to the sensible heat in the ash and refuse is usually very small, although if ashes are rejected from the furnace in molten state the loss may be appreciable. However, in this special case the loss may readily be calculated.

The possible errors of the items of the heat balance as discussed in the preceding paragraphs are to be considered as the maximum errors when all reasonable precautions have been taken to obtain accurate measurements of the factors involved. These errors are for the greater part accidental rather than constant. They may be plus or minus in direction and just as apt to be compensating as additive. Considered under the most unfavorable circumstances where they will be additive, the total error in the sum of the losses will be the sum of the errors of the individual losses. The total maximum error in obtaining the efficiency of a boiler and furnace by the heat-balance method may be determined therefore by summing up the individual errors. Thus the total error of a carefully made heat-balance efficiency test will be as follows:

1 Error in dry-gas loss, per cent	0.5
2 Error in loss from steam in flue gases, per cent	0.1
3 Error in loss in incomplete combustion, per cent	0.4
4 Error in radiation loss, per cent	0.2
5 Error due to sampling and analysis of fuel, per cent	0.2
Total error in efficiency, per cent	1.4

If there is added 0.1 per cent for the unmeasured losses, the total possible error in efficiency by this method is 1.5 per cent.

To those who like to report the results of a boiler test to the second decimal place, the admission of a possible error of 1.5 per cent in a heat-balance test would seem to condemn such a test. However, if one investigates the possibility of errors in a weighed-coal-and-water test, he will find that, even though all measurements are made with great care, the probable error of such a test will not be greatly different from that of the heat-balance test. Unlike the efficiency as determined from the heat balance, the efficiency as determined from a weighed-coal-and-water test is greatly affected by an error in the heat value of the fuel. An error of 150 B.t.u. in the heat value will cause an error in efficiency of over 1 per cent.

The heat absorbed per pound of steam evaporated is determined by the temperature of the feedwater and the temperature and pressure of superheated steam. The temperature of the feedwater can be determined to within 2 deg. fahr. and that of the superheated steam to within 5 deg. fahr. The combined error of these two temperatures is 0.4 per cent. Even neglecting the possibility of error in the weighing of coal and water, it is seen that the error of a weighed-coal-and-water test may easily be 1.5 per cent.

A comparison between the efficiency as obtained from weighed water and coal and the efficiency as determined by heat balance is shown in Figs. 6 and 7. These show graphically the results of a series of complete tests on a stoker-fired boiler and a pulverized-coal-fired boiler. In these charts the efficiency by weighed water and coal is plotted upward from the zero ordinate; the losses are plotted downward from the 100 per cent line. The amount by which the sum of the total losses does not meet the efficiency as obtained by the evaporative tests or the amount by which they overlap represents what usually is called the unaccounted-for losses, but which should more correctly be called the error of the tests.

It will be seen that the maximum error in these diagrams is 3 per cent. As it has been shown that the maximum probable error of both the heat-balance test and weighed-water-and-coal test is in the neighborhood of 1.5 per cent, a combined error of 3 per cent is possible.

It has not been the purpose of this paper to depreciate the weighed-coal-and-water test. Tests should be made by this method whenever possible. Such tests should also be checked

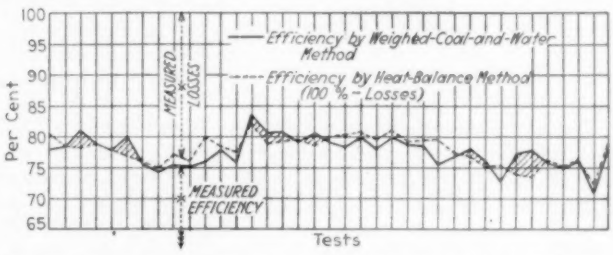


Fig. 6 Results of 36 Complete Weighed-Coal-and-Water Tests of Stoker-Fired Boiler, With Heat Balances to Check

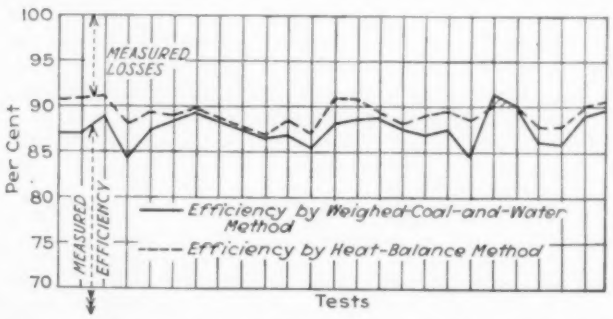


Fig. 7 Results of 25 Complete Weighed-Coal-and-Water Tests of Pulverized-Coal-Fired Boiler, With Heat Balances to Check

by means of a heat balance whenever possible. This paper has attempted to show that the efficiency as determined by the heat-balance method (or by the method of losses, as it would more accurately be called) should have equal value with an efficiency as obtained by a weighed-coal-and-water test.

Discussion

Vincent M. Frost.[2] The writer is much interested in the presentation of this subject as his company has made use of a combination of the standard weighed-coal-and-water method and the heat-balance method for several years. Our plants are equipped to weigh both coal and water for testing of the largest boilers. In weighing the water, however, it is necessary to do so at a temperature low enough to prevent flashing into steam, and this results in making the boiler tests with water considerably colder than normally supplied.

For many of our operating tests, therefore, we have weighed the coal, used the boiler-steam-flow meter to determine the boiler output and approximate rating, and after computing the losses, from the data of flue-gas analysis, flue-gas temperature, ultimate analysis and heat value of the coal, loss in the refuse, and making a definite allowance for radiation and unaccounted-for loss, determined the "probable efficiency," as we have termed it, by difference.

[2] Assistant to the General Superintendent of Generation, Public Service Electric and Gas Company, Newark, N. J. Mem. A.S.M.E.

Attention has been called to the possible error in the coal weights. In making tests of large stoker-fired boilers, we have frequently made the test of 96 hours in order to reduce the possible error due to the condition of the fuel bed to a negligible value.

The loss due to the carbon in the refuse is not an item to be dismissed lightly when one is attempting to chase down an elusive efficiency and determine whether the combustion is as efficient as might be or has been claimed.

With a stoker equipped with a clinker grinder and deep pit burning 20,000 to 30,000 lb. of coal per hour, not only is there a very large amount of refuse to be handled and sampled, but the refuse from a given test does not begin to be rejected from the furnace in from 6 to 12 hours after it entered the clinker pit.

The writer foresees one great advantage from the general use of the proposed scheme, in that a definite value for radiation and unaccounted-for will of necessity be used, and there should no longer be tests reported with high efficiencies and negative radiation and unaccounted-for losses in order to balance.

The results of any boiler test, no matter how made, which shows by a heat balance negative "radiation and unaccounted-for" should be taken with a "grain of salt."

It is believed that the radiation loss in many instances will be very much larger than "1 per cent at moderate ratings" as suggested by the author. Its actual value will be very much influenced by the type of setting of both boiler and furnace and the size of the furnace as compared to the size of the boiler.

On the eleventh page the equation for the efficiency by the heat-balance method is given as

$$\frac{\text{B.t.u. of coal} - \text{losses}}{\text{B.t.u. of coal}}$$

with the statement that, due to the fact that the B.t.u. of the coal appears in both the numerator and denominator, the error is largely compensated.

If the equation is rewritten into the form

$$1 - \frac{\text{Losses}}{\text{B.t.u. of coal}}$$

it will be seen that the value used of B.t.u. of the coal is a direct factor and the value of the error is from 0.25 to 0.35 per cent, depending upon the relative heat value of the coal as between, say, 12,000 and 14,000 B.t.u.

In the paper reference is made to the efficiency as determined from a weighed-coal-and-water test, and it is stated that an error of 150 B.t.u. in the heat value will cause an error in efficiency of over 1 per cent. This should be qualified to the extent of stating the heat value of the coal to which the 150 B.t.u. correction is to be made.

The efficiency as determined by a weighed-coal-and-water test may be expressed in a simplified form as follows:

$$\frac{\text{Lb. of steam per hour per lb. of coal} \times \text{B.t.u. per lb. of steam}}{\text{B.t.u. per lb. of fuel}}$$

This fraction has the same form as that given for the per cent losses as given above, and the B.t.u. per pound of coal thus appears as a direct factor. The largest error involved in the coal measurement of a test of a stoker-fired boiler is in determining the relative amount of fuel in the furnace at the end of the test as compared to that at the beginning.

A similar situation exists in the determination of the amount of pulverized fuel in the weighing bins at the beginning and end of the test, particularly if they are determined on a volumetric base.

A. A. Markson.[3] The paper is valuable to the profession, not only because of its subject, but because it is a most timely contribution. The writer has had some experience in boiler testing in which the purpose of the testing has always been to obtain data of value to the operation rather than in the mere compilation of academic figures. It is only natural, then, for him to have used the methods the author suggests, for they constitute a very powerful analytical weapon in which any skill of the engineer is repaid by the economy of securing the data. This is always a controlling factor in the work of a power-plant research department. In short, the operating returns obtained from any test are in the last analysis only valuable to the extent of the skill and ability employed in making the test, drawing the correct conclusions, and, finally, seeing that the conclusions are carried into execution.

The author makes a good point in the last paragraph in that his paper should really be called "Evaporative Efficiencies by the Method of Losses." The writer does not think this statement can be made too strong.

To illustrate: A series of tests were made on a Coxe stoker-fired boiler and economizer about five years ago at one of our stations. These tests were run at ratings of 200 to 375 per cent, and there were nine in the series of 12 to 24 hours' duration. The following items were obtained in order to make an error-proof heat balance:

1 Coal was weighed.
2 Water was weighed.
3 Cinders caught by cinder-vane fan were weighed.
4 All ashes were weighed.
5 Gas analyses and temperatures were obtained and checked with extraordinary care.
6 In addition the usual data called for in the paper were gathered.

It was rather surprising, then, to find that the real heat balance showed "unaccounted-for" 3.7 to 14.1 per cent, going up with the rating.

This was finally traced to an unexpectedly high cinder loss, the existence of which would never have been suspected if only the readily ascertainable losses had been determined.

On the other hand, large units have been under continuous testing at Kips Bay Station substantially in accordance with the author's outline. The "unaccounted-for" was estimated at 2 per cent. Recently five weighed tests were run, which lasted from 48 to 72 hours each. The "unaccounted-for" was between 2.8 and 2.2 per cent on all tests, which showed that the estimate of 2 per cent previously used was not a bad one.

It therefore seems that the method of losses is peculiarly adapted to powdered-coal furnaces and is not reliable to a sufficient extent on tests of stokers at high ratings and is likely to be misleading owing to the task of ascertaining the carbon losses. A few matters of technique occur in the paper which the writer would like to discuss.

The weighing of water is not necessary, even when facilities are at hand. In the writer's organization there are at least four test engineers who can meter water or steam in any quantity to better than 2 per cent. The steam and water meter in skilful hands is just as reliable as weighed water, although the precision is slightly less. What these men can do any engineer should be able to do if he will take the time to become acquainted with this Society's report on fluid meters.

One can easily picture an engineer in the case of the high "unaccounted-for" tests just mentioned if the output had been determined by means of a flow meter. He would blame the whole

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thing on the flow meter. This is wrong, and the continued persistence of steam engineers in refusing to rely on the meters is certainly a reflection on their own technique. Hydraulic engineers get good results without weighing half of Niagara Falls to a turbine. Why not steam engineers? If commercial recorders are not trusted, how about using manometers? If orifices and venturi tubes are unknown quantities, how about checking them?

In conclusion, it is the writer's opinion that in many cases the study of the often glossed over "unaccounted-for" is the way to more effective boiler testing. Deducting radiation losses which can be estimated closely, a theoretically good boiler-test series should have positive and negative "unaccounted-for." Using the maximum possible error is "safe" but not informative. The distribution of accidental errors in boiler testing follows the law of probability as it does in all measurements. If a test or a series of tests, when the data have been closely analyzed, shows a positive unaccounted-for beyond the precision of the data, this should be closely looked into because it represents a systematic rather than an accidental error and not infrequently pictures the possible room for improvement of efficiency.

HENRY KREISINGER.[4] The heat-balance method of making boiler tests is less costly than tests made by weighing water and coal. However, that does not mean that the heat-balance method of making a boiler test requires less skill and experience than the weighed-coal-and-water test. In order to be able to make an accurate heat-balance test one must be thoroughly familiar with the operation of the plant. He must know where the losses are before he attempts to measure them. If one is not familiar with the plant and its operation, he should spend several days at the plant watching the operation and studying all possible means in which losses may occur. He may even find it necessary to make some preliminary measurements in order to determine what apparatus and instruments he should use in measuring the losses during the test.

The author has described rather thoroughly the method of measuring the dry-gas losses, the losses due to water vapor in the products of combustion, and has outlined the method of estimating the radiation losses. He has also described the method of determining the incomplete combustion losses due to combustible in flue dust and in the refuse from the ashpit. These are the principal losses in the process of making steam. Of these losses the one due to combustible in the flue dust and cinders carried out the stack in the stoker-fired plants is usually given too little attention. A great amount of care is given to the sampling of ashpit refuse and the determination of its carbon content. The cinders going out the stack are usually passed by with an inaccurate estimate or very superficial measurement. And yet the loss due to combustible in the cinders may amount to several per cent, and if neglected may introduce serious error in the test. This is especially true when the boiler is fired with stokers of the forced-draft type. Therefore, an engineer who wishes to make a fairly accurate heat-balance test should give this one loss great attention and study the methods by which he can determine these losses accurately.

H. W. BROOKS.[5] Like Mr. Cross and Mr. Kreisinger, the writer has been brought up in that school of engineering thought which feels that a boiler test without a heat balance is not a boiler test. If one will read carefully the power test codes, he will find that the Society is substantially on record in the same belief. Generally speaking, therefore, both the weighed-water test and the heat-balance test are essential. Where conditions necessitate the sacrifice of one to the other, the thorough and experienced boiler tester will usually prefer to sacrifice the former.

Much remains, however, to be done in the perfection of technique in the heat-balance test. The writer is somewhat inclined to take issue with the author as to the rather narrow limitations of maximum error which he has established on the various items. As examples, only two specific cases may be mentioned—namely, "loss of unburned combustible" and "radiation and unaccounted-for." When dealing with the small all-cold Scotch marine boilers in the early days of the pulverized-fuel tests at the Philadelphia Navy Yards, when the writer first was learning to understand the then known limits of heat release per cubic foot of furnace volume in pulverized-coal furnaces, he used exactly the method outlined by the author in his own tests for solid unburned combustible. In spite of the fact that "radiation and unaccounted-for" ran in some cases as high as 21 per cent, the losses of unburned carbon as shown by the dust-sampling method rarely showed over 4 to 7 per cent. There was an obvious discrepancy here which was afterward checked up by the use of a rather complicated but quite accurate chemical balance computation suggested about three years ago by one of the professors of the University at Sheffield, England, and published by "Fuels in Science and Practice." This method was later worked out graphically by Mr. E. G. Bailey and presented before this Society. In this manner it was ascertained that the loss shown by the sampling method was less than 40 per cent of the actual loss. Later when the writer reduced the "radiation and unaccounted-for" to the eventual figure of around 6 to 8 per cent, he still found about the same ratio of discrepancy between the two methods. He has also found this discrepancy on many stationary boiler tests, proving that there is much yet to be learned in the sampling of pulverulent materials in suspension.

With regard to radiation loss, on the Philadelphia work prior to the powdered-coal tests a very accurate research had been made by the Navy for the specific purpose of determining radiation losses from the boiler. These tests had been made with oil firing and under specially careful and competent supervision. In them it was found that the method suggested of estimating radiating losses at 250 B.t.u. per sq. ft. lent itself to wide variations of inaccuracy, particularly at the lighter loads. Having had the same experience on stationary boiler tests where radiation was measured by the Nichols heat flowmeter (which accurately measures the radiation from masonry walls), the writer is inclined to believe that this is one of the most fruitful sources of error on light-load heat balances.

A negative "radiation and unaccounted-for" item is nearly always an evidence of lack of care and inaccuracy in the test. No boiler under fire can conceivably absorb heat from surrounding boiler-room atmospheric temperature. In other words no boiler can conceivably be a refrigerator in any possible manner. Heat balances, therefore, showing negative "radiation and unaccounted-for" can generally be regarded with considerable suspicion.

A. A. ADLER.[6] Boiler tests reported on the basis of water evaporated to fuel used will always make a strong appeal. This is due to the simplicity of the factors involved and the fact that it seems like a direct measurement. It may be true that the handling of water and coal, paralleling as nearly as possible operating conditions, will offer mechanical difficulties in an attempt at accurate measurement. The appeal becomes the stronger when the direct measurements are supported by a heat balance which includes gas analyses and flue-temperature readings.

Tests using gas analyses and flue temperature only are less

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[5] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[6] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

convincing because they depend on theoretical considerations not always clear to those for whom they are intended. There is always room for discussion as to where samples should be taken to be truly representative. Moreover, it makes a difference whether air leakage is through the fuel bed or if it takes place at other points.

The latter may be visualized from considerations of heat absorption in the furnace by radiation. Consider a furnace temperature of, say, 2800 deg. and a flue temperature of, say, 550 deg. under a certain operating condition. If a given quantity of air is admitted into the flue after the gas has passed over the heating surface of the boiler, the flue temperature may be dropped to, say, 500 deg. If a 50-deg. drop were to take place in the furnace, it would diminish the absorption all along the line on its way through the boiler. Consequently, from energy considerations the lesser absorption will be reflected in a higher flue temperature. The final temperature would be above 500 deg. In both cases the flue-gas reading would be substantially the same.

The paper is a valuable contribution in cases where comparable results are desired in the same or similar plants. Thus when one is seeking to improve plant economy, even though there be inaccuracies in the samples of flue gas and temperatures, the fact that all tests are made alike will tend to wipe out errors in measurement. The results will be comparable, although not accurate in the absolute sense. The comparatively low cost at which tests can be made in accordance with the scheme proposed by the author should find wide application.

B. G. Worth.[7] Two aspects of this paper impress one immediately. First, the close agreement of efficiencies determined simultaneously by both methods is quite remarkable, particularly when one considers the numerous factors involved. This suggests that the heat-balance method is perhaps more important as a means of checking the straight coal-and-water method than as an alternative method of easier application. And many will be hard to convince that the heat-balance method is, in fact, easier, even for larger plants.

The second aspect is the small variations in efficiency between the separate tests of each series. This variation, from recollection, was about 5 per cent, which appears to be a satisfactory result. If the coal-and-water method can come so close to the losses method, does not this rather negative the difficulties imputed to the coal-and-water method?

Tests are conducted generally for one or more of three reasons: to check apparatus-performance guarantees, plant-performance tests (often as basis for improvement programs), and for data on continuous operating cost.

In the case of performance tests the lines are often closely drawn and a few per cent, plus or minus, may make a great difference, depending on the liberality and practicality of the parties to the contract. In these cases absolute values are necessarily striven for.

With plant performance absolute values are not so important, for the usual use of such tests is to continuously improve the economy of the plant. If the results are inaccurate, but the inaccuracies are of the same relative nature, efforts toward improvement will be about as effective as if the results were absolutely true. In such cases it is a great deal better to have a relatively inaccurate test method which is actually used than a fine method which is seldom utilized because of its difficulties. It has been well said that "sustained mediocrity is preferable to intermittent perfection."

It is the writer's belief that there is a tendency to attach too much importance to the last per cent. No matter how carefully apparatus may be prepared and calibrated and samples of coal and ash taken, variations will occur, and sometimes they will cumulate and sometimes average out. This is shown most strikingly by the two sets of tests shown in the paper. A fair evaluation of the results would be a mean drawn through the evidently representative tests and throwing out the ones obviously wide of the mark.

If accuracy is considered a practical and not an academic matter, it will follow from inspection of the two sets of tests shown in the paper that either method of test has about equal chances of giving an acceptable result.

With this point conceded, the question of choice resolves itself into which method is the easier. This is a real problem in very large plants, and the difficulties of the coal-and-water method are, as the author stated, the reason for working out the losses method. But with smaller units, such as are generally found in industrial plants, such does not appear to be the case, and for such plants the writer is strongly of the opinion that the coal-and-water method is the easier, if not the better.

Just what plants the author had in mind is unknown, but it would seem that in practically all plants it is easier to measure coal than ash. Coal is carefully weighed in all plants, large or small, making any pretense of efficient management. Lack of care in this direction is certainly not imaginable in large public-utility plants, since coal is the major operating expense, and is moreover part of the ultimate overall basis of efficiency of the station, viz., coal per kilowatt-hour. Refinement in weighing coal should certainly not be difficult to attain in public-utility plants. With moderate-sized industrial plants it is actually not so easy, for many executives who will spend money freely on certified accountants and the like will balk at the cost of coal-and-water measuring equipment that would pay for itself in a year. There is, however, nothing difficult about coal weighing, and any well-designed plant should have the apparatus for such weighing built into the plant at the start.

Measuring and sampling ash is more difficult, and changes in composition are more likely than in coal. For ash weighing special apparatus must be provided in most cases, and is a dead loss, for it is not likely to be used in operative tests. When ashes are dumped into trucks, they may be weighed on truck scales with reasonable accuracy, but who is to say how much water was used to wet them before dumping into the truck. The same applies to dumping into railroad cars. But what about ashes which are disposed of by a water conveyor system. What about ash slagged off? To handle these seems a great deal more trouble than coal weighing. Again, changes in ash composition may take place very readily if the fireman is not vigilant. How is one to catch such changes? Of course, all ash inaccuracies are much less important than coal inaccuracies due to the fact that ash is only a small fraction of coal. Ash-measuring difficulties, however, appear formidable compared with the ease of measuring coal.

The real difficulty in the straight evaporative test is to measure the water. This can be easily done where feed temperature is low, but certainly is difficult when temperature is high. But with the possibility of measuring and checking steam output against feedwater and by various calibration methods determining the amount, character, and sign of instrument error, such determinations should be brought well within the reasonable error of the whole test. It is not easy to set up such accurate feedwater determination, for much patience and care are required to get flow meters to work dependably, but many engineers will agree that the effort and expense of such a proceeding should not be as great as that required for the heat-balance method.

Think of the difficulties in accurately measuring gas flow,

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gases composition, actual radiation, carbon loss up stack, etc.

Speaking from the standpoint of the moderate-sized industrial plant, it would seem to the writer that the heat-balance method is not justified in such causes.

In the cases of large plants where large sums of money, both in first cost and operation, are involved, the practical view would seem to be that both methods together are none too accurate, and that the author's loss method forms a very valuable check on the direct determination. It is a question whether it constitutes a good substitute.

JOHN BLIZARD.[8] The writer has read the paper with much interest. The author is one of the leading authorities on the subject and has devoted much time to developing means for measuring as accurately as possible the losses of energy from steam-generating equipment.

He gives a well-known and simple equation for determining the ratio of the mass of dry-flue gases to the carbon in that gas. This ratio may be represented in a still simpler form when the carbon monoxide is negligible.

This simple form is:

$$\frac{\text{Lb. dry gas}}{\text{Lb. carbon in gas}} = \frac{240}{CO_2} + 1$$

This equation is less accurate than the one given in the paper, but it will be found to be sufficiently accurate for almost all ordinary purposes.

Since the heat content of a wide range of steam coals is 18,000 B.t.u. per lb. of carbon, a very simple and useful expression may be derived from the foregoing equation to represent the heat content of the dry-flue gases per cent of the heat in the fuel burned. It is:

$$\left[\frac{0.32}{CO_2} + 0.0013\right][t_2 - t_1]$$

This will give somewhat too low a heat content for some of the low volatile coals, but its simplicity and reasonable accuracy warrant its frequent use.

The expression for representing the total heat of the steam in the flue gases may be expressed by:

$$1090 + 0.46\, t_2 - t_1$$

instead of by the expression given in the paper.

The value given here is the heat content of steam at 1 lb. absolute pressure, which is about the pressure of the steam in the flue gases.

AUTHOR'S CLOSURE

Mr. Frost has brought up several points. Unquestionably the sampling of ash from large stokers is a difficult problem. Where the stoker has a large ash chamber, the test should be prolonged or the stoker at least should be operated under test conditions long enough to insure a representative sample of ash. If the plant operator is willing to run a weighed-coal test for a period of 96 hours to reduce the error due to judging coal in the fuel bed, it should not be asking too much to have the heat-balance test prolonged to 12 hours to insure a representative sample of refuse.

The two expressions $\frac{\text{B.t.u. of coal} - \text{losses}}{\text{B.t.u. of coal}}$ and $1 - \frac{\text{Losses}}{\text{B.t.u. of coal}}$ are, of course, identical. The term "B.t.u. of coal" is just as much a direct factor in one as in the other. By substituting actual values and varying the B.t.u. of coal by 150, it will be seen that there will be an error of 0.25 and 0.21 per cent, respectively, for 12,000 and 14,000 B.t.u. coal. By using equivalent values in Mr. Frost's expression for the weighed-coal-and-water test efficiency, it will be seen that a 150 B.t.u. error in the heat value of the coal will cause an error of 1 per cent and 0.9 per cent, respectively, for 12,000 and 14,000 B.t.u. coal.

Mr. Markson's figures of 4 to 14 per cent "unaccounted-for" include an unmeasured loss of carbon carried up the stack with the flue gases. This item of the losses is generally overlooked in stoker tests. Even though the stack may appear clear, a considerable amount of cinder may be discharged from it. We often feel the "egg shells" under our feet when we walk around a stoker-fired plant. If this loss were as visible at the stack outlet as a similarly large loss in a pulverized-coal-fired boiler would be, it would receive more attention.

While this cinder loss is more difficult to determine than the flue-dust loss from a pulverized-coal-fired furnace, it is by no means an impossible task. The cinders are comparatively large particles and tend to segregate in the ducts. A vertical passage should be selected for the location of the sampling point or points. With ordinary care this cinder loss should be determined with the degree of accuracy claimed—namely, 25 per cent.

Mr. Brooks takes exception to the limits of error claimed for the various items of the losses. It has been assumed that painstaking care has been taken in the various measurements involved. The wide margin of cost between heat-balance tests and weighed-coal-and-water tests justifies this care. It is believed that with due precaution the measurements can be made within the degree of accuracy ascribed. The two series of tests plotted in Figs. 6 and 7 indicate such precision of measurement.

The specific losses mentioned by Mr. Brooks are the loss due to unburned combustible and the radiation loss. The accuracy ascribed to these losses in the paper are 20 per cent and 25 per cent, respectively. These represent rather liberal allowances for error. In cases where the loss due to unburned carbon is as high as 20 per cent, there is of course a possibility of a serious error in its measurement. Such a high loss, however, surely would manifest itself in some easily discernible way. Such high unaccounted-for losses are of rather infrequent occurrence. At least, few tests showing such high values for this item have been reported. A large "unaccounted-for" loss is more of a reflection on the accuracy of test methods than a small negative "unaccounted-for." If radiation has been closely estimated, the unaccounted-for loss consists almost entirely of errors. As these errors may be plus or minus, the unaccounted-for loss may be positive or negative. The fact that it is usually positive indicates a constant error in some measurements. Such an error may occur in the measurement of flue-gas temperature. This determination is usually in error on the low side due to radiation from the measuring instruments to the cooler walls of the duct.

The radiation loss from boiler settings is smaller than commonly assumed. There are plants now operating in which 1,000,000 lb. of water are evaporated hourly. This represents a coal consumption of approximately 100,000 lb. per hour. If 1 per cent of the heat generated is lost by radiation, this corresponds to the burning of 1000 lb. of coal per hour at 100 per cent efficiency to heat the boiler room. In any boiler room, if the air in the room is changed at the rate required for combustion, which is usually the case, the temperature of the boiler room would be about 40 to 50 deg. over the outside air temperature. This corresponds approximately to actual winter conditions. In the summer months this release rate would result in unbearable room temperatures.

The rate of heat release in the steam radiator, which represents an attempt to radiate as much heat as possible, is from 250 to

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300 B.t.u. per sq. ft. per hour. In the boiler setting, where everything possible is done to reduce the radiation and where average surface temperatures do not exceed those of a steam radiator, the rate of radiation surely will not be higher. Extremely high percentage radiations may occur at very low boiler ratings. Under banking conditions the radiation may approach or exceed 100 per cent. These conditions, however, are not representative of normal operation and surely not of test conditions.

Mr. Adler points out that weighing coal and water seems the more direct method and has a stronger appeal. This is largely a psychological matter. We like to deal with matter that we can see and feel. Modern physical measurements, however, deal largely with the invisible. We measure and weigh atoms and even electrons and measure visible and invisible light waves. It is largely a matter of getting used to the less familiar methods and using new tools.

While the location of air leaks may have an effect on magnitude of the dry-gas loss, as Mr. Adler states, it does not affect the measurement of the loss. The analysis of the gas and its temperature is a definite measure of the dry-gas loss, providing of course that the sample of gas and its temperature are taken at the same cross-section of the flue.

Mr. Worth points out the close agreement of results between the efficiencies obtained by weighed water and coal and by the heat-balance methods for the two groups of tests plotted in Figs. 6 and 7. He brings out the point that this is an indication of accuracy for the weighed-coal-and-water tests as well as for the heat-balance tests.

The two series of tests were selected because, in each case, considerable pains and expense were taken to install accurate water-weighing devices. The conditions at the plants enable all the water to be weighed to the system at moderate temperatures. Moreover, the amounts to be handled, while large, were not excessive. In short, the conditions at those two plants were in every way favorable for the accurate determination by weighing of all the water fed to the boiler. These two plants are unusual in this respect. In neither plant were the conditions for measuring losses more favorable than they would be in most other plants.

The question as to which is easier, the heat-balance method or the weighed-water-and-coal method, does not need be considered. Speaking paradoxically, when there is a choice, there is no choice; both methods should be used. There is no doubt that the heat-balance method is less expensive. While it may not be difficult in small plants to weigh coal and water, such plants are often limited in the amount of money they can spend for the purpose of testing, and the heat-balance method must therefore be used.

The weighing of ash is more difficult than the weighing of coal, because there is usually no provision for the routine weighing of ash as there is for coal. It is only necessary to establish rates. The heat balance test can be completed in a period not to exceed 6 or 8 hours. In small plants the entire ash for this period can be collected and weighed in wheelbarrows or ashcans without excessive labor. Larger plants are usually equipped with clinker grinders that deliver ash at uniform rates. In this case the discharge of ash for one hour can be collected and sampled. The test should be long enough so that the ash during this hour would result from burning the coal under the conditions of the test.

In pulverized-coal plants only a comparatively small percentage of the ash is deposited in the furnace. This can be weighed or even estimated with a degree of accuracy that will not introduce a serious error in efficiency. As a large part of the ash goes up the stack, the loss represented by this portion can be determined from the flue-dust sample. This stack loss also exists in stoker-fired boilers, and if neglected, as is commonly the case, appears in the item "unaccounted-for."

FSP-52-34

The Performance of Modern Steam-Generating Units

By C. F. HIRSHFELD[1] and G. U. MORAN[2], DETROIT, MICH

There has been much speculation during the past as to those factors which seem to be most responsible for steam-generating-unit outage. This is a question which always arises when a plant is being designed for a certain capacity factor, where the reliability or availability of the boiler units must be accurately foreseen.

Some have held that the outage increases directly with the severity of use. Others contend that the method of burning the fuel has a definite bearing on reliability. Others still have set different values for unit reliability on different kinds of fuel.

As the probable availability of units operating under certain conditions is a matter of utmost importance to the power-station designer and owner, it was thought that a complete and thorough study of the facts obtaining in the power-generating industry would be of great value. With that idea in mind the authors were requested by the Central Stations Committee of the Power Division to prepare this paper.

The study was made along the same general lines as the steam-turbine investigation made by the Prime Movers Committee of the N.E.L.A. The methods used in collecting information were as nearly identical as circumstances and the character of the units studied would permit. Daily operation logs were filled out for the period of one year by the operators of some two hundred boilers scattered about the country in sixty-odd stations. The data resulting were compiled by one group of men in such a manner that the results might be readily compared.

The units have been arranged in groups or classes according to methods of fuel burning and the working steam pressures, and then numbered in order according to their average rates of evaporation. With the data arranged in such a manner it has been possible to arrive at some rather definite conclusions.

THE studies reported in this paper were made at the request of the Central Stations Committee of the Power Division of the A.S.M.E. That committee outlined briefly what it had in mind, and the authors have followed these directions in so far as it proved possible to do so.

The rapid improvement in the thermal performance of steam-operated generating stations which has occurred in the past two decades has been amply treated in the literature of this subject.[3] Another and equally important development has not been presented as adequately. Reference is made to the better use of the dollar in the design and construction of such stations. Since the cost of producing power must include, among other things, both the cost of fuel and the charges proportional to investment, it is obvious that thermal efficiency purchased at an increase of investment which exactly balanced it would be almost meaningless economically except possibly for the questionable virtue of conserving the nation's fuel supply. There might be other indirect advantages such as the release of transportation facilities by the decreased demand for fuel, release of man power for other purposes, and so on. However, the statement made will obviously hold with respect to immediate and direct results.

It is an interesting fact that the increase of thermal efficiency that has been made has not been purchased by a counterbalancing increase of investment. A real advance has therefore been made in an economic sense as well as in a purely technical one.

During the period here under discussion the wages of unskilled and of skilled labor have also increased. As both types are used in steam-power production, it is but natural to find that design has progressed along such lines that it has also resulted in a marked decrease in the number of man-hours required per unit of output.

TABLE 1 NUMBER OF UNITS INCLUDED IN THE STUDY DIVIDED ACCORDING TO THE N.E.L.A. GEOGRAPHICAL DIVISIONS

Division	Number of units	Division	Number of units
Eastern	63	North Central	2
East Central	23	Northwest	0
Great Lakes	40	Pacific Coast	7
Middle Atlantic	0	Rocky Mountain	4
Middle West	18	Southeastern	3
New England	3	Southwestern	23
Total	186		

It is not pertinent to the present paper to discuss in detail the various means by which such desirable results have been produced. Suffice it for present purposes to say that among other changes the steam-generating units have increased greatly in size, in rate of output per unit of surface, and in complexity as well as in efficiency.

Once more in a general sense, low investment is not of itself a complete criterion on the dollar side of the picture. A large investment which can be and is used, for, say, 80 per cent of each complete year during the life of a plant, may represent a better economic solution than a smaller investment made to accomplish the same result, but usable only during, say, 50 per cent of the entire year. That is, the extent to which equipment can be kept in as continuous service as existing conditions indicate to be ideally desirable is also an important measure.

Thermal efficiency is comparatively easy of proof. Investment made is merely a question of fact. The performance factor, however, is a question of experience, and any one individual's intimate experience must necessarily be limited to a comparatively small number of installations. It results that while the world now knows a lot about the thermal efficiency of steam-operated generating plants and a lot about the first cost of such plants, it knows very little indeed about the extent to which various features of design influence the ability of those plants to produce when, as, and if required.

A beginning was made in the latter field some years ago when the Prime Movers Committee of the National Electric Light Association undertook an annual statistical study of the performances of representative steam-turbine installations for the

[1] Chief, Research Department, Detroit Edison Company. Mem. A.S.M.E.

[2] Research Department, Detroit Edison Company.

[3] See A.I.E.E. Power Generation Committee Reports; Trans. A.S.M.E., Contributions of Fuels and Steam Power Divisions; N.E.L.A. Proceedings, Prime Movers Committee Reports.

Contributed by the Power Division and presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

purpose of determining the extent and cause of unavoidable outages. The nature of the problem in that case was such that little could be determined with respect to the effect of the major features of design. But the facts with respect to ability to perform could be and were determined. Also, the weak parts of the rapid changes in the design of such equipment which have occurred in the past few years and in view of the economic importance of the ability of steam-generating equipment to remain on the line for very long periods of time, it appeared that such a statistical study would be desirable. The request of the Central Stations

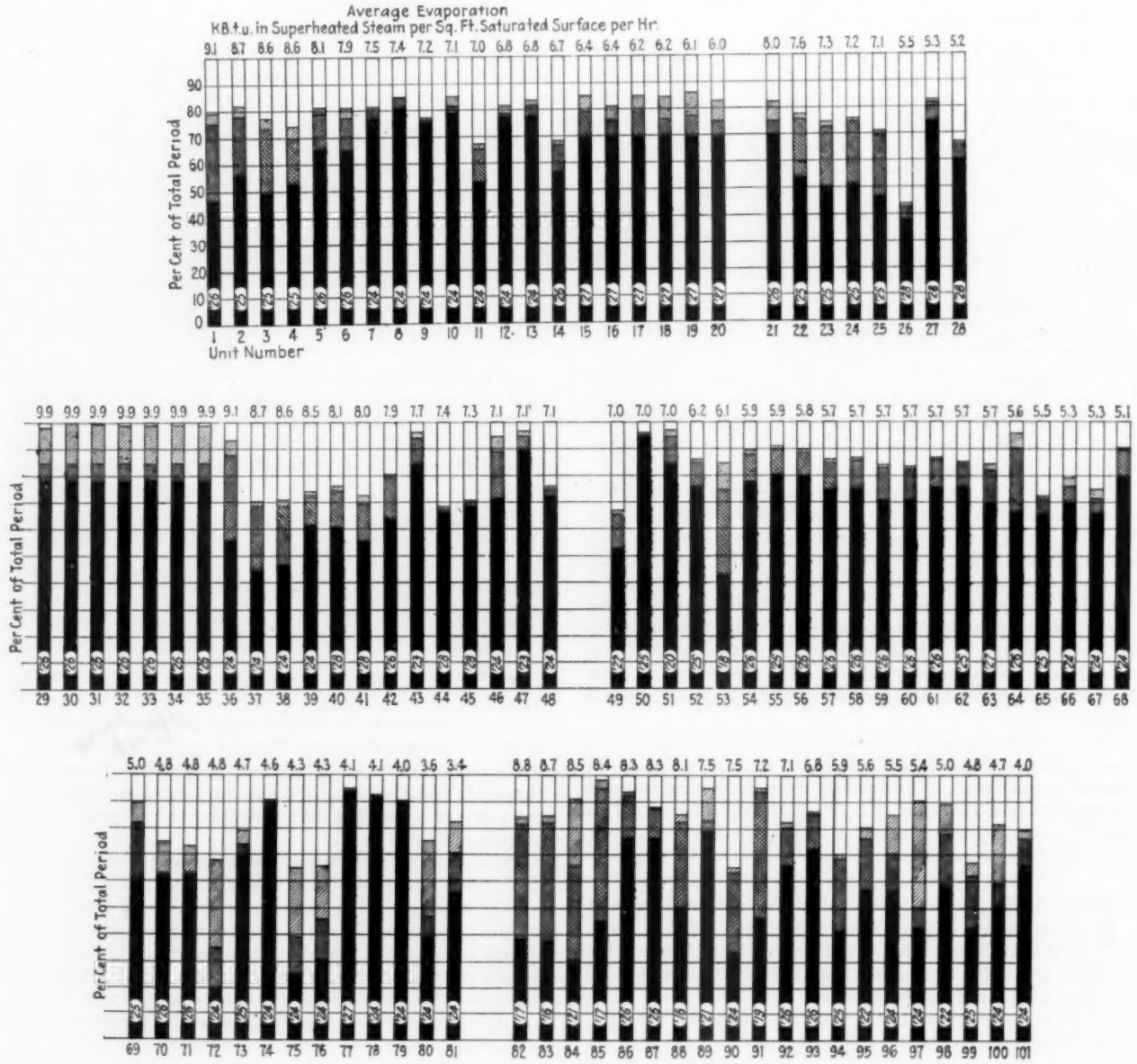


FIG. 1 RECORD OF OPERATION—STOKER-FIRED UNITS

UNITS	LEGENDS	
1–20, inc., 500 to 700-lb. pressure class	White	Crippled
21–28, inc., 400 to 499-lb. pressure class	Single hatching	Reserve
29–81, inc., 300 to 399-lb. pressure class	Double hatching	Banked
82–101, inc., 200 to 299-lb. pressure class	Black	Steaming

various assemblies were indicated, so that designers', builders', and operators' efforts could be concentrated on the weaknesses thus disclosed. A measuring stick was constructed and normal values were determined so that each individual had a means of ascertaining whether his own performance was good, bad, or normal.

No corresponding statistical studies of the performance of steam-generating equipment had been made. In view of the Committee referred to above was, in essence, a request to do just that sort of thing.

It is the hope of every central-station designer and operator that some day the steam-generating equipment may be brought to such a stage of perfection of design and construction that, when properly operated, it will have the same degree of availability for service as characterizes the modern steam turbine, attached generator, and subsidiary or auxiliary equipment.

Method Used in Making Study

The study was made along the same general lines as those of the steam-turbine investigation already referred to. The methods used in collecting information were as nearly identical as circumstances and the character of the units studied would permit. It was intended to include boilers operating in industrial plants as well as in central stations. It proved impossible to get the necessary cooperation from industrial plants, and the results herein contained apply only to central stations.

Those persons filling in the logs were requested to give the primary cause of each shutdown. The total outage for each unit was subdivided by the authors according to the reports sent in. The results of the compilation for a twelve-month period are shown in Figs. 1 to 8, inclusive, and also in Tables 2 to 4, inclusive.

Several stations, particularly those in industrial plants, did not send in reports for a period long enough to entitle them to an even footing with those reporting over the full period of twelve

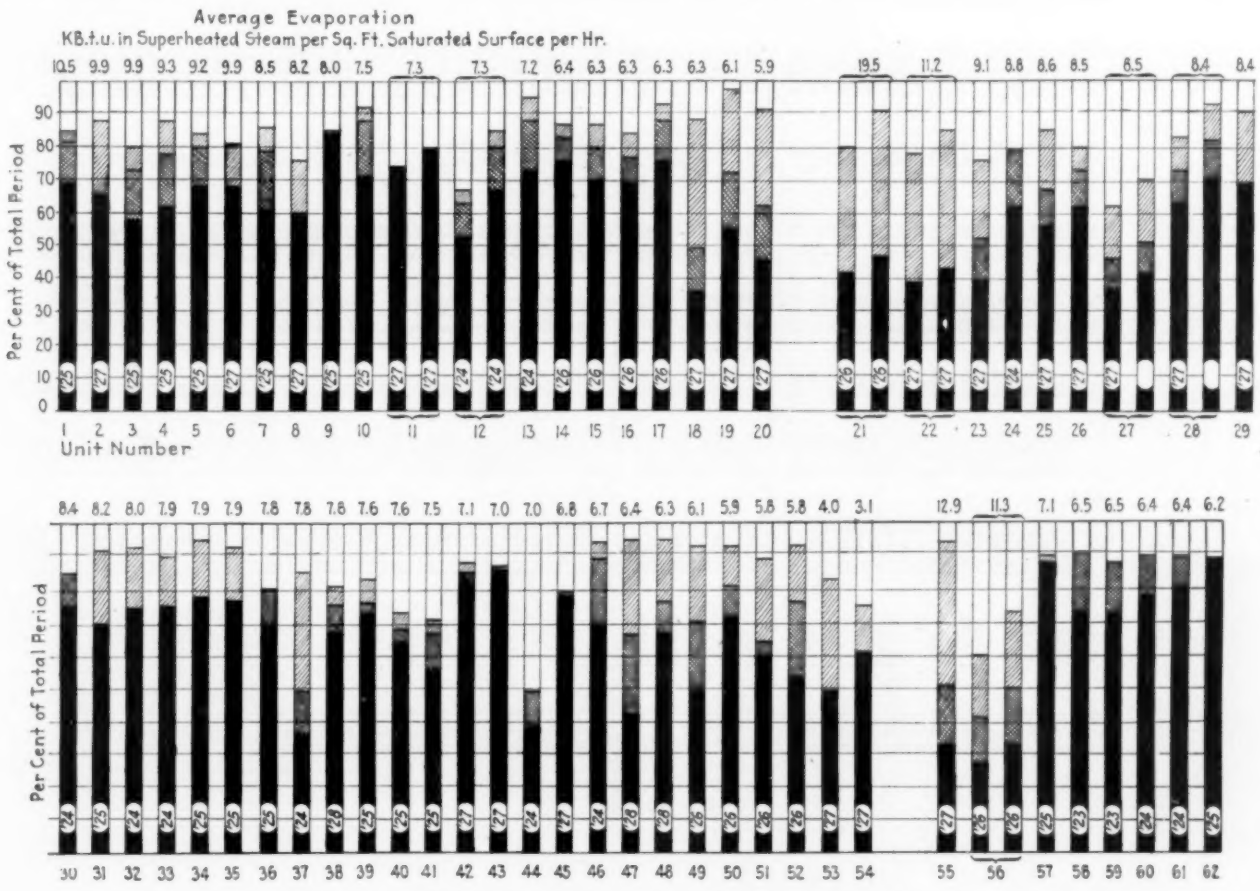


Fig. 2 Record of Operation—Pulverized-Fuel-Fired Units

Units	Legends
1–20, inc., 400 to 499-lb. class	White .....Crippled
21–54, inc., 300 to 399-lb. class	Single hatching .....Reserve
55–62, inc., 200 to 299-lb. class	Double hatching .....Banked
	Black .....Steaming

Notes: When excessive outage has been due to changes in design or other causes not directly chargeable to the unit, the unit has been presented both with and without such periods.

Daily operation logs were printed and furnished to the operators of 234 boilers scattered about the country in 64 stations. An indication of the distribution of the units studied is given in Table 1. A facsimile of the log sheet is published as in Appendix No. 1 to this paper.

Information has been gathered only on boilers operating above 250 lb. per sq. in. gage pressure and containing over 10,000 sq. ft. of heating surface, except that a few with slightly smaller surfaces and operating at very high ratings were included. Although steam reheating boilers were reported on, they have been omitted from this study. It was thought by the authors that they might be included in a future paper supplementing the present one, as a discussion of their particular problems would necessitate more detailed data than are now available.

months. This was also true of certain new installations which were put into operation some time after the investigation started. As the authors did not wish to throw out data indiscriminately, it was decided that the facts shown by a continuous report of ten months should be considered as being sufficiently indicative of actual conditions to warrant the inclusion of such reports in the paper.

These revisions in the original list of units reduced the total to 186 boilers in 50 stations, or an average of slightly over 3 units per station. The largest number from one station was 15, and the smallest, 1.

It must be realized, of course, that two persons or companies may have different opinions with regard to the interpretation of the headings shown on the log. For instance, there may be

TABLE 2 DESCRIPTIVE DATA AND DESIGN FACTORS

Unit No.	Steam pressure, lb. per sq. in. gage	Steam temperature, deg. fahr.	Heating surface in square feet: Boiler	Superheater	Water-cooled walls	Hearth screen	Air-cooled walls	Economizer	Air preheater	Total saturated heating surface	Volume of combustion chamber, cu. ft.	Projected grate area, sq. ft.	Ratio economizer to saturated surface	Ratio furnace volume to saturated surface	Ratio grate area to saturated surface	Heat value of fuel, B.t.u. per lb.		Unit No.
Stoker-Fired Units, 500 to 700 Lb. per Sq. In. Gage Pressure																		
S-1	625	725	14,086	2009	..	..	..	8,503	9,544	14,086	4,395	365	0.60	0.37	0.026	10,460	Traveling grate	S-1
S-2	625	735	14,086	2225	..	..	..	8,503	9,544	14,086	4,970	365	0.60	0.37	0.026	10,460	Traveling grate	S-2
S-3	625	725	14,086	2257	..	..	..	8,503	9,544	14,086	4,970	365	0.60	0.37	0.026	10,460	Traveling grate	S-3
S-4	625	725	14,086	2171	..	..	..	8,503	9,544	14,086	4,970	365	0.60	0.37	0.026	10,460	Traveling grate	S-4
S-5	625	725	14,086	2255	..	..	..	8,503	9,544	14,086	4,395	365	0.60	0.37	0.026	10,460	Traveling grate	S-5
S-6	625	725	14,086	2170	..	..	..	8,503	9,544	14,086	4,395	365	0.60	0.37	0.026	10,460	Traveling grate	S-6
S-7	650	725	14,086	2485	..	..	..	8,837	...	14,086	5,760	365	0.63	0.41	0.026	11,240	Traveling grate	S-7
S-8	650	725	14,086	2485	..	..	..	8,837	...	14,086	5,760	365	0.63	0.41	0.026	11,240	Traveling grate	S-8
S-9	650	725	14,086	2485	..	..	..	8,837	12,000	14,086	5,760	365	0.63	0.41	0.026	11,240	Traveling grate	S-9
S-10	650	725	14,086	2485	..	..	..	8,837	...	14,086	5,760	365	0.63	0.41	0.026	11,240	Traveling grate	S-10
S-11	575	750	16,615	2872	..	..	..	10,250	9,948	16,615	6,000	449	0.60	0.36	0.027	10,215	Traveling grate	S-11
S-12	650	725	14,086	2485	..	..	..	8,837	...	14,086	5,760	365	0.63	0.41	0.026	11,240	Traveling grate	S-12
S-13	650	725	14,086	2485	..	..	..	8,837	...	14,086	5,760	365	0.63	0.41	0.026	11,240	Traveling grate	S-13
S-14	575	750	17,280	3850	1253	..	..	7,376	15,913	18,533	5,750	444	0.86	0.31	0.025	10,215	Traveling grate	S-14
S-15	675	725	17,962	2800	..	..	..	3,011	81,000	17,962	7,790	528	0.17	0.44	0.03	10,834	Traveling grate	S-15
S-16	675	725	17,962	2800	..	..	..	3,011	81,000	17,962	7,790	528	0.17	0.44	0.03	10,834	Traveling grate	S-16
S-17	675	725	17,962	2800	..	..	..	3,011	81,000	17,962	7,790	528	0.17	0.44	0.03	10,834	Traveling grate	S-17
S-18	675	725	17,962	2800	..	..	..	3,011	81,000	17,962	7,790	528	0.17	0.44	0.03	10,834	Traveling grate	S-18
S-19	675	725	17,962	2800	..	..	..	3,011	81,000	17,962	8,500	528	0.17	0.48	0.03	10,834	Traveling grate	S-19
S-20	675	725	17,962	2800	..	..	..	3,011	81,000	17,962	7,790	528	0.17	0.44	0.03	10,834	Traveling grate	S-20
Stoker-Fired Units, 400 to 499 Lb. per Sq. In. Gage Pressure																		
S-21	400	707	15,697	2822	597	..	..	7,515	22,072	16,294	7,780	337	0.46	0.48	0.02	13,841	Underfeed	S-21
S-22	420	700	10,438	2925	..	..	..	6,480	...	10,438	5,400	270	0.62	0.52	0.026	11,570	Underfeed	S-22
S-23	420	700	10,438	2925	..	..	..	6,480	...	10,438	5,400	270	0.62	0.52	0.026	11,570	Underfeed	S-23
S-24	420	700	10,438	2925	..	..	..	6,480	...	10,438	5,400	270	0.62	0.52	0.026	11,570	Underfeed	S-24
S-25	420	700	10,438	2925	..	..	..	6,480	...	10,438	5,400	270	0.62	0.52	0.026	11,570	Underfeed	S-25
S-26	400	720	22,900	4350	5336	..	..	...	90,000	28,236	8,530	500	..	0.30	0.02	14,098	Underfeed	S-26
S-27	400	720	22,900	4350	5336	..	..	...	90,000	28,236	8,530	500	..	0.30	0.02	14,098	Underfeed	S-27
S-28	400	720	22,900	4350	5336	..	..	...	90,000	28,236	8,530	500	..	0.30	0.02	14,098	Underfeed	S-28
Stoker-Fired Units, 300 to 399 Lb. per Sq. In. Gage Pressure																		
S-29	350	700	16,396	3625	..	..	..	10,703	...	16,396	6,736	257	0.67	0.41	0.016	13,080	Underfeed	S-29
S-30	350	680	16,396	3625	250	..	..	10,703	...	16,646	6,736	257	0.66	0.40	0.015	13,080	Underfeed	S-30
S-31	350	700	16,396	3625	..	..	..	10,703	...	16,396	6,736	257	0.65	0.41	0.016	13,080	Underfeed	S-31
S-32	350	700	16,396	3625	..	..	..	10,703	...	16,396	6,736	257	0.65	0.41	0.016	13,080	Underfeed	S-32
S-33	350	700	16,396	3625	..	..	..	10,703	...	16,396	6,736	257	0.65	0.41	0.016	13,080	Underfeed	S-33
S-34	350	700	16,396	3625	..	..	..	10,703	...	16,396	6,736	257	0.65	0.41	0.016	13,080	Underfeed	S-34
S-35	350	700	16,396	3625	..	..	..	10,703	...	16,396	6,736	257	0.65	0.41	0.016	13,080	Underfeed	S-35
S-36	350	630	14,579	4220	..	..	..	7,728	...	14,579	4,050	370	0.53	0.28	0.026	14,199	Underfeed	S-36
S-37	385	692	10,636	5000	575	..	..	11,340	...	11,211	5,200	324	1.0	0.47	0.029	13,128	Underfeed	S-37
S-38	385	692	10,636	5000	575	..	..	11,340	...	11,211	5,200	324	1.0	0.47	0.029	13,128	Underfeed	S-38
S-39	325	650	15,089	4052	..	..	..	12,250	8,954	15,089	4,800	410	0.81	0.32	0.027	10,395	Traveling grate	S-39
S-40	385	700	12,340	5704	749	..	..	4,800	56,000	13,089	5,500	385	0.37	0.42	0.03	13,128	Underfeed	S-40
S-41	385	700	12,340	5704	749	..	..	4,800	56,000	13,089	5,500	385	0.37	0.42	0.03	13,128	Underfeed	S-41
S-42	385	720	23,600	4130	..	..	..	...	...	23,600	8,960	394	..	0.38	0.017	14,098	Underfeed	S-42
S-43	325	700	13,828	..	..	..	..	8,400	...	13,828	5,400	182	0.61	0.39	0.013	11,286	Traveling grate	S-43
S-44	360	725	11,959	4500	850	..	..	5,880	...	12,809	6,850	361	0.46	0.53	0.028	11,225	Underfeed	S-44
S-45	360	725	11,959	4500	850	..	..	5,880	...	12,809	6,850	361	0.46	0.53	0.028	11,225	Underfeed	S-45
S-46	375	700	19,743	2938	..	..	190	11,091	...	19,743	9,000	374	0.56	0.46	0.019	13,965	Underfeed	S-46
S-47	325	700	13,828	..	..	..	..	8,400	...	13,828	5,400	182	0.61	0.39	0.013	11,286	Traveling grate	S-47
S-48	360	725	11,959	4500	850	..	..	5,880	...	12,809	6,850	361	0.46	0.54	0.028	11,225	Underfeed	S-48
S-49	385	700	12,337	5527	760	..	..	4,800	56,000	13,097	5,500	385	0.37	0.42	0.03	13,128	Underfeed	S-49
S-50	325	700	18,933	5620	..	..	..	11,464	...	18,933	8,400	494	0.60	0.45	0.026	11,286	Traveling grate	S-50
S-51	325	700	13,828	..	..	..	..	8,400	...	13,828	4,160	182	0.61	0.30	0.013	11,286	Traveling grate	S-51
S-52	385	720	23,600	4130	..	..	..	...	...	23,600	9,510	345	..	0.40	0.015	14,098	Underfeed	S-52
S-53	350	630	12,322	3174	..	..	..	7,728	...	12,322	3,800	240	0.63	0.31	0.02	14,199	Underfeed	S-53
S-54	385	720	23,600	4130	..	..	..	...	...	23,600	9,510	345	..	0.40	0.015	14,098	Underfeed	S-54
S-55	385	720	23,600	4130	..	..	..	...	...	23,600	9,510	345	..	0.40	0.015	14,098	Underfeed	S-55
S-56	385	720	23,600	4130	..	..	..	...	...	23,600	9,510	345	..	0.40	0.015	14,098	Underfeed	S-56
S-57	385	720	23,600	4130	..	..	..	...	...	23,600	9,510	345	..	0.40	0.015	14,098	Underfeed	S-57
S-58	385	720	23,600	4130	..	..	..	...	...	23,600	8,960	394	..	0.38	0.017	14,098	Underfeed	S-58
S-59	385	720	23,600	4130	..	..	..	...	...	23,600	8,960	394	..	0.38	0.017	14,098	Underfeed	S-59
S-60	385	720	23,600	4130	..	..	..	...	...	23,600	8,960	394	..	0.38	0.017	14,098	Underfeed	S-60
S-61	385	720	23,600	4130	..	..	..	...	...	23,600	8,960	394	..	0.38	0.017	14,098	Underfeed	S-61
S-62	385	720	23,600	4130	..	..	..	...	...	23,600	9,510	345	..	0.40	0.015	14,098	Underfeed	S-62
S-63	385	720	23,600	4130	..	..	..	...	...	23,600	8,960	394	..	0.38	0.017	14,098	Underfeed	S-63
S-64	350	650	17,257	2424	..	..	..	...	...	17,257	4,014	202	..	0.23	0.012	10,939	Traveling grate	S-64
S-65	310	650	13,850	2735	1060	..	..	...	...	14,910	6,900	280	..	0.46	0.019	13,890	Underfeed	S-65
S-66	310	600	13,850	2735	1064	..	..	...	...	14,914	6,900	280	..	0.46	0.019	13,890	Underfeed	S-66
S-67	310	600	13,850	2735	1060	..	..	...	...	14,910	6,900	280	..	0.46	0.019	13,890	Underfeed	S-67

S-68	310	650	14,092	1900	907	..	..	...	.	14,999	6,900	280	..	0.46	0.019	13,890	Underfeed	S-68
S-69	350	650	17,257	2424	..	..	..	...	..	17,257	4,014	202	..	0.23	0.012	10,939	Traveling grate	S-69
S-70	375	700	17,257	1877	..	..	..	...	..	17,257	4,687	483	..	0.27	0 028	10,541	Traveling grate	S-70
S-71	375	700	17,257	1877	..	..	..	...	...	17,257	4,687	483	...	0.27	0.028	10,541	Traveling grate	S-71
S-72	350	700	11,959	2010	..	..	..	3,245	...	11,959	6,358	272	0.27	0.53	0.023	11,225	Underfeed	S-72
S-73	375	700	17,257	1877	..	..	..	...	...	17,257	4,687	483	..	0.27	0.028	10,541	Traveling grate	S-73
S-74	300	650	10,084	?	..	..	..	...	...	10,084	2,886	179	..	0.29	0.018	12,048	Underfeed	S-74
S-75	350	700	11,959	2010	..	..	..	3,245	...	11,959	6,358	304	0.27	0.53	0 025	11,225	Underfeed	S-75
S-76	350	700	11,959	2010	..	..	..	3,245	...	11,959	6,358	272	0.27	0.53	0 023	11,225	Underfeed	S-76
S-77	300	650	10,084	?	..	..	..	...	...	10,084	2,886	179	..	0.29	0 018	12,048	Underfeed	S-77
S-78	300	650	10,084	?	..	..	..	...	...	10,084	2,886	179	..	0.29	0 018	12,048	Underfeed	S-78
S-79	300	650	10,084	?	..	..	..	...	...	10,084	2,886	179	..	0.29	0.018	12,048	Underfeed	S-79
S-80	300	700	18,772	4238	..	..	..	...	...	18,772	5,318	258	..	0.28	0.014	13,968	Underfeed	S-80
S-81	300	700	18,772	4238	..	..	..	...	...	18,772	5,100	257	..	0.27	0.014	13,968	Underfeed	S-81
Stoker-Fired Units, 200 to 299 Lb. per Sq. In. Gage Pressure																		
S-82	275	750	12,515	3815	..	..	..	9,435	...	12,515	4,250	?	0.75	0.34	?	12,376	Underfeed	S-82
S-83	275	750	12,515	3815	..	..	..	9,435	...	12,515	4,250	?	0.75	0 34	?	12,376	Underfeed	S-83
S-84	275	600	13,840	3863	..	..	..	8,256	...	13,840	4,150	284	0.60	0.30	0.02	13,029	Underfeed	S-84
S-85	275	750	12,515	3815	..	..	..	9,435	...	12,515	4,250	?	0.75	0.34	?	12,376	Underfeed	S-85
S-86	275	650	11,120	1999	210	..	..	6,674	3,600	11,330	4,900	309	0.58	0 43	0.027	13,475	Underfeed	S-86
S-87	275	650	11,120	1999	210	..	..	6,674	3,600	11,330	4,900	309	0.58	0.43	0.027	13,475	Underfeed	S-87
S-88	275	750	12,515	3815	..	..	..	9,435	...	12,515	4,250	?	0.75	0.34	?	12,376	Underfeed	S-88
S-89	275	600	12,625	3863	..	..	..	8,256	...	12,625	4,450	264	0.66	0.36	0.021	13,029	Underfeed	S-89
S-90	275	750	12,100	4000	?	..	..	3,145	...	12,100	4,250	?	0.26	0.36	?	12,376	Underfeed	S-90
S-91	275	750	12,515	3815	..	..	..	9,435	...	12,515	4,250	?	0.75	0.34	?	12,376	Underfeed	S-91
S-92	275	650	11,120	1999	210	..	..	6,674	3,600	11,330	4,900	309	0.58	0.43	0.027	13,475	Underfeed	S-92
S-93	275	650	11,120	1999	210	..	..	6,674	3,600	11,330	4,900	309	0.58	0.43	0.027	13,475	Underfeed	S-93
S-94	265	600	14,217	2582	488	..	..	..	50,276	14,705	6,650	310	..	0.45	0.021	13,727	Underfeed	S-94
S-95	290	700	28,210	6338	..	..	..	16,350	...	28,210	10,800	377	0.58	0.39	0.014	12,700	Underfeed	S-95
S-96	290	700	29,130	5900	..	..	..	16,350	...	29,130	15,000	474	0.56	0.52	0.016	12,700	Underfeed	S-96
S-97	290	700	29,130	5900	..	..	..	16,350	...	29,130	15,000	474	0.56	0.52	0.016	12,700	Underfeed	S-97
S-98	290	700	28,210	6338	..	..	..	16,350	...	28,210	10,800	377	0.58	0.39	0.014	12,700	Underfeed	S-98
S-99	265	600	14,217	2582	488	..	..	...	42,000	14,705	6,650	310	..	0.45	0.021	13,727	Underfeed	S-99
S-100	265	620	14,217	2582	..	..	131	6,853	...	14,217	7,200	310	0 48	0.51	0.022	13,780	Underfeed	S-100
S-101	265	600	14,217	2582	..	..	157	5,250	22,072	14,217	6,650	310	0.37	0.47	0.022	13,727	Underfeed	S-101
Pulverized-Fuel-Fired Units, 400 to 499 Lb. per Sq. In. Gage Pressure																		
P-1	400	700	13,920	6000	..	?	1150	8,448	?	13.920	14,300	...	0.60	1.0	...	9,751	Storage	P-1
P-2	400	750	18,096	2870	830	..	1400	...	?	18.922	14.267	...	..	0.75	...	10,609	Storage	P-2
P-3	400	700	13,920	6000	..	?	1150	8,448	...	13.920	14,300	...	0.60	1.0	...	9,751	Storage	P-3
P-4	400	700	13.920	6000	..	?	1150	8,448	...	13.920	14,300	...	0 60	1.0	...	9,751	Storage	P-4
P-5	416	700	29,087	6070	..	1220	2400	18,849	...	30,307	25.140	...	0.63	0.83	...	13,630	Storage	P-5
P-6	400	750	18,092	2901	830	..	1400	...	?	18,922	14,267	...	..	0.75	...	10,609	Storage	P-6
P-7	400	700	13,920	6000	..	?	1150	8,448	...	13,920	14.300	...	0.60	1 0	...	9,751	Storage	P-7
P-8	400	750	17,864	3608	790	337	1400	...	?	18,991	14,267	...	..	0.75	...	10,609	Storage	P-8
P-9	400	700	21,620	3545	2578	..	..	...	16,400	24.198	18,000	...	..	0.75	...	12,385	Storage	P-9
P-10	416	700	29,087	6070	..	1220	2400	18,849	...	30,307	25.140	...	0.63	0.83	...	13,630	Storage	P-10
P-11	400	700	21,620	3545	2578	..	..	...	16,400	24,198	18.000	...	..	..	...	12,385	Storage	P-11
P-12	416	700	29.087	6070	..	1220	2400	18,849	...	30,307	25,140	...	0.63	0.83	...	13,630	Storage	P-12
P-13	416	700	29,087	6070	..	1220	2400	18,849	...	30,307	25,140	...	0.63	0.83	...	13,630	Storage	P-13
P-14	400	700	18,756	4000	1805	715	1150	...	...	21,276	18,539	...	..	0.88	...	7,565	Storage	P-14
P-15	400	700	18,756	4000	1805	715	1157	...	...	21,276	18,539	...	..	0.88	...	7,565	Storage	P-15
P-16	400	700	18,756	4000	1805	715	1157	...	...	21,276	18,539	...	..	0.88	...	7,565	Storage	P-16
P-17	400	700	18,756	4000	1805	715	1157	...	...	21,276	18,539	...	..	0.88	...	7,565	Storage	P-17
P-18	400	700	15,360	?	1450	320	..	...	21,600	17,130	8,250	...	..	0.48	...	12,880	Unit	P-18
P-19	400	700	15,360	?	1450	320	..	...	21,600	17,130	8,250	...	..	0.48	...	12,880	Unit	P-19
P-20	400	700	15,360	?	1450	320	..	...	21,600	17,130	8,250	...	..	0.48	...	12,880	Unit	P-20
Pulverized-Fuel-Fired Units, 300 to 399 Lb. per Sq. In. Gage Pressure																		
P-21	375	690	5,940	3000	2468	..	..	8,850	41,700	8,408	12,397	...	1.0	1.5	...	10,395	Storage	P-21
P-22	375	690	6,811	2430	1772	1402	..	5,250	25,000	9,985	5,000	...	0.52	0.50	...	10,395	Unit	P-22
P-23	385	700	9,571	3639	842	483	729	5,016	10,980	10,896	12,800	...	0.46	1.2	...	13,938	Storage	P-23
P-24	375	640	10,850	6288	..	275	1652	2,470	...	11,125	8.160	...	0.23	0.74	...	10,250	Storage	P-24
P-25	385	700	9,571	3639	842	483	729	5,016	10,980	10,896	12,800	...	0.46	1.2	...	13,938	Storage	P-25
P-26	385	700	9,571	3639	842	483	729	5,016	10,980	10,896	12,800	...	0.46	1.2	...	13,938	Storage	P-26
P-27	385	700	9,571	3639	842	483	729	5,016	10,980	10,896	12,800	...	0.46	1.2	...	13,938	Storage	P-27
P-28	385	700	9,571	3639	842	483	729	5,016	10,980	10,896	12,800	...	0.46	1.2	...	13,938	Storage	P-28
P-29	350	660	14,770	2130	2646	685	..	...	17,280	18 101	11,000	...	..	0.61	...	13,804	Storage	P-29
P-30	375	640	10,850	6288	..	275	1652	2,470	...	11,125	8,160	...	0.23	0.74	...	10,250	Storage	P-30
P-31	350	660	14,770	2130	..	900	3275	7,776	...	15,670	11,000	...	0.49	0.70	...	13,804	Storage	P-31
P-32	350	660	14,770	2130	..	450	3275	7,776	...	15,220	11,000	...	0.51	0.72	...	13,804	Storage	P-32
P-33	350	660	14,770	2130	..	450	3275	7,776	...	15,220	11,000	...	0.51	0.72	...	13,804	Storage	P-33
P-34	350	660	14,770	2130	..	450	3275	7,776	...	15,220	11,000	...	0.51	0.72	...	13,804	Storage	P-34
P-35	350	660	14,770	2130	..	450	3275	7,776	...	15,220	11,000	...	0.51	0.72	...	13,804	Storage	P-35
P-36	380	700	20,468	4094	2050	..	?	...	11,700	22,518	13,500	...	..	0.60	...	12,385	Storage	P-36
P-37	375	640	10,850	6288	..	275	1652	2,470	...	11,125	8,160	...	0.23	0.74	...	10,250	Storage	P-37
P-38	385	700	9,571	3639	842	483	729	5,016	10,980	10,896	12,800	...	0.46	1.2	...	13,938	Storage	P-38
P-39	380	700	20,468	4094	2050	..	?	...	11,700	22,518	13,500	...	..	0.60	...	12,385	Storage	P-39
P-40	380	700	20,468	4094	2050	..	?	...	11,700	22,518	13,500	...	..	0.60	...	12,385	Storage	P-40

TABLE 2 DESCRIPTIVE DATA AND DESIGN FACTORS (*Continued from page 269*)

Unit No.	Steam pressure, lb. per sq. in. gage	Steam temperature, deg. fahr.	Heating surface in square feet: Boiler	Superheater	Water-cooled walls	Hearth screen	Air-cooled walls	Economizer	Air preheater	Total saturated heating surface	Volume of combustion chamber, cu. ft.	Projected grate area, sq. ft.	Ratio economizer to saturated surface	Ratio furnace volume to saturated surface	Ratio grate area to saturated surface	Heat value of fuel, B.t.u. per lb.		Unit No.
P-41	380	700	20,468	4094	2050	..	?	...	11,700	22,518	13,500	...	..	0.60	...	12,385	Storage	P-41
P-42	305	675	27,780	5480	1980	520	..	...	112,000	30,280	4,800	...	..	0.60	...	13,163	Storage	P-42
P-43	300	700	13,138	?	850	..	...	3,240	45,000	13,988	5,270	...	0.23	0.38	...	12,048	Unit	P-43
P-44	375	640	10,850	6288	..	275	1652	2,470	...	11,125	8,160	...	0.23	0.74	...	10,250	Storage	P-44
P-45	300	700	13,138	?	850	..	..	3,240	45,000	13,988	5,270	...	0.23	0.38	...	12,048	Unit	P-45
P-46	317	610	17,650	?	..	630	?	13,700	...	18,280	12,000	...	0.75	0.66	...	13,508	Storage	P-46
P-47	390	725	30,600	6550	..	792	2490	21,120	...	31,392	30,600	...	0.67	0.98	...	12,723	Storage	P-47
P-48	390	725	30,600	6550	..	792	2490	21,120	...	31,392	30,600	...	0.67	0.98	...	12,723	Storage	P-48
P-49	390	725	30,600	6550	..	792	2490	21,120	...	31,392	30,600	...	0.67	0.98	...	12,723	Storage	P-49
P-50	390	725	30,600	6550	..	792	2490	21,120	...	31,392	30,600	...	0.67	0.98	...	12,723	Storage	P-50
P-51	390	725	30,600	6550	..	792	2490	21,120	...	31,392	30,600	...	0.67	0.98	...	12,723	Storage	P-51
P-52	390	725	30,600	6550	..	792	2490	21,120	...	31,392	30,600	...	0.67	0.98	...	12,723	Storage	P-52
P-53	330	585	12,795	4780	1705	..	..	...	53,400	14,500	9,600	...	..	0.66	...	13,095	Unit	P-53
P-54	330	605	13,135	4100	1214	..	..	...	53,400	14,349	9,600	...	..	0.67	...	13,095	Unit	P-54
			Pulverized-Fuel-Fired Units, 200 to 299 Lb. per Sq. In. Gage Pressure															
P-55	275	750	12,515	7100	2044	..	..	9,504	...	14,559	11,000	...	0.65	0.76	...	12,376	Unit	P-55
P-56	275	750	12,515	7100	2725	..	..	9,504	...	15,240	10,200	...	0.62	0.67	...	12,376	Unit	P-56
P-57	275	675	27,680	5657	1425	640	1030	...	13,000	29,745	17,050	..	..	0.57	...	13,163	Storage	P-57
P-58	265	725	30,600	9040	..	835	1800	22,080	...	31,445	26,000	..	0.71	0.84	...	12,370	Storage	P-58
P-59	265	725	30,600	9040	..	835	1800	22,080	...	31,445	26,000	...	0.71	0.84	...	12,370	Storage	P-59
P-60	265	725	30,600	9040	..	835	1800	22,080	...	31,445	26,000	...	0.71	0.84	...	12,370	Storage	P-60
P-61	265	725	30,600	9040	..	835	1800	22,080	...	31,445	26,000	...	0.71	0.84	...	12,370	Storage	P-61
P-62	275	675	26,510	5657	1425	640	1030	...	13,000	28,575	17,050	...	..	0.61	...	13,163	Storage	P-62
			Oil- and Gas-Fired Units															
OG-1	375	700	14,200	3210	609	..	..	...	74,600	14,809	5,335	...	..	0.36	...	...	Comb. oil and gas	OG-1
OG-2	450	750	34,160	6553	3220	..	..	...	51,232	37,380	19,800	...	..	0.54	...	...	Comb. oil and gas	OG-2
OG-3	425	725	15,630	4137	2729	338	..	...	84,000	18,359	16,500	...	..	0.90	...	...	Oil	OG-3
OG-4	400	725	15,340	5780	600	..	224	6,804	32,800	15,940	8,000	...	0.43	0.50	...	...	Comb. oil and gas	OG-4
OG-5	425	725	17,440	4137	2729	338	..	...	84,000	20,507	16,500	...	..	0.83	...	...	Oil	OG-5
OG-6	425	725	17,440	4137	2729	338	..	...	84,000	20,507	16,500	...	..	0.83	...	...	Oil	OG-6
OG-7	375	700	15,300	3712	1440	414	394	...	16,850	17,154	8,650	...	..	0.50	...	...	Comb. oil and gas	OG-7
OG-8	375	700	15,400	3587	1263	349	396	...	16,850	17,012	8,650	...	..	0.51	...	...	Comb. oil and gas	OG-8
OG-9	400	700	25,560	4072	..	..	1614	...	21,574	22,560	9,700	...	..	0.43	...	...	Comb. oil and gas	OG-9
OG-10	350	650	17,041	2361	..	..	952	...	...	17,041	5,502	...	..	0.32	...	...	Comb. oil and gas	OG-10
OG-11	350	650	17,041	2361	..	..	952	...	...	17,041	5,502	...	..	0.32	...	...	Comb. oil and gas	OG-11
OG-12	350	650	17,041	2361	..	..	952	...	...	17,041	5,502	...	..	0.32	...	...	Comb. oil and gas	OG-12
OG-13	400	700	25,560	4072	..	..	1614	...	21,574	22,560	9,700	...	..	0.43	...	...	Comb. oil and gas	OG-13
OG-14	385	700	25,450	3331	..	..	1614	...	23,025	25,450	9,700	...	..	0.39	...	...	Comb. oil and gas	OG-14
OG-15	350	650	17,041	2361	..	..	952	...	...	17,041	5,502	...	..	0.32	...	...	Comb. oil and gas	OG-15
OG-16	375	685	12,500	2400	330	..	..	1,665	...	12,830	6,600	...	0.13	0.52	...	...	Comb. oil and gas	OG-16
OG-17	400	700	25,560	4072	..	..	1614	...	21,574	22,560	9,700	...	..	0.43	...	...	Comb. oil and gas	OG-17
			Miscellaneous Combinations															
M-1	375	700	16,863	5000	2570	540	..	...	25,452	19,973	14,300	...	..	0.72	...	...	Unit pulverizers with oil and gas	M-1
M-2	375	700	16,863	5000	2570	540	..	...	25,452	19,973	14,300	...	..	0.72	...	...		M-2
M-3	375	700	16,863	5000	2570	540	..	...	25,452	19,973	14,300	...	..	0.72	...	...		M-3
M-4	375	685	12,500	2400	330	..	..	1,665	...	12,830	6,600	...	0.13	0.52	...	...		M-4
M-5	375	685	12,500	2400	330	..	..	1,665	...	12,830	6,600	...	0.13	0.52	...	...	Traveling grates with oil and gas	M-5
M-6	375	685	12,500	2400	330	..	..	1,665	...	12,830	6,600	...	0.13	0.52	...	...		M-6

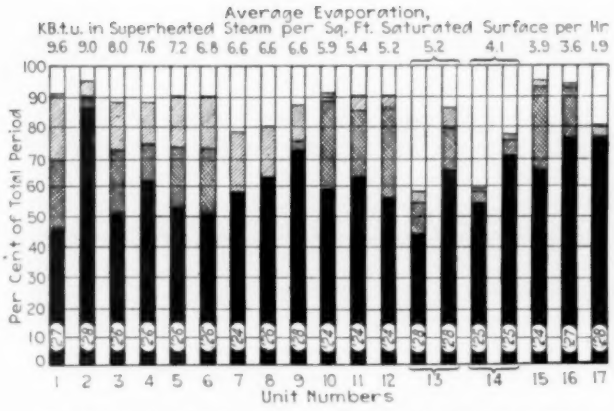


FIG. 3 RECORD OF OPERATION—OIL- AND GAS-FIRED UNITS

LEGENDS: White.....Crippled
Single hatching.....Reserve
Double hatching.....Banked
Black.....Steaming

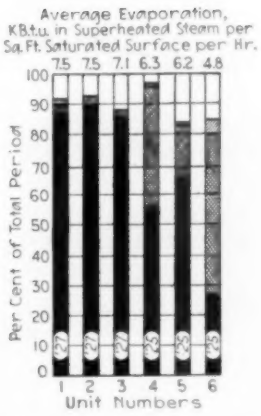


FIG. 4 RECORD OF OPERATION—MISCELLANEOUS COMBINATIONS

LEGENDS: White.....Crippled
Single hatching.....Reserve
Double hatching.....Banked
Black.....Steaming

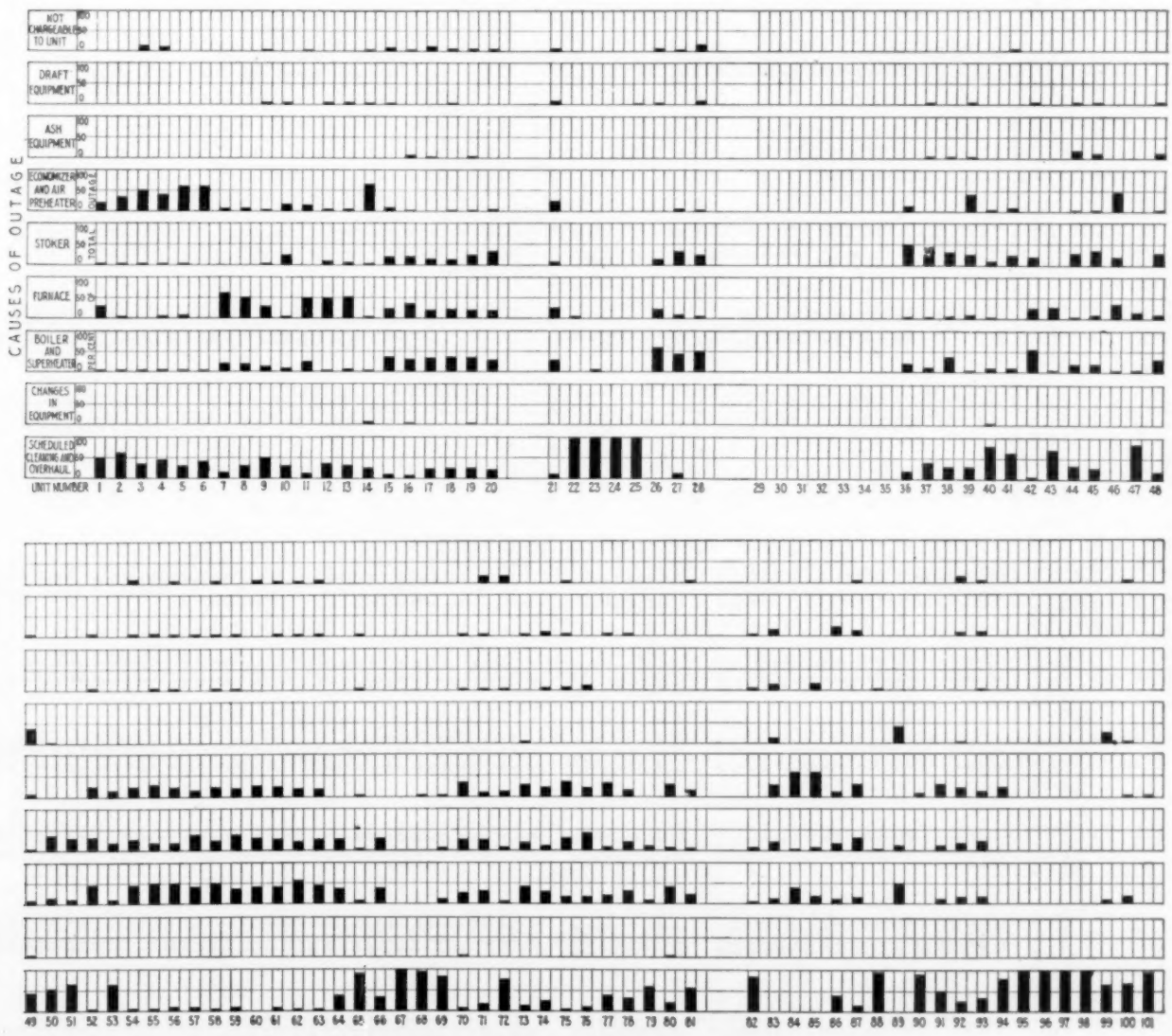


FIG. 5 ANALYSIS OF OUTAGE—STOKER-FIRED STEAM-GENERATING UNITS

(Crippled periods analyzed as to cause and shown as percentages of total outage.)

UNITS: 1–20, inc., 500 to 700-lb. class; 21–28, inc., 400 to 499-lb. class; 29–81, inc., 300 to 399-lb. class; 82–101, inc., 200 to 299-lb. class

TABLE 3 ANALYSIS OF TIME UTILIZATION OF INDIVIDUAL STEAM-GENERATING UNITS

Unit No.	Percentages of total period: Steaming period, per cent	Banked period, per cent	Reserve period, per cent	Crippled period, per cent	Outages classified as to cause expressed as percentages of total outage: Scheduled cleaning and overhaul, per cent	Changes in equipment, per cent	Boiler and superheater, per cent	Furnace, per cent	Stoker, per cent	Pulverizing equipment and burners, per cent	Economizer air preheater, per cent	Ash equipment, per cent	Draft equipment, per cent	Not chargeable to unit, per cent	Availability factor	Use factor	Unit No.
Stoker-Fired Units, 500 to 700 Lb. per Sq. In. Gage Pressure																	
S-1	47.5	28.2	3.1	21.1	46.0	..	3.8	29.4	0.5	..	20.3	..	..	..	78.9	75.7	S-1
S-2	56.3	21.9	3.5	18.3	60.2	..	2.8	1.8	1.0	..	34.2	..	..	..	81.7	78.2	S-2
S-3	48.4	25.0	3.6	23.0	33.0	..	1.1	..	3.3	..	49.5	..	..	12.8	77.0	73.4	S-3
S-4	53.0	16.1	4.8	26.1	43.0	..	4.7	4.7	0.6	..	36.6	..	..	10.4	73.9	69.1	S-4
S-5	66.2	13.1	1.7	19.0	28.6	..	4.1	6.5	0.4	..	60.4	..	..	..	81.0	79.3	S-5
S-6	65.1	12.4	3.4	19.1	37.8	..	2.2	..	..	..	60.0	..	..	..	80.9	77.5	S-6
S-7	77.5	2.0	1.7	18.8	12.5	..	20.0	61.4	..	..	6.1	..	..	..	81.2	79.5	S-7
S-8	81.2	3.7	0.1	15.0	27.9	..	17.5	47.9	..	..	6.7	..	..	..	85.0	84.9	S-8
S-9	74.9	0.5	1.3	23.3	50.2	..	11.9	29.1	1.2	..	3.1	..	3.5	1.0	76.7	75.4	S-9
S-10	78.8	2.8	3.2	15.2	10.4	..	23.6	50.2	..	..	14.3	..	..	1.5	84.8	81.6	S-10
S-11	52.7	12.8	1.4	33.1	45.5	..	9.3	0.9	22.8	..	17.3	..	4.2	..	66.9	65.5	S-11
S-12	77.5	1.0	3.2	18.3	34.9	..	1.8	47.8	7.6	..	4.6	..	3.3	..	81.7	78.5	S-12
S-13	78.0	3.6	1.9	16.5	30.8	..	5.6	51.4	5.9	..	3.0	..	2.6	..	83.5	81.6	S-13
S-14	56.6	10.0	1.4	32.0	23.9	6.0	3.2	0.9	0.5	64.6	..	0.9	..	..	68.0	66.6	S-14
S-15	68.4	10.0	6.4	15.2	8.2	..	34.9	22.8	18.4	..	8.1	..	0.6	6.8	84.8	78.4	S-15
S-16	69.1	6.7	5.0	19.2	4.9	0.7	31.2	35.3	19.5	..	1.2	6.4	..	0.5	80.8	75.8	S-16
S-17	68.0	10.1	6.6	15.3	21.7	..	32.9	19.4	13.9	..	0.4	1.3	0.3	10.0	84.7	78.1	S-17
S-18	69.0	6.7	8.3	16.0	23.0	..	35.9	22.0	12.8	..	1.3	..	1.1	3.9	84.0	75.7	S-18
S-19	70.0	7.4	9.2	13.4	14.2	0.9	33.8	19.2	23.8	..	0.9	2.7	..	4.6	86.6	77.4	S-19
S-20	69.1	5.2	9.4	16.3	17.8	..	27.9	13.8	33.2	..	3.9	..	0.3	2.9	83.7	74.3	S-20
Stoker-Fired Units, 400 to 499 Lb. per Sq. In. Gage Pressure																	
S-21	69.4	5.8	7 3	17.5	7.9	..	27.4	23.3	6.8	..	23.6	..	6.3	4.7	82.5	75.2	S-21
S-22	53.8	21.0	1.3	23.9	99.0	..	..	1.0	..	..	..	..	..	..	76.1	74.8	S-22
S-23	50.2	22.1	1.8	25.9	99.5	..	0.5	..	..	..	..	..	..	..	74.1	72.6	S-23
S-24	51.0	23.4	1.3	24.2	100.0	..	..	..	..	..	..	..	..	..	75.8	74.4	S-24
S-25	46.7	22.9	1.3	29.1	100.0	..	..	..	..	..	..	..	..	..	70.9	69.6	S-25
S-26	38.2	4.2	0.6	57.0	0.4	..	58.4	21.0	14.2	..	..	..	0.9	5.1	43.0	42.4	S-26
S-27	74.9	7.0	0.8	17.3	11.3	..	41.3	7.9	32.9	..	4.1	..	..	2.5	82.7	81.9	S-27
S-28	60.4	5.6	0.7	33.3	..	..	48.7	3.5	24.3	..	9.9	..	8.0	14.6	66.7	66.0	S-28
Stoker-Fired Units, 300 to 399 Lb. per Sq. In. Gage Pressure																	
S-29	78.6	5.8	14.5	1.1	No analysis of outage given										98.9	84.4	S-29
S-30	78.3	5.7	16.0	..	No analysis of outage given										..	84.0	S-30
S-31	78.6	5.8	14.9	0.7	No analysis of outage given										99.3	84.4	S-31
S-32	78.6	5.8	14.4	1.2	No analysis of outage given										98.8	84.4	S-32
S-33	78.6	5.8	13.8	1.8	No analysis of outage given										98.2	84.4	S-33
S-34	78.6	5.8	13.9	1.7	No analysis of outage given										98.3	84.4	S-34
S-35	78.8	5.8	14.2	1.2	No analysis of outage given										98.8	84.6	S-35
S-36	55.9	31.4	6.3	6.4	18.7	..	19.8	0.9	48.0	..	12.6	..	..	..	93.6	87.3	S-36
S-37	44.7	23.9	1.4	30.0	39.2	..	10.2	3.6	44.2	..	..	1.8	1.0	..	70.0	68.6	S-37
S-38	46.1	22.6	2.2	29.1	28.6	..	36.3	3.9	30.3	..	..	0.9	..	..	70.9	68.7	S-38
S-39	61.7	11.2	0.9	26.2	27.5	..	0.9	6.1	24.5	..	41.0	..	..	..	73.8	72.9	S-39
S-40	60.5	13.6	1.9	24.0	77.8	3.7	7.5	1.7	6.5	..	2.8	..	..	..	76.0	74.1	S-40
S-41	55.7	13.6	3.1	27.6	62.6	..	6.7	0.3	20.1	..	8.2	..	..	1.7	72.4	69.3	S-41
S-42	63.9	15.5	0.6	20.0	0.5	..	52.6	22.6	16.6	..	..	..	3.8	3.8	80.0	79.4	S-42
S-43	84.4	10.2	2.1	3.3	70.0	..	2.8	27.2	..	..	..	..	..	..	96.7	94.6	S-43
S-44	66.3	0.7	0.8	32.2	..	16.1	3.8	27.8	..	..	0.9	15.8	1.1	..	67.8	67.0	S-44
S-45	68.6	1.0	1.3	29.1	..	18.3	8.4	34.3	..	..	1.6	9.8	0.9	..	70.9	69.6	S-45
S-46	71.5	17.3	5.9	5.3	..	..	1.9	34.7	17.8	..	45.6	..	..	..	94.7	88.8	S-46
S-47	89.1	5.6	1.7	3.6	83.5	..	2.3	14.2	..	..	..	..	..	..	96.4	94.7	S-47
S-48	72.5	2.3	0.5	24.7	15.3	..	30.0	8.7	28.5	..	2.3	11.7	3.5	..	75.3	74.8	S-48
S-49	52.4	13.1	0.8	33.7	44.2	1.6	6.1	2.4	4.6	..	39.2	..	1.6	..	66.3	65.5	S-49
S-50	94.1	0.3	1.9	3.7	51.7	..	11.9	34.6	..	..	1.8	..	..	..	96.3	94.4	S-50
S-51	84.3	10.1	2.3	3.3	64.9	..	7.6	27.5	..	..	..	..	..	..	96.7	94.4	S-51
S-52	75.4	9.3	0.7	14.6	1.9	..	43.1	29.6	23.2	..	..	1.6	0.6	..	85.4	84.7	S-52
S-53	42.2	32.4	11.1	14.3	63.0	..	6.3	16.3	14.4	..	..	..	..	..	85.7	74.6	S-53
S-54	77.4	9.9	1.6	11.1	1.3	..	42.1	25.4	22.2	..	..	..	1.6	7.4	88.9	87.3	S-54
S-55	80.2	9.5	0.5	9.8	3.5	..	50.3	15.8	26.2	..	..	0.7	3.5	..	90.2	89.7	S-55
S-56	79.5	8.5	0.2	11.8	6.7	..	50.0	17.3	20.2	..	..	0.8	2.7	2.3	88.2	88.0	S-56
S-27	74.6	10.3	0.6	14.5	6.4	..	41.8	36.8	14.1	..	..	..	0.9	..	85.5	84.9	S-57
S-58	75.0	10.3	0.4	14.3	3.7	0.2	48.0	21.5	22.1	..	..	1.3	1.3	1.9	85.7	85.3	S-58
S-59	70.5	12.2	0.8	16.5	6.9	..	36.2	35.7	18.2	..	..	0.7	2.3	..	83.5	82.7	S-59
S-60	70.6	11.4	1.0	17.0	..	..	42.0	28.6	24.6	..	..	..	..	4.8	83.0	82.0	S-60
S-61	75.1	11.2	0.5	13.2	6.5	..	41.0	25.9	23.1	..	..	..	1.4	2.1	86.8	86.3	S-61
S-62	75.3	8.4	0.8	15.5	2.9	..	37.2	21.8	16.9	..	..	..	0.6	0.6	84.5	83.7	S-62
S-63	69.7	11.5	2.7	16.1	4.0	..	45.2	27.0	17.3	..	..	..	1.3	5.2	83.9	81.2	S-63
S-64	66.0	23.9	5.2	4.9	36.4	..	36.4	27.2	..	..	..	..	..	..	95.1	89.9	S-64
S-65	65.0	6.2	0.4	28.4	88.0	..	6.7	1.8	3.3	..	..	0.1	0.1	..	71.6	71.2	S-65
S-66	69.6	5.6	3.3	21.5	32.5	..	38.9	28.6	..	..	..	..	..	..	78.5	75.2	S-66

S-67	65.3	5.6	3.1	26.0	99.7	..	..	0.3	..	..	..	..	..	..	74.0	70.9	S-67
S-68	79.1	9.9	0.9	10.1	95.2	..	2.4	..	2.4	..	..	..	..	..	89.9	89.0	S-68
S-69	61.0	21.8	5.4	11.8	82.6	..	10.7	4.8	1.9	..	..	..	..	..	88.2	82.8	S-69
S-70	61.5	0.6	12.2	25.7	8.2	3.7	28.4	24.8	30.7	..	..	2.1	2.1	..	74.3	62.1	S-70
S-71	62.5	0.3	9.5	27.7	17.0	..	32.4	25.7	8.6	..	0.5	1.9	..	13.9	82.3	62.8	S-71
S-72	19.6	13.6	34.0	32.8	75.2	..	5.9	5.9	10.6	..	..	2.4	..	..	67.2	33.2	S-72
S-73	69.0	4.5	6.3	20.2	12.1	..	41.2	17.9	27.6	..	0.6	..	0.6	..	79.8	73.5	S-73
S-74	88.6	1.2	..	10.2	25.0	..	30.3	11.7	21.2	..	..	5.3	6.5	..	89.8	89.8	S-74
S-75	24.8	13.9	25.7	35.6	4.2	..	18.2	30.0	36.1	..	..	6.1	3.7	1.7	64.4	38.7	S-75
S-76	30.0	15.5	20.3	34.2	10.3	..	18.5	40.8	19.5	..	..	10.9	..	..	65.8	45.5	S-76
S-77	93.2	..	1.2	5.6	38.2	..	21.3	6.1	32.8	..	..	..	16.0	..	94.4	93.2	S-77
S-78	91.0	..	1.2	7.8	30.2	..	34.2	20.3	15.3	..	..	..	..	..	92.2	91.0	S-78
S-79	90.5	..	..	9.5	57.1	..	9.1	10.2	18.8	..	..	2.5	2.3	..	90.5	90.5	S-79
S-80	38.1	8.3	28.6	25.0	19.0	2.1	43.2	6.1	29.6	..	..	..	..	..	75.0	46.4	S-80
S-81	55.7	14.8	12.3	17.2	55.0	..	24.2	3.8	14.8	..	..	..	..	2.2	82.8	70.5	S-81
Stoker-Fired Units, 200 to 299 Lb. per Sq. In. Gage Pressure																	
S-82	38.6	43.3	2.5	15.6	81.2	..	5.2	8.8	..	..	..	3.7	1.1	..	84.4	81.9	S-82
S-83	38.1	44.6	1.8	15.5	..	..	13.7	20.3	27.4	..	11.2	13.7	13.7	..	84.5	82.7	S-83
S-84	30.6	35.6	25.4	8.4	..	..	38.7	2.2	59.1	..	..	..	..	..	91.6	66.2	S-84
S-85	45.2	50.5	2.3	2.0	..	..	18.8	7.1	58.0	..	..	16.1	..	..	98.0	95.7	S-85
S-86	77.0	15.3	1.4	6.3	36.2	..	12.7	17.3	9.7	..	..	..	24.1	..	93.7	92.3	S-86
S-87	76.5	10.8	0.5	12.2	13.2	..	17.0	30.0	25.5	..	..	..	11.7	2.6	87.8	87.3	S-87
S-88	50.6	32.1	2.5	14.8	91.5	..	5.6	0.8	..	..	..	2.1	..	..	85.2	82.7	S-88
S-89	79.3	3.8	12.4	4.5	..	..	47.9	11.6	..	..	40.5	..	..	..	95.5	83.1	S-89
S-90	33.1	30.2	1.7	35.0	88.6	..	4.3	..	7.1	..	..	..	..	..	65.0	63.3	S-90
S-91	46.4	47.4	1.7	4.5	47.0	..	12.9	12.1	28.0	..	..	..	..	..	95.5	93.8	S-91
S-92	66.2	14.5	1.9	17.4	24.2	..	16.8	17.8	18.9	..	0.8	..	8.6	12.9	82.6	80.7	S-92
S-93	72.8	12.4	0.9	13.9	32.6	..	18.4	24.2	11.4	..	..	2.3	9.6	1.5	86.1	85.2	S-93
S-94	41.7	27.2	1.3	29.8	78.6	..	1.0	..	20.4	..	..	..	..	..	70.2	68.9	S-94
S-95	57.4	19.5	3.8	19.3	.....				No analysis of outage.						80.7	76.9	S-95
S-96	56.8	13.9	14.9	14.4	.....				No analysis of outage.						85.6	70.7	S-96
S-97	43.7	6.4	42.6	7.3	.....				No analysis of outage.						92.7	50.1	S-97
S-98	58.6	20.3	10.7	10.4	.....				No analysis of outage.						89.6	78.9	S-98
S-99	42.7	19.6	4.4	33.3	65.0	..	8.3	..	..	..	26.7	..	..	..	66.7	62.3	S-99
S-100	52.3	6.7	24.4	16.6	69.0	..	19.4	..	3.3	..	5.0	..	..	3.3	83.4	59.0	S-100
S-101	66.9	9.9	2.8	20.4	97.4	..	..	..	2.6	..	..	..	..	..	79.6	76.8	S-101
Pulverized-Fuel-Fired Units, 400 to 499 Lb. per Sq. In. Gage Pressure																	
P-1	68.5	13.9	2.2	15.4	5.0	..	27.8	45.7	..	16.2	..	5.3	..	..	84.6	82.4	P-1
P-2	65.7	..	22.1	12.2	..	..	11.5	81.4	..	7.1	..	..	..	..	87.8	65.7	P-2
P-3	58.0	14.9	7.7	19.4	9.9	7.8	62.2	9.4	..	6.6	..	4.1	..	..	80.6	72.9	P-3
P-4	61.5	17.2	29.1	12.2	8.9	4.8	21.7	42.4	..	18.5	..	..	3.7	..	87.6	78.7	P-4
P-5	68.1	11.9	4.8	15.2	68.0	..	10.7	18.0	..	..	..	3.3	..	..	84.8	80.0	P-5
P-6	68.0	..	13.2	18.8	..	..	47.2	52.8	..	..	..	..	..	..	81.2	68.0	P-6
P-7	61.2	18.1	6.1	14.6	31.7	3.1	47.9	4.8	..	7.6	1.9	1.9	1.1	..	85.4	79.3	P-7
P-8	59.0	0.9	15.9	24.3	11.6	..	44.0	38.1	..	..	2.8	3.5	..	..	75.7	59.9	P-8
P-9	82.8	1.1	1.0	15.1	14.2	..	51.4	..	..	4.6	1.5	27.6	0.7	..	84.9	83.9	P-9
P-10	71.0	16.7	3.9	8.4	57.0	..	8.8	16.0	..	..	52.0	..	..	13.0	91.6	87.7	P-10
P-11	72.0	0.3	1.0	26.7	..	..	82.0	17.0	..	..	..	..	1.0	..	73.3	72.3	P-11
P-12	53.0	10.0	3.8	33.2	10.1	65.0	..	20.8	..	..	4.1	..	..	..	66.8	63.0	P-12
P-13	73.0	15.0	6.9	5.1	5.2	..	12.5	60.3	..	..	..	..	..	22.0	94.9	88.0	P-13
P-14	75.6	7.2	4.6	12.6	46.5	1.7	7.8	4.6	..	..	..	..	37.2	2.2	87.4	82.8	P-14
P-15	70.2	10.2	6.2	13.4	27.9	..	39.2	15.7	..	8.2	..	3.7	2.8	2.5	86.6	80.4	P-15
P-16	68.9	8.4	6.5	16.2	25.4	..	54.5	10.3	..	2.0	..	6.0	1.8	..	83.8	77.3	P-16
P-17	75.9	12.2	5.2	6.7	61.2	..	32.0	..	..	..	..	6.8	..	..	93.3	88.1	P-17
P-18	36.5	12.6	38.5	12.4	19.6	27.8	2.9	..	..	49.7	..	..	..	..	87.6	49.1	P-18
P-19	55.1	16.9	25.3	2.7	11.2	77.6	..	..	..	11.2	..	..	..	..	97.3	82.0	P-19
P-20	45.8	15.9	29.0	9.3	0.6	37.1	4.5	..	..	57.8	..	..	..	..	90.7	61.7	P-20
Pulverized-Fuel-Fired Units, 300 to 399 Lb. per Sq. In. Gage Pressure																	
P-21	41.6	..	38.4	20.0	25.6	39.2	0.7	4.5	..	25.8	2.8	..	..	1.4	80.0	41.6	P-21
P-22	39.4	..	38.6	22.0	6.5	63.4	..	1.6	..	11.8	4.8	..	11.9	..	78.0	39.4	P-22
P-23	39.4	12.4	24.3	23.9	2.0	37.0	18.0	22.0	..	..	3.0	..	18.0	..	76.1	51.8	P-23
P-24	62.0	16.5	0.8	20.7	76.1	..	..	6.3	..	..	17.6	..	..	..	79.3	78.5	P-24
P-25	55.7	21.0	8.6	14.7	6.0	49.0	4.0	..	..	..	31.0	..	..	..	85.3	76.7	P-25
P-26	61.6	11.5	6.4	20.5	3.0	53.0	24.0	10.0	..	1.0	8.0	..	1.0	..	79.5	73.1	P-26
P-27	37.3	8.2	16.9	37.6	20.0	28.0	13.0	5.0	..	..	19.0	..	15.0	..	62.4	45.5	P-27
P-28	63.0	9.7	9.9	17.4	6.0	63.0	17.0	..	..	..	11.0	3.0	..	..	82.6	72.7	P-28
P-29	68.8	..	22.5	8.7	69.9	..	7.7	6.8	..	..	..	2.3	13.3	..	91.3	68.8	P-29
P-30	74.5	10.0	0.3	15.2	86.5	..	7.7	..	..	..	5.8	..	..	..	84.8	84.5	P-30
P-31	70.0	..	22.0	8.0	22.2	..	17.0	17.8	..	..	1.7	1.7	1.7	37.9	92.0	70.0	P-31
P-32	75.0	0.2	18.2	6.6	4.2	..	10.4	73.2	..	8.0	..	..	4.2	..	93.4	75.2	P-32
P-33	75.4	0.3	14.1	10.2	13.4	..	45.8	22.2	..	3.9	2.2	3.7	8.8	..	89.8	75.7	P-33
P-34	77.2	0.6	16.8	5.4	2.4	..	9.5	83.0	..	..	..	..	5.1	..	94.6	77.8	P-34
P-35	76.3	0.5	15.9	7.3	8.8	..	16.5	72.8	..	..	..	1.9	..	..	92.7	76.8	P-35
P-36	67.2	7.4	6.7	18.7	0.8	..	37.2	41.4	..	7.1	..	5.6	7.9	..	81.3	74.6	P-36
P-37	67.5	12.3	0.2	20.0	68.4	..	29.3	..	..	..	2.3	..	..	..	80.0	79.8	P-37
P-38	35.6	13.7	35.8	14.9	29.0	31.0	20.0	16.0	..	..	..	2.0	2.0	..	85.1	49.3	P-38
P-39	73.3	3.0	6.8	16.9	12.4	..	37.0	23.8	..	13.2	..	1.5	12.1	..	83.1	76.3	P-39
P-40	64.0	4.3	4.6	27.1	1.6	..	9.8	52.2	..	33.6	0.6	..	..	2.2	72.9	68.3	P-40
P-41	56.2	10.3	4.5	29.0	5.2	..	27.1	60.4	..	3.1	..	4.2	..	..	71.0	66.5	P-41

TABLE 3 ANALYSIS OF TIME UTILIZATION OF INDIVIDUAL STEAM-GENERATING UNITS (*Continued from page 273*)

Unit No.	Percentages of total period: Steaming period, per cent	Banked period, per cent	Reserve period, per cent	Crippled period, per cent	Outages classified as to cause expressed as percentages of total outage: Scheduled cleaning and overhaul, per cent	Changes in equipment, per cent	Boiler and superheater, per cent	Furnace, per cent	Stoker, per cent	Pulverizing equipment and burners, per cent	Economizer air preheater, per cent	Ash equipment, per cent	Draft equipment, per cent	Not chargeable to unit, per cent	Availability factor	Use factor	Unit No.
P-42	82.1	1.8	4.4	11.7	27.3	1.7	7.1	6.5	..	..	45.2	..	12.2	..	88.3	83.9	P-42
P-43	85.6	..	1.5	12.9	79.8	..	11.1	5.9	..	..	0.2	2.5	0.5	..	87.1	85.6	P-43
P-44	38.2	11.0	0.2	50.6	..	..	99.1	..	..	..	0.9	..	..	..	49.4	49.2	P-44
P-45	78.1	..	1.5	20.4	54.4	..	13.8	17.5	..	0.7	7.9	1.9	3.8	..	79.6	78.1	P-45
P-46	69.2	19.7	4.8	6.3	13.0	65.0	..	22.0	..	..	..	..	..	..	93.7	88.9	P-46
P-47	42.3	24.2	28.2	5.3	38.5	..	53.5	..	..	..	8.0	..	..	..	94.7	66.5	P-47
P-48	67.3	8.4	19.5	4.8	45.3	..	54.7	..	..	..	..	..	..	..	95.2	75.7	P-48
P-49	49.0	21.2	22.6	7.2	50.1	..	34.3	15.6	..	..	..	..	..	..	92.8	70.2	P-49
P-50	72.2	8.7	12.1	7.0	46.9	..	53.1	..	..	..	..	..	..	..	93.0	80.9	P-50
P-51	59.9	4.2	25.3	10.6	18.5	..	81.5	..	..	..	..	..	..	..	89.4	64.1	P-51
P-52	52.8	23.2	17.5	6.5	56.4	..	43.6	..	..	..	..	..	..	..	93.4	76.0	P-52
P-53	48.0	1.0	34.0	17.0	..	..	69.0	..	..	..	10.0	5.0	12.0	4.0	83.0	49.0	P-53
P-54	60.0	1.3	13.2	25.5	..	..	30.7	..	..	45.7	20.6	..	2.2	0.8	74.5	61.3	P-54
Pulverized-Fuel-Fired Units, 200 to 299 Lb. per Sq. In. Gage Pressure																	
P-55	32.6	18.5	32.4	16.5	..	53.5	14.3	30.6	..	..	1.6	..	..	..	83.5	51.1	P-55
P-56	26.9	14.0	19.0	40.1	7.0	44.0	..	48.0	..	..	..	..	..	1.0	59.9	40.9	P-56
P-57	88.4	0.3	1.6	9.7	85.5	..	8.4	..	..	..	2.6	..	0.5	3.0	90.3	88.7	P-57
P-58	73.0	17.5	..	9.5	15.5	1.4	46.4	30.3	..	..	6.4	..	..	..	90.5	90.5	P-58
P-59	72.5	15.0	..	12.5	1.5	..	65.3	23.2	..	1.1	8.9	..	..	..	87.5	87.5	P-59
P-60	78.2	11.6	..	10.2	14.0	..	54.5	27.5	..	..	..	..	4.0	..	89.8	89.8	P-60
P-61	81.0	8.5	..	10.5	19.3	..	36.0	2.3	..	1.3	..	..	41.1	..	89.5	89.5	P-61
P-62	88.3	0.6	..	11.1	45.9	1.3	50.0	..	..	..	..	..	2.8	..	88.9	88.9	P-62
Oil- and Gas-Fired Units																	
OG-1	46.4	22.8	21.8	9.0	100.0	..	..	..	..	..	..	..	..	..	91.0	69.2	OG-1
OG-2	86.0	4.1	4.8	5.1	29.7	..	62.5	2.2	..	..	..	..	5.6	..	94.9	90.1	OG-2
OG-3	51.0	21.4	15.8	11.8	65.4	..	30.5	4.1	..	..	..	..	..	..	88.2	72.4	OG-3
OG-4	62.1	12.3	14.0	11.6	62.7	..	23.5	11.9	..	..	1.9	..	..	..	88.4	74.4	OG-4
OG-5	52.9	20.0	16.6	10.5	37.0	54.5	7.9	0.6	..	..	..	..	..	..	89.5	72.9	OG-5
OG-6	50.8	21.8	17.8	9.6	66.5	13.4	3.4	16.7	..	..	..	..	..	..	90.4	72.6	OG-6
OG-7	57.5	0.5	20.2	21.8	16.5	..	78.2	2.3	..	..	0.5	..	..	2.5	78.2	58.0	OG-7
OG-8	62.6	0.5	16.7	20.2	39.6	..	36.4	21.8	..	..	..	..	..	2.2	79.8	63.1	OG-8
OG-9	72.4	2.7	11.4	13.5	50.2	..	24.9	24.9	..	..	..	..	..	..	86.5	75.1	OG-9
OG-10	59.0	29.2	2.8	9.0	100.0	..	..	..	..	..	..	..	..	..	91.0	88.2	OG-10
OG-11	63.2	21.3	5.2	10.3	77.4	22.6	..	..	..	..	..	..	..	..	89.7	84.5	OG-11
OG-12	55.9	30.3	4.1	9.7	100.0	..	..	..	..	..	..	..	..	..	90.3	86.2	OG-12
OG-13	43.8	9.8	4.5	41.9	..	..	22.6	..	..	..	..	..	..	77.4	58.1	53.6	OG-13
OG-14	53.7	3.9	1.8	40.6	29.0	..	15.0	..	..	..	..	..	..	56.0	59.4	57.6	OG-14
OG-15	65.0	27.8	1.8	5.4	100.0	..	..	..	..	..	..	..	..	..	94.6	92.8	OG-15
OG-16	76.0	17.1	0.8	6.1	27.3	..	41.0	31.7	..	..	..	..	..	..	93.9	93.1	OG-16
OG-17	75.4	0.2	4.5	19.9	55.7	..	0.8	43.5	..	..	..	..	..	..	80.1	75.6	OG-17
Miscellaneous Combinations																	
M-1	87.4	2.5	0.1	10.0	64.3	..	15.0	3.3	..	3.4	2.9	..	11.1	..	90.0	89.9	M-1
M-2	90.4	2.7	..	6.9	71.0	..	4.0	5.4	..	14.5	4.0	..	1.1	..	93.1	93.1	M-2
M-3	86.1	2.1	..	11.8	79.4	..	4.4	4.7	..	1.5	5.4	..	4.6	..	88.2	88.2	M-3
M-4	56.4	40.0	0.3	3.3	48.5	..	51.5	..	..	..	..	..	..	..	96.7	96.4	M-4
M-5	65.6	17.7	0.6	16.1	5.4	50.0	7.2	37.4	..	..	..	..	..	..	83.9	83.3	M-5
M-6	26.6	53.4	4.5	15.5	..	..	7.4	27.6	65.0	..	..	..	..	..	84.5	80.0	M-6

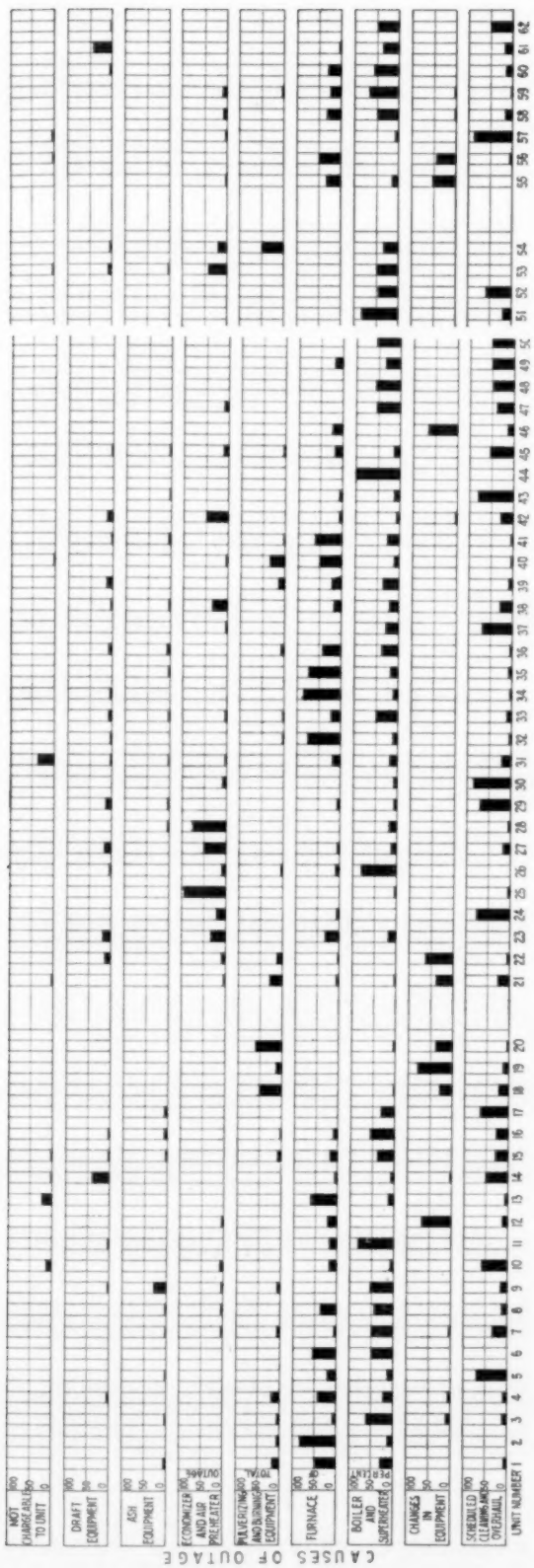


FIG. 6 ANALYSIS OF OUTAGE—PULVERIZED-COAL-FIRED STEAM-GENERATING UNITS

UNITS: 1–20, inc., 400 to 499-lb. class; 21–54, inc., 300 to 399-lb. class; 55–62, inc., 200 to 299-lb. class (Crippled periods classified as to cause and shown as percentages of total outage.)

divided opinion as to when a unit is crippled and when in reserve. The authors have based all their calculations and conclusions on the following definitions which were sent to the field with the log sheets and which the individuals supplying data were requested to observe.

1 A "steaming" unit is one which is passing steam to the line.

2 A "banked" unit is one in which approximate line pressure is maintained with no flow of steam to the line.

3 A "reserve" unit is one which is cold and in which the pressure is near zero, gage, but which may be put on the line in the time it normally takes to lay and light the fire and bring the steam pressure to that of the line.

4 A "crippled" unit is one which is in such a condition that it is incapable of being put on the line within the time it normally takes to bring a boiler from cold to steaming.

In apportioning the reported crippled hours among the various causes, considerable difficulty arose. The authors attempted to get the primary cause of outage in each case and were usually successful in doing so, but many occasions arose in which more than one part of the unit was worked on at one time. For instance, the boiler might be shut down and crippled for repairs to the furnace, but the operators might very naturally take advantage of the opportunity to make extensive repairs to the boiler or stoker. In cases like this the shutdown was charged to the furnace for as long a period as such furnace repairs were reported to extend. Any further crippled time was charged to the secondary job. It was rather difficult to make such distinctions in some cases. However, as the compilation of data was made by a single group of men in the office of the authors, the results should be at least reasonably comparable.

In several cases it was found that a considerable portion of the outage was due to voluntary crippling of the unit for extensive changes in equipment, such as the installation of a new type of wall or burner. In the strictest sense this time should be charged to the unit as outage, because the investment represented in it is idle during such reconstruction and because the actual facts regarding hours available are being determined. Since one might contend, however, that the unit was taken off for reconstruction at a time when it was not needed and could have been kept in service if required, such cases have been shown on the charts both with and without these remodeling periods.

With the modern trend toward complicated design, the old terms, "boiler horsepower" and "per cent of rating" have become less and less descriptive of actual facts. Indeed, there has been a marked tendency throughout the country to discard the old terms and express the performance of steam-generating units in terms of heat delivered per unit of heating surface, furnace volume, or grate surface. The authors, in presenting the results of their study, have used several such methods of expression.

DEFINITIONS AND CONVENTIONS

In addition to the definitions of "Steaming," "Banked," "Reserve," and "Crippled" already given, the following definitions and conventions have been used:

1 The "Total Heat Supplied Water and Steam per Hour" has been calculated as the difference in heat content between economizer inlet and superheater outlet. The result is expressed throughout in thousands of B.t.u. per hour, which unit has been abbreviated as "K.B.t.u. per hour."

2 The "Saturated Heating Surface" has been considered as the surface of the boiling vessel itself as reported by the operator, plus the surface of any hearth screens or water-cooled wall tubes as reported by the operator. Economizer surface is not included in any case.

3 The "Heat Transfer per Square Foot of Saturated Heating Surface per Hour" was obtained by dividing the quantity defined

TABLE 4 INDIVIDUAL HEAT-TRANSFER RATES OF STEAM-GENERATING UNITS

Unit No.	Total steam evaporated, 1000 lb.	Total steaming hours	Average evaporation 1000 lb. per hr.	Saturated steam: Total heat, B.t.u. per lb.	Saturated steam: Heat generated in Boiler, B.t.u. per lb.	Saturated steam: Heat generated in Entire unit, B.t.u. per lb.	Superheated steam: Total heat, B.t.u. per lb.	Superheated steam: Heat generated in Boiler, B.t.u. per lb.	Superheated steam: Heat generated in Entire unit, B.t.u. per lb.	Average evaporation per unit of: Saturated steam surface, lb. per sq. ft. per hr.	Average evaporation per unit of: Furnace volume, lb. per cu ft. per hr.	Average evaporation per unit of: Projected grate area, lb. per sq. ft. per hr.	Average heat output of boiler proper: In saturated steam, K.B.t.u. per sq. ft. sat. surf. per hr.	Average heat output of boiler proper: In superheated steam, K.B.t.u. per sq. ft. sat. surf. per hr.	Average heat output of entire unit represented by: Saturated steam surface, K.B.t.u. per sq. ft. per hr.	Average heat output of entire unit represented by: Superheated steam: Saturated steam surface, K.B.t.u. per sq. ft. per hr.	Average heat output of entire unit represented by: Superheated steam: Furnace volume, K.B.t.u. per cu. ft. per hr.	Average heat output of entire unit represented by: Superheated steam: Projected grate area, K.B.t.u. per per sq. ft. per hr.	Unit No.
Stoker-Fired Units, 500 to 700 Lb. per Sq. In. Gage Pressure																				
S-1	384,540	3585	108	1212	925	1032	1363	1076	1183	7.7	24.6	29.6	7.1	8.3	7.9	9.1	29.0	35.0	S-1
S-2	473,500	4536	104	1212	920	1032	1363	1071	1183	7.4	21.0	28.5	6.8	7.9	7.6	8.7	24.8	33.6	S-2
S-3	404,502	3907	103	1212	928	1032	1363	1079	1183	7.3	20.7	28.2	6.8	7.9	7.5	8.6	24.4	33.2	S-3
S-4	441,360	4270	103	1212	928	1032	1363	1079	1183	7.3	20.7	28.2	6.8	7.9	7.5	8.6	24.4	33.2	S-4
S-5	517,458	5329	97	1212	939	1032	1363	1090	1183	6.9	22.0	26.6	6.5	7.5	7.1	8.1	26.0	31.4	S-5
S-6	494,400	5248	94	1212	905	1032	1363	1086	1183	6.7	21.0	25.8	6.1	7.3	6.9	7.9	25.3	30.2	S-6
S-7	514,382	5739	90	1212	906	1032	1362	1086	1182	6.4	15.6	24.7	5.8	7.7	6.6	7.5	18.9	29.2	S-7
S-8	527,835	6005	88	1212	926	1032	1362	1076	1182	6.3	15.3	24.1	5.9	6.8	6.5	7.4	18.1	28.4	S-8
S-9	474,813	5545	86	1212	917	1032	1362	1067	1182	6.1	14.9	23.6	5.6	6.6	6.3	7.2	17.6	27.8	S-9
S-10	494,277	5829	85	1212	937	1032	1362	1087	1182	6.0	14.7	23.2	5.6	6.5	6.2	7.1	17.3	27.4	S-10
S-11	482,810	4608	105	1211	843	943	1380	1012	1112	6.3	17.5	23.7	5.3	6.4	5.9	7.0	19.4	26.3	S-11
S-12	467,104	5742	82	1212	926	1032	1362	1076	1182	5.8	14.2	22.4	5.4	6.3	6.0	6.8	16.8	26.4	S-12
S-13	469,163	5770	81	1212	910	1032	1362	1090	1182	5.8	14.0	22.2	5.3	6.3	6.0	6.8	16.5	26.2	S-13
S-14	554,446	4945	112	1211	853	943	1380	1022	1112	6.0	19.5	25.2	5.1	6.1	5.6	6.7	21.6	27.0	S-14
S-15	578,589	5967	97	1213	989	1045	1360	1136	1192	5.4	12.5	18.4	5.7	6.2	5.7	6.4	14.9	21.9	S-15
S-16	578,771	6038	96	1213	989	1045	1360	1136	1192	5.4	12.3	18.2	5.7	6.2	5.7	6.4	14.6	21.6	S-16
S-17	560,500	5937	94	1213	993	1045	1360	1140	1192	5.2	12.4	17.8	5.5	5.9	5.5	6.2	14.4	21.2	S-17
S-18	563,256	6016	94	1213	984	1045	1360	1131	1192	5.2	12.1	17.8	5.5	5.9	5.5	6.2	14.4	21.2	S-18
S-19	545,789	6118	89	1213	990	1045	1360	1137	1192	5.1	10.5	16.8	5.4	5.8	5.4	6.1	12.5	20.0	S-19
S-20	549,801	6028	91	1213	990	1045	1360	1137	1192	5.0	11.2	17.2	5.3	5.7	5.3	6.0	13.9	20.4	S-20
Stoker-Fired Units, 400 to 499 Lb. per Sq. In. Gage Pressure																				
S-21	660,647	6058	109	1207	941	1039	1365	1099	1197	6.7	14.0	32.4	6.3	7.4	7.0	8.0	16.8	38.9	S-21
S-22	333,067	4344	77	1208	919	1028	1360	1061	1025	7.4	14.3	28.6	6.8	6.8	7.6	7.6	14.7	29.4	S-22
S-23	296,011	4035	74	1208	935	1028	1360	1061	1025	7.1	13.7	27.4	6.7	6.6	7.3	7.3	14.1	28.2	S-23
S-24	301,719	4117	73	1208	942	1028	1360	1061	1025	7.1	13.5	27.0	6.6	6.6	7.2	7.2	13.9	27.8	S-24
S-25	277,906	3855	72	1208	929	1028	1360	1061	1025	6.9	13.3	26.6	6.4	6.4	7.1	7.1	13.7	27.4	S-25
S-26	328,094	2212	148	1207	909	909	1372	1074	1074	5.2	17.4	29.6	4.7	5.5	5.5	5.5	18.7	31.8	S-26
S-27	597,935	4284	139	1207	914	914	1372	1079	1079	4.9	16.3	27.8	4.5	5.3	5.3	5.3	17.5	29.9	S-27
S-28	477,589	3523	135	1207	916	916	1372	1081	1081	4.8	15.8	27.0	4.4	5.2	5.2	5.2	17.0	29.0	S-28
Stoker-Fired Units, 300 to 399 Lb. per Sq. In. Gage Pressure																				
S-29	967,330	6909	140	1206	898	1003	1364	1056	1161	8.5	20.8	54.5	7.6	9.0	8.8	9.9	24.2	63.2	S-29
S-30	977,244	6882	140	1206	898	1003	1364	1056	1161	8.5	20.8	54.5	7.6	9.0	8.8	9.9	24.2	63.2	S-30
S-31	967,330	6909	140	1206	898	1003	1364	1056	1161	8.5	20.8	54.5	7.6	9.0	8.8	9.9	24.2	63.2	S-31
S-32	967,330	6909	140	1206	898	1003	1364	1056	1161	8.5	20.8	54.5	7.6	9.0	8.8	9.9	24.2	63.2	S-32
S-33	967,330	6909	140	1206	898	1003	1364	1056	1161	8.5	20.8	54.5	7.6	9.0	8.8	9.9	24.2	63.2	S-33
S-34	967,330	6909	140	1206	898	1003	1364	1056	1161	8.5	20.8	54.5	7.6	9.0	8.8	9.9	24.2	63.2	S-34
S-35	967,330	6909	140	1206	898	1003	1364	1056	1161	8.5	20.8	54.5	7.6	9.0	8.8	9.9	24.2	63.2	S-35
S-36	571,807	4871	120	1206	876	988	1325	995	1107	8.2	29.6	32.4	7.2	8.2	8.1	9.1	32.8	36.0	S-36
S-37	320,717	3869	83	1208	912	1028	1358	1062	1178	7.4	16.0	25.6	6.7	7.9	7.6	8.7	18.9	30.2	S-37
S-38	358,573	4111	82	1208	926	1028	1358	1076	1178	7.3	15.8	25.3	6.8	7.9	7.5	8.6	18.6	29.8	S-38
S-39	551,191	5260	105	1206	908	1125	1338	1040	1157	7.3	21.9	25.6	6.6	7.6	8.2	8.5	25.4	29.7	S-39
S-40	467,962	5279	89	1208	952	1028	1358	1106	1182	6.9	16.2	23.1	6.5	7.7	7.1	8.1	19.0	27.3	S-40
S-41	411,760	4867	85	1208	955	1028	1358	1109	1182	6.8	15.5	22.0	6.4	7.5	7.0	8.0	18.3	26.0	S-41
S-42	764,928	4224	180	1208	916	1028	1373	1081	1081	7.3	20.0	45.7	6.6	7.9	6.6	7.9	23.4	49.2	S-42
S-43	897,788	7372	85	1206	914	1026	1365	1073	1185	7.0	15.7	46.6	5.9	7.0	6.7	7.7	18.5	54.6	S-43
S-44	443,813	5688	78	1207	920	1027	1378	1091	1198	6.8	11.4	21.6	5.7	6.8	6.4	7.4	13.7	25.9	S-44
S-45	455,955	5890	78	1207	920	1027	1378	1091	1198	6.7	11.4	21.6	5.6	6.7	6.3	7.3	13.7	25.9	S-45
S-46	725,741	6233	118	1207	906	1019	1362	1061	1174	6.5	13.1	31.6	5.5	6.5	6.2	7.1	15.3	37.0	S-46
S-47	629,638	7778	81	1206	921	1026	1365	1080	1185	6.5	15.0	44.5	5.5	6.5	6.2	7.1	17.8	52.7	S-47
S-48	473,318	6217	76	1207	922	1027	1378	1093	1198	6.5	11.1	21.0	5.4	6.5	6.1	7.1	13.3	25.2	S-48
S-49	357,998	4564	78	1208	952	1028	1358	1106	1182	6.5	14.2	20.0	5.6	6.5	6.1	7.0	16.8	23.8	S-49
S-50	624,828	8216	109	1206	913	1026	1365	1086	1199	6.4	13.0	22.1	5.4	6.4	6.1	7.0	15.6	26.5	S-50
S-51	578,052	7371	79	1206	913	1026	1365	1086	1199	6.4	19.0	43.5	5.4	6.4	6.1	7.0	15.6	26.5	S-51
S-52	858,883	6576	130	1208	918	918	1373	1083	1083	6.2	13.7	37.7	5.2	6.2	5.2	6.2	14.8	40.7	S-52
S-53	244,926	3666	67	1206	973	988	1325	1092	1107	6.0	17.6	27.9	5.3	6.0	5.4	6.1	19.5	31.0	S-53
S-54	870,154	6751	129	1208	917	917	1373	1082	1082	5.9	13.6	37.5	5.0	5.9	5.0	5.9	14.7	40.5	S-54
S-55	865,269	6969	125	1208	915	915	1373	1080	1080	5.9	13.1	36.2	5.1	5.9	5.1	5.9	14.2	39.1	S-55
S-56	901,864	6917	130	1208	895	895	1373	1060	1060	5.8	13.7	37.7	4.9	5.8	4.9	5.8	14.5	40.0	S-56
S-57	805,083	6515	124	1208	917	917	1373	1082	1082	5.7	13.0	36.0	4.9	5.7	4.9	5.7	14.1	39.9	S-57
S-58	820,196	6546	125	1208	917	917	1373	1082	1082	5.7	13.9	31.7	4.9	5.7	4.9	5.7	15.0	34.2	S-58
S-59	778,429	6164	126	1208	919	919	1373	1084	1084	5.7	14.0	32.0	4.9	5.7	4.9	5.7	15.1	34.6	S-59
S-60	769,505	6166	124	1208	918	918	1373	1083	1083	5.7	13.8	31.5	4.9	5.7	4.9	5.7	14.9	34.0	S-60
S-61	814,961	6551	124	1208	918	918	1373	1083	1083	5.7	13.8	31.5	4.9	5.7	4.9	5.7	14.9	34.0	S-61
S-62	801,721	6561	122	1208	911	911	1373	1082	1082	5.7	12.8	35.4	4.8	5.7	4.8	5.7	13.8	38.2	S-62
S-63	734,849	6218	118	1208	917	917	1373	1082	1082	5.7	13.2	30.0	4.9	5.7	4.9	5.7	14.2	32.4	S-63

S-64	203,888	2330	86	1206	1009	1009	1336	1139	1139	4.9	21.5	43.0	4.9	5.6	4.9	5.6	24.5	49.0	S-64
S-65	418,595	5557	75	1205	995	995	1312	1102	1102	5.0	10.9	26.8	5.0	5.5	5.0	5.5	12.0	29.5	S-65
S-66	435,120	6041	72	1205	997	997	1312	1104	1104	4.8	10.4	25.7	4.8	5.3	4.8	5.3	11.4	28.3	S-66
S-67	405,307	5603	72	1205	993	993	1312	1100	1100	4.8	10.4	25.7	4.7	5.3	4.7	5.3	11.4	28.3	S-67
S-68	440,605	6508	68	1205	995	995	1312	1129	1129	4.5	9.9	24.2	4.5	5.1	4.5	5.1	11.2	27.4	S-68
S-69	187,684	2154	83	1206	908	908	1336	1038	1038	4.8	20.7	41.5	4.4	5.0	4.4	5.0	21.5	43.2	S-69
S-70	394,099	5371	73	1207	984	984	1362	1139	1139	4.2	15.6	15.1	4.1	4.8	4.1	4.8	17.8	17.2	S-70
S-71	397,823	5452	73	1207	971	971	1362	1126	1126	4.2	15.6	15.1	4.1	4.8	4.1	4.8	17.5	17.0	S-71
S-72	81,463	1686	49	1206	1003	1026	1378	1161	1184	4.1	7.7	18.0	4.1	4.8	4.2	4.8	9.1	21.2	S-72
S-73	434,812	6170	71	1207	986	986	1362	1141	1141	4.1	15.2	14.7	4.1	4.7	4.1	4.7	17.3	16.8	S-73
S-74	218,041	5209	42	1205	967	967	1340	1102	1102	4.2	14.5	23.4	4.1	4.6	4.1	4.6	16.0	25.8	S-74
S-75	104,822	2113	48	1206	1003	1026	1378	1161	1184	4.0	7.6	15.9	4.0	4.2	4.1	4.3	9.0	18.8	S-75
S-76	106,445	2464	43	1206	1002	1026	1378	1160	1184	3.6	6.8	15.8	3.6	4.2	3.7	4.3	8.0	18.6	S-76
S-77	201,490	5494	37	1205	968	968	1340	1103	1103	3.7	12.8	20.6	3.7	4.1	3.7	4.1	14.1	22.6	S-77
S-78	200,255	5350	37	1205	968	968	1340	1103	1103	3.7	12.8	20.6	3.7	4.1	3.7	4.1	14.1	22.6	S-78
S-79	190,221	5321	36	1205	968	968	1340	1103	1103	3.7	12.5	20.0	3.6	4.0	3.6	4.0	13.8	22.0	S-79
S-80	202,355	3384	59	1205	997	997	1367	1159	1159	3.1	11.1	22.8	3.1	3.6	3.1	3.6	12.9	26.4	S-80
S-81	273,134	4965	55	1205	998	998	1367	1160	1160	2.9	10.8	21.4	2.9	3.4	2.9	3.4	12.5	24.8	S-81
Stoker-Fired Units, 200 to 299 Lb. per Sq. In. Gage Pressure																			
S-82	198,102	2202	90	1204	936	1031	1395	1127	1222	7.2	21.2	..	6.8	8.1	7.4	8.8	25.8	..	S-82
S-83	194,325	2177	89	1204	931	1031	1395	1122	1222	7.1	21.0	..	6.6	7.9	7.3	8.7	25.6	..	S-83
S-84	272,815	2687	101	1204	1003	1068	1314	1113	1178	7.3	24.4	35.6	7.3	8.1	7.8	8.5	28.8	42.0	S-84
S-85	222,214	2575	86	1204	924	1031	1395	1115	1222	6.9	20.2	..	6.4	7.7	7.1	8.4	24.6	..	S-85
S-86	467,384	5829	80	1204	956	1049	1341	1093	1186	7.0	16.3	25.9	6.7	7.6	7.4	8.3	19.2	30.6	S-86
S-87	455,536	5754	80	1204	956	1049	1341	1093	1186	7.0	16.3	25.9	6.7	7.6	7.4	8.3	19.2	30.6	S-87
S-88	238,282	2888	83	1204	924	1031	1395	1115	1222	6.6	19.5	..	6.1	7.4	6.8	8.1	23.8	..	S-88
S-89	561,202	6928	81	1204	1003	1068	1314	1113	1178	6.4	18.2	30.6	6.4	7.1	6.9	7.5	31.5	36.2	S-89
S-90	138,656	1891	74	1204	936	1031	1395	1127	1222	6.1	..	..	5.7	6.9	6.3	7.5	..	..	S-90
S-91	196,043	2651	74	1204	949	1031	1395	1140	1222	5.9	17.4	..	5.6	6.7	6.1	7.2	21.2	..	S-91
S-92	343,270	5008	69	1204	956	1052	1341	1093	1189	6.0	14.1	22.3	5.7	6.5	6.3	7.1	16.5	26.6	S-92
S-93	358,537	5506	65	1204	956	1049	1341	1093	1186	5.7	13.3	21.0	5.5	6.2	6.0	6.8	15.8	25.0	S-93
S-94	250,833	3414	74	1203	1060	1060	1314	1171	1171	5.0	11.1	23.9	5.3	5.9	5.3	5.9	13.0	28.0	S-94
S-95	657,876	5004	131	1204	945	1022	1367	1108	1185	4.7	12.2	34.8	4.5	5.2	4.8	5.6	14.5	41.5	S-95
S-96	679,895	5077	134	1204	952	1022	1367	1115	1185	4.6	11.6	28.3	4.4	5.1	4.7	5.5	13.8	33.7	S-96
S-97	485,554	3724	131	1204	951	1022	1367	1114	1185	4.5	11.4	27.6	4.3	5.0	4.6	5.4	13.6	32.8	S-97
S-98	609,225	5114	119	1204	947	1022	1367	1110	1185	4.2	11.0	31.6	4.0	4.7	4.3	5.0	13.1	37.6	S-98
S-99	223,531	3732	60	1203	1060	1060	1314	1171	1171	4.1	9.0	19.7	4.3	4.8	4.3	4.8	10.5	23.1	S-99
S 100	259,864	4566	57	1203	955	1064	1325	1077	1186	4.0	7.9	18.4	3.8	4.3	4.2	4.7	9.4	21.9	S-100
S-101	286,515	5838	49	1203	955	1060	1314	1066	1171	3.4	7.4	15.8	3.3	3.6	3.6	4.0	8.6	18.6	S-101
Pulverized-Fuel-Fired Units, 400 to 499 Lb. per Sq. In. Gage Pressure																			
P-1	721,770	5983	121	1208	920	1020	1362	1074	1174	8.7	8.5	..	8.0	9.4	8.9	10.5	9.9	..	P-1
P-2	872,740	5147	170	1208	915	915	1389	1096	1096	9.0	11.9	..	8.2	9.9	8.2	9.9	13.1	..	P-2
P-3	595,710	5067	117	1208	920	1020	1362	1074	1174	8.4	8.2	..	7.6	9.0	8.6	9.9	9.6	..	P-3
P-4	592,690	5353	110	1208	920	1020	1362	1074	1174	7.9	7.7	..	7.3	8.5	8.1	9.3	9.0	..	P-4
P-5	997,758	5068	197	1208	910	1020	1360	1062	1172	7.8	7.9	..	7.1	8.3	8.0	9.2	9.2	..	P-5
P-6	916,224	5946	154	1208	915	915	1389	1096	1096	8.1	10.8	..	7.4	8.9	7.4	8.9	11.9	..	P-6
P-7	571,985	5345	107	1208	920	1020	1362	1074	1174	7.2	7.5	..	6.6	7.7	7.3	8.5	8.8	..	P-7
P-8	721,338	5024	144	1208	915	915	1389	1096	1096	7.5	10.1	..	6.9	8.2	6.9	8.2	11.1	..	P-8
P-9	1,190,890	6680	178	1208	913	913	1369	1074	1074	7.4	9.9	..	6.8	8.0	6.8	8.0	10.6	..	P-9
P-10	1,025,450	5258	195	1208	910	1020	1360	1062	1172	6.4	7.8	..	5.8	6.8	6.5	7.5	9.1	..	P-10
P-11	924,873	5564	166	1208	913	913	1369	1074	1074	6.8	9.2	..	6.2	7.3	6.2	7.3	9.8	..	P-11
P-12	735,056	3918	188	1208	910	1020	1360	1062	1172	6.2	7.5	..	5.6	6.6	6.3	7.3	8.8	..	P-12
P-13	998,240	5386	185	1208	910	1020	1360	1062	1172	6.1	7.4	..	5.5	6.5	6.2	7.2	8.7	..	P-13
P-14	846,699	6605	128	1208	905	905	1362	1059	1059	6.0	6.9	..	5.4	6.4	5.4	6.4	7.3	..	P-14
P-15	770,400	6130	126	1208	905	905	1362	1059	1059	5.9	6.8	..	5.3	6.3	5.3	6.3	7.2	..	P-15
P-16	755,201	6044	125	1208	905	905	1362	1059	1059	5.9	6.8	..	5.3	6.3	5.3	6.3	7.2	..	P-16
P-17	815,610	6615	123	1208	905	905	1362	1059	1059	5.8	6.7	..	5.3	6.3	5.3	6.3	7.1	..	P-17
P-18	341,522	3439	99	1208	940	940	1362	1094	1094	5.8	12.0	..	5.3	6.3	5.3	6.3	13.1	..	P-18
P-19	468,063	5097	92	1208	940	940	1362	1094	1094	5.6	11.2	..	5.3	6.1	5.3	6.1	12.2	..	P-19
P-20	397,040	3845	100	1208	940	940	1362	1094	1094	5.4	12.1	..	5.1	5.9	5.1	5.9	13.2	..	P-20
Pulverized-Fuel-Fired Units, 300 to 399 Lb. per Sq. In. Gage Pressure																			
P-21	515,928	3616	143	1207	909	1027	1357	1059	1195	17.0	11.6	..	15.5	18.0	17.5	19.5	13.3	..	P-21
P-22	335,645	3440	98	1207	909	1027	1357	1059	1145	9.8	19.4	..	8.9	10.4	10.1	11.2	22.2	..	P-22
P-23	246,939	2899	85	1208	915	1015	1362	1069	1169	7.8	6.6	..	7.1	8.4	7.9	9.1	7.7	..	P-23
P-24	260,459	3121	83	1207	994	1049	1328	1115	1170	7.5	10.1	..	7.4	8.4	7.9	8.8	11.8	..	P-24
P-25	329,546	4104	80	1208	915	1015	1362	1069	1169	7.3	6.3	..	6.7	7.8	7.4	8.5	7.4	..	P-25
P-26	362,134	4579	79	1208	915	1015	1362	1069	1169	7.3	6.2	..	6.7	7.8	7.4	8.5	7.2	..	P-26
P-27	223,971	2791	80	1208	915	1015	1362	1069	1169	7.3	6.3	..	6.7	7.8	7.4	8.5	7.4	..	P-27
P-28	353,502	4593	78	1208	915	1015	1362	1069	1169	7.2	6.1	..	6.6	7.7	7.3	8.4	7.1	..	P-28
P-29	780,807	6003	130	1207	1027	1027	1342	1162	1162	7.2	11.8	..	7.4	8.4	7.4	8.4	13.7	..	P-29
P-30	299,085	3758	80	1207	994	1049	1328	1115	1170	7.2	8.9	..	7.1	8.0	7.6	8.4	11.5	..	P-30
P-31	684,362	6106	112	1207	927	1027	1342	1062	1162	7.1	10.2	..	6.6	7.6	7.3	8.2	11.9	..	P-31
P-32	686,044	6554	105	1207	927	1027	1342	1062	1162	6.9	9.6	..	6.4	7.3	7.1	8.0	11.2	..	P-32
P-33	681,873	6564	104	1207	927	1027	1342	1062	1162	6.8	9.5	..	6.3	7.2	7.0	7.9	11.0	..	P-33
P-34	698,676	6746	104	1207	927	1027	1342	1062	1162	6.8	9.5	..	6.3	7.2	7.0	7.9	11.0	..	P-34
P-35	695,934	6661	104	1207	927	1027	1342	1062	1162	6.8	9.5	..	6.3	7.2	7.0	7.9	11.0	..	P-35
P-36	894,422	5421	165	1208	912	912	1367	1071	1071	7.3	12.2	..	6.7	7.8	6.7	7.8	13.0	..	P-36
P-37	247,516	3367	74	1207	994	1049	1328	1115	1170	6.7	9.1	..	6.6	7.5	7.0	7.8	10.6	..	P-37
P-38	292,388	3877	75	1208	915	1015	1362	1069	1169	6.7	5.9	..	6.1	7.1	6.8	7.8	6.9	..	P-38

TABLE 4 INDIVIDUAL HEAT-TRANSFER RATES OF STEAM-GENERATING UNITS (*Continued from page 277*)

Unit No.	Total steam evaporated, 1000 lb.	Total steaming hours	Average evaporation 1000 lb. per hr.	Saturated steam: Total heat, B.t.u. per lb.	Saturated steam: Heat generated in Boiler, B.t.u. per lb.	Saturated steam: Heat generated in Entire unit, B.t.u. per lb.	Superheated steam: Total heat, B.t.u. per lb.	Superheated steam: Heat generated in Boiler, B.t.u. per lb.	Superheated steam: Heat generated in Entire unit, B.t.u. per lb.	Average evaporation per unit of Saturated steam surface, lb. per sq. ft. per hr.	Average evaporation per unit of Furnace volume, lb. per cu. ft. per hr.	Average evaporation per unit of Projected grate area, lb. per sq. ft. per hr.	Average heat output of boiler proper: In saturated steam, K.B.t.u. per sq. ft. sat. surf. per hr.	Average heat output of boiler proper: In superheated steam, K.B.t.u. per sq. ft. sat. surf. per hr.	Average heat output of entire unit represented by Superheated steam: Saturated steam surface, K.B.t.u. per sq. ft. per hr.	Average heat output of entire unit represented by Superheated steam: Saturated steam surface, K.B.t.u. per sq. ft. per hr.	Average heat output of entire unit represented by Superheated steam: Furnace volume, K.B.t.u. per cu ft. per hr.	Average heat output of entire unit represented by Superheated steam: Projected grate area, K.B.t.u. per sq. ft. per hr.	Unit No.
P-39	979,462	6098	160	1208	912	912	1367	1071	1071	7.1	11.9	..	6.5	7.6	6.5	7.6	12.7	..	P-39
P-40	822,484	5152	159	1208	912	912	1367	1071	1071	7.1	11.8	..	6.5	7.6	6.5	7.6	12.6	..	P-40
P-41	714,880	4545	157	1208	912	912	1367	1071	1071	7.0	11.6	..	6.4	7.5	6.4	7.5	12.4	..	P-41
P-42	1,436,626	7136	201	1205	907	907	1353	1055	1055	6.7	13.5	..	6.1	7.1	6.1	7.1	14.9	..	P-42
P-43	442,537	5049	87	1205	907	967	1367	1069	1129	6.2	16.5	..	5.6	6.6	6.0	7.0	18.6	..	P-43
P-44	129,433	1921	67	1207	994	1049	1328	1115	1170	6.0	8.2	..	5.9	6.7	6.3	7.0	9.7	..	P-44
P-45	385,093	4596	84	1205	907	967	1367	1069	1129	6.0	16.0	..	5.4	6.4	5.8	6.8	18.1	..	P-45
P-46	620,970	6084	102	1205	907	1010	1317	1019	1122	5.6	8.5	..	5.1	5.7	5.7	6.7	9.5	..	P-46
P-47	588,174	3483	169	1208	920	1020	1376	1088	1188	5.4	5.5	..	5.0	5.9	5.5	6.4	6.5	..	P-47
P-48	847,005	5094	166	1208	920	1020	1376	1088	1188	5.3	5.4	..	4.9	5.8	5.4	6.3	6.4	..	P-48
P-49	682,086	4282	159	1208	920	1020	1376	1088	1188	5.1	5.2	..	4.7	5.6	5.2	6.1	6.2	..	P-49
P-50	987,977	6297	157	1208	920	1020	1376	1088	1188	5.0	5.1	..	4.6	5.4	5.1	5.9	6.1	..	P-50
P-51	799,668	5243	153	1208	920	1020	1376	1088	1188	4.9	5.0	..	4.5	5.3	5.0	5.8	5.9	..	P-51
P-52	707,221	4616	152	1208	920	1020	1376	1088	1188	4.9	5.0	..	4.5	5.3	5.0	5.8	5.9	..	P-52
P-53	219,085	4176	53	1206	998	998	1301	1093	1093	3.7	5.5	..	3.7	4.0	3.7	4.0	6.0	..	P-53
P-54	205,861	5098	41	1206	998	998	1312	1104	1104	2.8	4.3	..	2.8	3.1	2.8	3.1	4.8	..	P-54
							Pulverized-Fuel-Fired Units, 200 to 299 Lb. per Sq. In. Gage Pressure												
P-55	286,195	1863	154	1204	936	1026	1395	1129	1217	10.6	15.1	..	9.9	12.0	10.9	12.9	18.5	..	P-55
P-56	213,651	1536	141	1204	936	1026	1395	1129	1217	9.3	12.8	..	8.7	10.5	9.6	11.3	15.6	..	P-56
P-57	1,550,923	7804	198	1204	911	911	1355	1062	1062	6.7	11.6	..	6.1	7.1	6.1	7.1	12.3	..	P-57
P-58	1,066,331	6377	168	1203	910	1025	1382	1089	1204	5.4	6.5	..	4.9	5.9	5.5	6.5	7.8	..	P-58
P-59	1,063,182	6339	168	1203	910	1025	1382	1089	1204	5.4	6.5	..	4.9	5.9	5.5	6.5	7.8	..	P-59
P-60	1,141,413	6828	167	1203	910	1025	1382	1089	1204	5.3	6.4	..	4.8	5.8	5.4	6.4	7.7	..	P-60
P-61	1,184,966	7069	168	1203	910	1029	1382	1089	1204	5.3	6.5	..	4.8	5.8	5.4	6.4	7.8	..	P-61
P-62	1,281,404	7703	161	1204	911	911	1355	1062	1062	5.8	9.5	..	5.3	6.2	5.3	6.2	10.1	..	P-62
							Oil- and Gas-Fired Units												
OG-1	534,935	4052	132	1207	921	921	1362	1076	1076	8.9	24.8	..	8.2	9.6	8.2	9.6	27.0	..	OG-1
OG-2	2,239,528	7086	316	1209	878	878	1387	1056	1056	8.5	16.0	..	7.5	9.0	7.5	9.0	17.0	..	OG-2
OG-3	606,113	4359	139	1209	921	921	1374	1084	1084	7.4	8.4	..	6.8	8.0	6.8	8.0	9.2	..	OG-3
OG-4	556,517	5119	109	1208	896	928	1376	1064	1096	6.9	13.6	..	6.2	7.3	6.4	7.6	15.0	..	OG-4
OG-5	627,940	4616	136	1209	921	921	1374	1086	1086	6.6	8.3	..	6.1	7.2	6.1	7.2	9.0	..	OG-5
OG-6	573,978	4432	129	1209	921	921	1374	1086	1086	6.3	7.8	..	5.8	6.8	5.8	6.8	8.5	..	OG-6
OG-7	561,306	5014	112	1207	891	891	1362	1014	1014	6.5	12.9	..	5.8	6.6	5.8	6.6	13.0	..	OG-7
OG-8	586,112	5258	111	1207	891	891	1362	1014	1014	6.5	12.8	..	5.8	6.6	5.8	6.6	12.9	..	OG-8
OG-9	405,029	3646	158	1208	925	925	1361	1075	1075	6.1	16.3	..	5.7	6.6	5.7	6.6	17.6	..	OG-9
OG-10	428,021	5146	83	1207	921	921	1336	1050	1050	5.6	15.1	..	5.2	5.9	5.2	5.9	15.9	..	OG-10
OG-11	475,330	5523	86	1207	921	921	1336	1050	1050	5.1	15.6	..	4.7	5.4	4.7	5.4	16.4	..	OG-11
OG-12	402,505	4873	83	1207	921	921	1336	1050	1050	4.9	15.1	..	4.5	5.2	4.5	5.2	15.9	..	OG-12
OG-13	270,434	2209	123	1208	925	925	1361	1075	1075	4.8	12.7	..	4.4	5.2	4.4	5.2	13.7	..	OG-13
OG-14	230,414	2348	98	1208	925	925	1361	1075	1075	3.8	10.1	..	3.5	4.1	3.5	4.1	10.9	..	OG-14
OG-15	252,800	5684	63	1207	921	921	1336	1050	1050	3.7	11.4	..	3.4	3.9	3.4	3.9	12.0	..	OG-15
OG-16	254,101	6373	40	1207	959	1024	1354	1106	1171	3.1	6.1	..	3.0	3.4	3.1	3.6	7.1	..	OG-16
OG-17	270,380	5819	47	1208	925	925	1361	1075	1075	1.8	4.8	..	1.7	1.9	1.7	1.9	5.2	..	OG-17
							Miscellaneous Combinations												
M-1	972,538	6900	140	1207	919	919	1362	1074	1075	7.0	9.8	..	6.4	7.5	6.4	7.5	10.6	..	M-1
M-2	1,082,089	7712	140	1207	919	919	1362	1074	1074	7.0	9.8	..	6.4	7.5	6.4	7.5	10.6	..	M-2
M-3	890,503	6784	131	1207	919	919	1362	1074	1074	6.6	9.2	..	6.1	7.1	6.1	7.1	9.9	..	M-3
M-4	323,844	4733	69	1207	959	1024	1354	1106	1171	5.4	10.5	..	5.2	6.0	5.5	6.3	12.3	..	M-4
M-5	376,108	5521	68	1207	959	1024	1354	1106	1171	5.3	10.3	..	5.1	5.9	5.4	6.2	12.1	..	M-5
M-6	118,041	2227	53	1207	959	1024	1354	1106	1171	4.1	8.0	..	3.9	4.5	4.2	4.8	9.4	..	M-6

in (1) by the quantity defined in (2), on the eleventh page. The result is expressed throughout in thousands of B.t.u. per square foot per hour.

4 "Furnace Volume" has been taken as reported by the operator. It is expressed throughout in cubic feet.

5 The "Rate of Heat Transfer per Cubic Foot of Furnace Volume" has been obtained by dividing the quantity defined in (1) above by the quantity defined in (4) above. It is expressed throughout in thousands of B.t.u. transfer per cubic foot of furnace volume per hour. Note particularly that this does not refer to heat liberated within the furnace but to heat absorbed by water and steam.

6 "Grate Area" has been taken as reported by the operator. It is expressed throughout as projected area in square feet.

7 "Availability Factor" has been taken as the ratio between the sum of the hours during which the unit was steaming, banked, or available for service, to the total number of hours in the period.

8 "Use Factor" has been taken as the ratio between the sum of the hours during which the unit was steaming and banked to the total number of hours in the period.

9 In all charts the age of each boiler has been shown by placing upon the bar representing its performance a numeral indicating

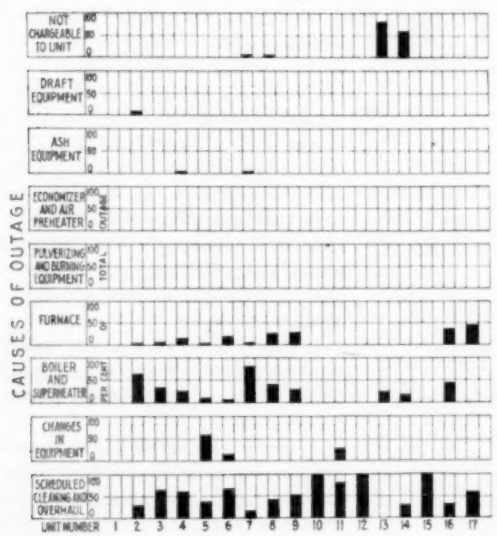


FIG. 7 ANALYSIS OF OUTAGE—OIL- AND GAS-FIRED STEAM-GENERATING UNITS

(Crippled periods classified as to cause and shown as percentages of total outage.)

the year in which it first produced steam. For convenience the two last digits only have been used, thus 1925 is indicated by '25.

10 Each boiler reported on has been given a designating symbol so that it can be identified throughout. The symbol consists of a letter and a numeral. The letter indicates the class of fuel or firing and the numeral indicates the particular boiler in that class. Thus "S-3" means the No. 3 unit in the stoker-fired class. For the sake of simplicity the letter has been omitted in the charts which indicate on their face the class to which they apply.

CONCLUSIONS

The authors feel it necessary to enter here a word of caution. They do not believe it safe to draw final conclusions from a single year's records of such a comparatively small number of boilers, particularly when the numerical values on which the study has been based are subject to the variables arising from different personal factors. The conclusions which follow should therefore be regarded rather as indications, and subject to correction if and when more complete data become available.

Averages of various sorts as obtained from this study are collected in Tables 5 to 8 and Figs. 9 and 10. It may be said, in general, that the results thus far deduced indicate that details of design and construction and operating skill are far more important in obtaining a high availability factor than are such things as rate of output per unit of surface, type of fuel, method of firing, and many of the other factors which have been widely discussed in this connection.

Study of the diagrams and charts indicates that boiler outage is not proportional to severity of use. In fact, it happens that the figures show more outage for many boilers working at a low rate of heat transfer than for those working at comparatively high rates for which data happened to be obtained. This is illustrated in Fig. 9. For easy interpretation imagine the triangle to represent the spreading flame of pulverized fuel, the square to represent the rectangular stoker, and the circle to stand for the initial letter of "other" in "other fuels and methods."

Rather curiously a similar condition is indicated with respect to rate of heat transfer per unit of furnace volume, as shown in Fig. 10.

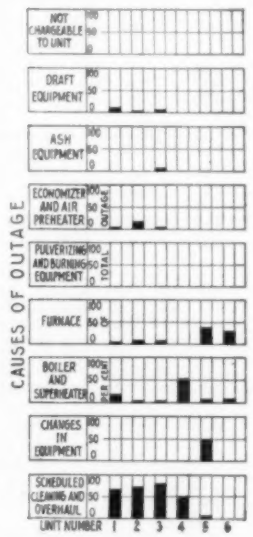


FIG. 8 ANALYSIS OF OUTAGE—MISCELLANEOUS COMBINATIONS

(Crippled periods classified as to cause and shown as percentages of total outage.)

If taken at their face values the results obtained would indicate a slight increase in availability with increasing severity of use. If this be the fact, it is reasonable to suppose that it follows from more careful design and operation in the case of equipment which is driven at the higher rates. The diversity of performances at substantially similar rates of output per unit of surface and per unit of furnace volume would seem to confirm such an interpretation.

Inspection of Figs. 9 and 10 shows plainly that there is no clear evidence of the superiority of either of the recognized methods of firing coal. Points representing stoker firing and pulverized-fuel firing are so well intermingled as to indicate that design and operation are undoubtedly of far more significance with respect to availability than is the method of firing adopted.

An effort was made to obtain further light on this subject by calculating some sort of weighted averages. The results are shown in Table 5 in which the averages are based on the Net Crippled Hours. This expression is intended to cover the author's best estimates of the part of the Total Crippled Hours chargeable to the fuel-burning equipment and furnace. A high degree of precision cannot be claimed because of the unavoidable inaccuracies due to the original reports having been filled out by different individuals with different interpretations, standards, and methods. The averages reported are weighted in the sense that they were derived from the totals of Total Hours Reported and the totals of Net Crippled Hours.

The numerical values obtained appear to indicate highest availability for oil- and gas-fired units, slightly lower availability for pulverized-fuel-fired units, and lowest availability for stoker-fired units. Study of the underlying figures as tabulated indicates that the stoker-fired average is materially lowered by the figure for the 400–499-lb. class. An inspection of the data underlying the tabulated figures shows the low availability of this particular

TABLE 5 COMPARISON OF AVAILABILITY FACTORS WITH DIFFERENT METHODS OF BURNING FUEL

Pressure class	No. of units	No. of plants	Average age of units, years	Total hours reported	Total crippled hours	Crippled hours not directly chargeable	Net crippled hours	Availability factor (Net crippled hours)
A—Stoker-Fired Units								
500–700	20	4	3.6	162,499	32,485	8,497	23,988	0.852
400–499	8	3	2.8	58,368	15,805	971	14,834	0.745
300–399	53	13	4.3	435,769	73,726	3,811	69,915	0.840
200–299	20	6	6.5	151,423	22,593	1,236	21,357	0.859
Totals	101	..	...	808,077	144,609	14,515	130,094	...
Average of stoker-fired group (101 units)								**0.839**
B—Pulverized-Fuel-Fired Units								
400–499	20	6	3.0	152,838	22,213	2,167	20,046	0.869
300–399	34	10	2.8	267,418	41,263	8,832	32,431	0.879
200–299	8	3	4.7	63,840	8,800	1,762	7,038	0.890
Totals	62	..	...	484,096	72,276	12,761	59,515	...
Average of pulverized-fuel group (62 units)								**0.877**
C—Oil- and Gas-Fired Units								
.....	17	7	2.9	133,876	18,179	4,331	13,848	**0.897**

TABLE 6 COMPARISON OF AVAILABILITY FACTORS OF DIFFERENT PRESSURE CLASSES

Pressure class	No. of units	No. of plants	Average age of units, years	Total hours reported	Crippled hours	Crippled hours not directly chargeable	Net crippled hours	Availability factor
500–700	20	4	3.6	162,499	32,485	8,497	23,988	0.852
400–499	36	13	2.9	271,498	46,494	5,399	41,095	0.849
300–399	96	27	3.6	776,772	124,697	13,940	110,757	0.858
200–299	28	9	6.0	220,263	31,393	2,998	28,495	0.87

TABLE 7 COMPARISON OF AVAILABILITY FACTORS OF UNITS HAVING AIR PREHEATERS WITH THOSE NOT HAVING AIR PREHEATERS

	Total hours reported	Crippled hours	Crippled hours not directly chargeable	Net crippled hours	Availability factor
A—Stoker-Fired Units					
With preheaters.....	240,519	53,770	5540	48,230	0.800
Without preheaters....	567,540	90,839	8975	81,864	0.856
B—Pulverized-Fuel Units					
With preheaters.....	221,014	40,693	7283	33,410	0.849
Without preheaters....	263,082	31,583	5478	26,105	0.901
C—Oil- and Gas-Fired Units					
With preheaters.....	90,531	14,644	3484	11,160	0.877
Without preheaters....	43,345	3,535	847	2,688	0.938

TABLE 8 TURBINE-UNIT AVAILABILITY[1]

Year of operation	Number of units	Availability factor
1	49	0.858
2	28	0.901
3	13	0.861
4	13	0.861
5	16	0.852
6	22	0.870
7	14	0.790
8	12	0.835
9	4	0.798
10	11	0.900

[1] Turbine data taken from Serial Report of the Prime Movers Committee Engineering National Section, National Electric Light Association, 1926–1927.

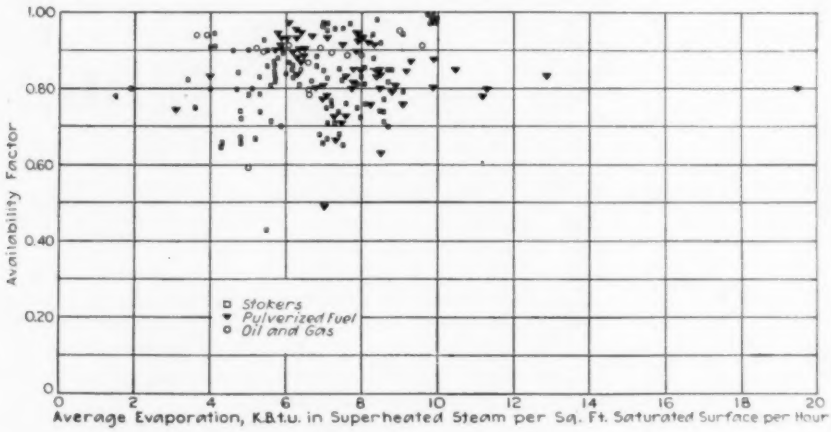


FIG. 9 EFFECT OF STEAMING RATE ON AVAILABILITY

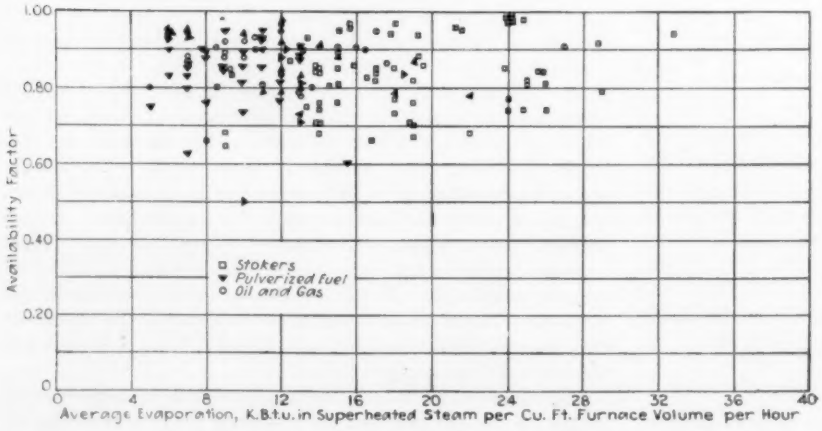


FIG. 10 EFFECT OF STEAMING RATE ON AVAILABILITY

class to be caused by the inclusion of certain cases which may be properly regarded as abnormal. This is as one would expect, since there is no reason why this particular class should, per se, be so radically different from the other class reported. If the figures for the 400–499-lb. class be omitted, the weighted average availability factor for the stokers becomes 0.846, roughly about as much lower than the value for pulverized-fuel firing as the value for that method is below oil and gas firing. While this is about what one would expect, the authors reiterate that it is not deemed proper to regard such figures as more than indications of probability. The records of availability reported herein for many stoker-fired boilers certainly show that such equipment need not have a low availability factor.

It should be noted that six boilers were reported as using both oil and coal firing. These were omitted when deriving the values given in Table 5.

A study of availability factors of the various pressure classes has been made and the results are assembled in Table 6. There is hardly a significant difference between the various final figures except that the higher availability factor for the lowest pressure class may represent a real difference. At least it is in line with what one might expect.

The situation is quite different with respect to the effect of air preheaters on availability. Results are given in Table 7. It is evident that for all classes tabulated, the air preheater, on the average, decreases the availability factor by about five points. The showing is so consistent that it is believed reasonable to assume that it represents fact. Air-cooled furnace walls were not considered as air preheaters as they seldom raised the temperature of the combustion air above 130 deg. fahr. It should be noted that outages due to things not affected by the use of preheated air were excluded in calculating the factors in Table 7, so that the final values represent the effects of preheated air as accurately as the character of the data permitted.

No conclusions could be drawn with respect to the age of equipment upon availability. When it is realized that most of the units reported on have been placed in service within the past five years, this finding appears reasonable. The older units are not yet old enough to have reached that point where excessive maintenance or even rebuilding is necessary; the younger units are not yet old enough to have passed out of that period during which small changes and modifications are required.

Attempts were made to produce comparison on several other bases with little or no success. The comparatively small number of units and the short operating period covered placed very definite limitations on the possibilities in this respect.

As a matter of interest, Table 8 is included. It shows availability factors for complete turbine units, including auxiliaries. Comparison of the figures in this table with those in the earlier tables will show the degree of real progress made in recent years toward making the availability of steam-generating equipment as high as that of the principal steam-using equipment.

Appendix No. 1

BOILER OPERATION LOG USED IN COLLECTING DATA

The American Society of Mechanical Engineers
Power Division

BOILER OPERATION LOG

Company ..... City ..... State .....

Station ..... Boiler No. ..... Week Ending ..... 192...

I GENERAL DATA

		Sun.	Mon.	Tues.	Wed.	Thur.	Fri.	Sat.
1. Hours Steaming								
2. Hours Banked								
3. Hours in Reserve—Cold								
4. Hours Crippled for Repair								
5. Total Water Evaporated	1000 lb.							
6. Avg. Output during Steaming Period	1000 lb./hr.							
7. Maximum Output	1000 lb./hr.							
8. Avg. Draft in Combustion Chamber	—in. H_2O							
9. Avg. Draft at Boiler Exit	—in. H_2O							
10. Avg. Flue Gas Temp. at Boiler Exit	—°F							
11. Avg. Flue Gas Temp. Entering Stack	—°F							
12. Avg. Temp. of Feed Water Entering Boiler	—°F							
13. Avg. Temp. of Air Entering Furnace	—°F							
14. Avg. Carbon Dioxide in Flue Gas in Last Pass	—%							
15. Boiler Mud Drum Blown Down								
16. Water Walls Blown Down								
17. Water Screen Blown Down								
18. Soot Blown (a) Boiler								
(b) Economizer								
(c) Air Preheater								

II COAL AND ASH ANALYSES—Coal Taken as Fired

(a) Heat Value as Fired	B. t. u./lb.	
(b) Moisture in Coal	—%	
(c) Ash	—%	
(d) Fixed Carbon	—%	
(e) Volatile Matter	—%	
(f) Sulphur	—%	
(g) Fusing Temp. of Ash	—°F	
(h) Unburned Carbon in Refuse	—%	

III KIND OF COAL BURNED

(a) Kind of Coal (Anth., Bit. etc.)	
(b) Grade of Coal (Mine Run, Slack, etc.)	
(c) District (State or County)	

(OVER)

IV OIL ANALYSIS

(a) Grade of Oil Used		
(b) Gravity	—°Baumé	
(c) Flash Point	—°F	
(d) Heat Value	—B. t. u./lb.	

V ANALYSIS OF BOILER LOADING—Based on Boiler H. S. & Water Wall H. S.

Rating—%	50	75	100	125	150	175	200	225	250	275	300	325	350	375	400	425	450	475	500
Hours																			

VI CAUSES OF BOILER OUTAGE AND REMARKS

Sunday

Monday

Tuesday

Wednesday

Thursday

Friday

Saturday

Discussion

J. R. Baker.[4] A paper of this character suggests almost innumerable questions, more perhaps than comments, so that the writer will ask to be pardoned if his remarks are in the form of questions regarding the causes of unavailability rather than an addition to the basic data presented in the paper.

One question that immediately arises is that of effect of use factor upon availability, inasmuch as operating skill is concluded to be a determining factor in obtaining high availability. Some engineers advocate taking a boiler out of service for cleaning and inspection every four to six weeks, while others suggest much longer intervals, even extending to a year. It would be interesting to know if a study of the daily operating logs would indicate which system of operation and preventive maintenance leads to higher availability. In regard to this point also, and with due recognition of the statistical accuracy of the average stated in the paper, it seems that the average availability factors of stoker-fired units and pulverized-fuel-fired units are inverse to the average use factors for the two types of installation—i.e., stokers average 0.769 and pulverized-fuel boilers 0.723, for use factors. Are such figures as significant as the average availability factors?

Again it would seem to be logical in the case of boilers to expect that a factor embracing both severity of use and the use factor, or length of use in the particular period, would be of importance in determining the availability factor.

With regard to Figs. 9 and 10, the conclusion with respect to severity of use does not seem apparent from an inspection, which indicates that availability decreases with increase of severity of use for pulverized-fuel-fired units, in the range between 6 and 12 K.B.t.u. for instance. The plotted points do, however, appear to indicate a slight increase of availability for stokers in Fig. 9, but more nearly a constant or decreasing value in Fig. 10. If, however, note is made that the points at 10 K.B.t.u., which tend to throw the stoker availability upward with severity of use, are derived probably in one station solely, the conclusion regarding stokers would also appear doubtful.

The conclusion with respect to air preheaters is of interest

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since there has been a discussion recently tending to show that air preheaters deteriorate more rapidly on stoker installations than on pulverized-fuel-fired boilers. Evidently both types of installations are affected equally as to availability. It would be of interest to have further discussion of the causes of outages due to preheaters; i.e., is the trouble in the preheater elements, in the cleaning devices, with associated fans, or with duct work?

With regard to boiler and superheater outages, could an explanation be given to indicate the proportion of outages—as between boiler-water conditions and structural defects, for example?

Another point of interest which may be evident in the detailed figures, but which is not mentioned in the body of the paper, is the effect of boiler size and number of component parts upon availability. There are of course many factors involved, but if there can be any generalization it might be of interest to state whether any difference is apparent.

HENRY KREISINGER.[5] The paper gives us a standard of the availability of steam-generating units for service. We know now that properly installed units can be expected to be available for operation about 85 per cent of the time; that is, approximately six days out of a week.

In Table 2 the last two columns give the heat value of the coal and the kind of stoker or type of firing. It is interesting to note that traveling grates are used with low heat value of fuel and underfeed type of stoker with high heat value coals. That is, traveling-grate stokers are used with high-ash low-grade coals, whereas underfeed stokers are used with high-grade high-carbon Eastern coals. It so happens that the low-grade coals are also free-burning coals and the ash is generally more fusible than the ash of the high-grade Eastern coals. The high-ash coals are therefore more easily burned on the traveling-grate stoker, which does not agitate the fuel bed. The agitation of the fuel bed mixes ash into the burning coal and causes troublesome clinkers.

The high-grade Eastern coals are also coking coals, and therefore cannot be handled satisfactorily on traveling grates. They are burned with much better results on the underfeed stoker, which agitates the fuel bed, thereby breaking the caked-up fuel.

The first 20 stoker-fired units which have been grouped as the highest pressure units are also all fired with traveling grates. Examination of Fig. 5 shows that, if anything, the outage on account of the stoker is less than with the other groups. This is probably one of the reasons why this type of stoker was selected for the low-grade fuels.

Comparison of the last two columns of Table 5 for the stoker-fired and pulverized-coal units shows that the heat value of the coals burned on stokers is generally higher than for the units burning pulverized coal. There are a number of stoker units burning coal exceeding 14,000 B.t.u., and none of them uses coal below 10,000 B.t.u. On the other hand, there are no pulverized-coal units burning coal having heat value as high as 14,000 B.t.u., and there are a number of them burning fuel with heat value of less than 10,000 B.t.u.

Of the total 62 units burning pulverized coal, only 10 units are direct-fired. Apparently there are more units in the central stations fired by the storage system than by the unit system. Although recently a number of central stations were built with the unit system, they have not been operated long enough for reliable records.

In Fig. 9 the pulverized-coal points tend to group themselves toward the side of the higher evaporation per square foot of water-evaporating surface. This is probably due to the fact that pulverized-coal units are more generally equipped with water-cooled furnaces than is the case with stoker-fired units. Water-heating surface placed in the furnace is more effective in evaporating water than heating surface in the bank of tubes of a boiler. This is especially true if the heating surface in the furnace is bare.

In Fig. 10 the pulverized-coal points group themselves toward the side of the lower evaporation per cubic foot of combustion space. This is undoubtedly due to the fact that in the case of pulverized coal all the heat must be liberated in the combustion space, and therefore the volume of the combustion space must be larger than with stoker firing, where only about 40 per cent of the heat is liberated in the combustion space, the other 60 per cent of the heat being liberated in the fuel bed.

BERT HOUGHTON.[6] The paper covering the reliability and availability of steam-generating units is most interesting, and the authors are to be congratulated in compiling so much information within the time they have been engaged in the work. As they have pointed out, no rigid conclusions can be drawn from the data so far assembled, but certainly the writer is as much surprised as they are that increased punishment of the equipment due to high ratings appears from their data to give longer life or greater availability. The writer's experience and observation in the past do not bear this out, and it might be that if instead of average loading one could obtain "load-duration curves," and also a few typical daily cycles of operation to indicate the range and rapidity of change in load and severity of loading, somewhat different conclusions would be reached. Apparently some attempt has been made to evaluate this loading characteristic in the manner of drawing up the diagrams to Fig. 1, but graphically the comparison is not fundamental as certain numerical characteristics at the top of the graph must be visualized at the same time.

The comparison made by the authors on the basis of average evaporation in K.B.t.u. in superheated steam per square foot of saturated surface per hour is apt to mislead, and the comparison should preferably be made on the basis of heat absorbed by the boiler and water screen in K.B.t.u. per hour per square foot of saturated surface. This would make the comparison of all boiler installations uniform and eliminate the heat absorbed in the superheater, economizer, and air preheater, which items should be treated separately. There are a number of boilers in the tabulation with and without economizers and air preheaters and with different amounts of superheating surface.

From the standpoint of the boiler alone without regard to the heat-balance cycle of the station, the work done by the saturated surface has nothing to do with the heat transfer in the economizers or preheaters and superheaters. Therefore the saturated surface cannot be credited with the additional heat transfer of these other surfaces even though there is some interrelation in the final results and temperature of the steam. In order to bring out this point, the writer has taken the No. 54 boiler at Hudson Avenue and figured out the heat transfer; first, with the economizer and superheater; second, without the superheater; and, third, without the economizer and superheater. The results are in K.B.t.u. per hour per square foot of saturated surface. Assuming the last-mentioned method as the basis, there is a difference between the method used in the paper and the method advocated which varies from 19.3 to 29.3 per cent at different ratings. Neglecting the heat absorbed by the superheater only, this difference varies from 4.2 to 11.7 per cent. In other words, where the comparison is being given between installations with and without economizers, air preheaters, and with difference in superheaters, the only basic way to compare

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[6] Operating Superintendent, Brooklyn Edison Company, Brooklyn, N. Y. Mem. A.S.M.E.

their boiler and furnace performances is on the basis of heat absorbed by their separate surfaces. It is therefore proposed that the definitions on the eleventh page numbered 1 and 3 be changed to suit the scheme of comparison just outlined.

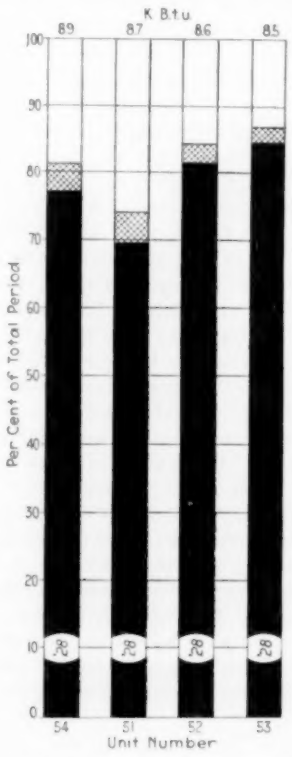


FIG. 11 RECORD OF OPERATION—STOKER-FIRED UNITS
(For 400 to 490-lb. class.)

LEGENDS

White.....Crippled
Double hatching.....Banked
Black.....Steaming

	UNITS			
Boiler No.	51	52	53	54
Steaming hours	6080	7092	7352	6748
Banked hours	379	243	240	336
Out hours	2301	1425	1168	1676
Total hours	8760	8760	8760	8760
Available hours	6459	7335	7592	7084
Availability factor	0.7373	0.8373	0.8667	0.8087
Steaming hours, per cent of total	69.41	80.96	83.93	77.03
Banked hours, per cent of total	4.32	2.77	2.74	3.84
Out hours, per cent of total	26.27	16.27	13.33	19.13
Total steam generated, lb. 10^3	1,155,733	1,324,905	1,355,973	1,305,553
Average evaporated lb. per hour 10^3	190	187	184	193
Heat absorbed entire unit, B.t.u. per lb.	1155	1155	1155	1155
K.B.t.u. in superheated steam per square foot of saturated surface per hour	8.7	8.6	8.5	8.9

Enlarging upon the earlier comment on the detail boiler-operating cycle, the authors have neglected to give the peak performance of the boilers, and this has a marked influence on the outage of stokers and boilers due to slagging, clinkering, possible burning out of tuyères, depreciation of clinker grinder, etc. The writer believes that it would be decidedly worth while to obtain the average sustained steaming capacity of a group of boilers of from four to five hours for the period of at least six consecutive days as a basis for determining what really can be expected of a boiler house. Such work has been carried out at the Hudson Avenue Station, and the results indicate quite a difference between the mathematical determination of the comparison of the capacity of a boiler house from individual boiler test and what may be expected as an average from a group of eight to twenty boilers under normal operating conditions.

In addition to the various items tabulated, it would be most interesting if the authors could obtain the per cent efficiency obtained with the various installations. There might be some connection between outage and efficiency of the boiler house, which relation would be well worth while to analyze.

The writer has compared the fifth row of boilers at the Hudson Avenue Station, which are in the 400-lb. class, with the data submitted in the paper. Fig. 11 is a chart showing this comparison. Due to the fact that the fifth row of boilers was not put in operation until shortly after the initial date for starting the tabulations, these particular boilers were left out of the authors' paper, but this information is furnished as part of the discussion.

EDWARDS E. FISH.[7] The authors are to be commended for the pains which they have taken to gather the unusual statistics on which this paper has been based. The comparisons that have been drawn from them can be of real value. The introduction, development, and growth of the steam turbine and the consequent insistent and increasing demands for more steam at higher temperatures and pressures have been responsible for the very great development in boiler practice, which confessedly has been the follower and not the leader in this development.

The very fact that statistics of this nature could cover only a relatively short period is evidence of the rapidity of the growth.

There has been much to learn by both operators and manufacturers, which knowledge could be attained only by practical experience, and it is largely this experience that guides designers in avoiding objectionable features and evolving correct designs.

In an investigation of this kind there must necessarily be a limit to the minutiae into which it is possible to go, but it would be interesting and instructive to know in detail the reasons for outages. Figs. 6, 7, and 8 divide these causes between the various equipment of a boiler plant, but do not state the character of the troubles. A number of possible causes applying to each of the several subdivisions will suggest themselves to one's mind in thinking it over.

The boiler-operation log shown as Appendix No. 1 does not suggest any causes. It would seem as if such would be of very great service, both to manufacturers of the various types of equipment and to designing engineers, in influencing future design, and that space for their inclusion should be provided in any future effort to gather information.

It would rather naturally be expected that maintenance of the steam-generating end of a power plant would be a greater problem and more expensive than that of the prime mover. However, the statistics in Tables 5 and 8 seem to indicate that, on the average, the percentage of crippled periods of the former is about the same as the latter. This is a tribute to the intelligence and care with which steam-generating units are handled, as well as evidence that manufacturers have profited by experience.

Generally speaking, public utilities have been pioneers in the developments in the power-plant field, and they have primarily been of great service to each other and secondarily to the steam-using industries, of all of which this paper is an illustration.

The writer is especially pleased to note that the authors specifically call attention to the fact that the information presented forms a comparatively narrow premise from which a broad conclusion should not be drawn. There is always the temptation to take figures at their face value without giving due

[7] Vice-President, Heine Boiler Company, New York, N. Y. Mem. A.S.M.E.

consideration to the conditions under which they were gathered and which govern their true significance.

The influence of air preheaters on the availability factor is interesting. One would be inclined to expect less favorable results in connection with stoker-fired plants than in the case of pulverized fuel or oil and gas, but the differences as shown by the statistics are not so very different, although the indications are that it is less with pulverized fuel than with the others.

Investigations of this sort supply one more item by which the judgment of power-plant designers may to some extent be guided.

It is to be hoped that either the authors of this paper or someone else will in due time extend the investigation.

JOHN H. LAWRENCE.[8] The Central Station Committee of the Power Division is to be complimented for initiating such an important study as that covered by this paper, and the authors are to be congratulated for compiling such an enormous amount of information in such a way that it is readily interpreted. It is a very difficult job to induce various companies and engineers to furnish information as it is requested, but in this case the authors have apparently secured very good results.

It is to be regretted that this study was not initiated some years ago, at least before the development of the water-cooled furnace. There is no doubt that the water-cooled furnace has materially increased the availability factor of boiler units, as the most serious cause of outages in the average plant was maintenance on the furnace brickwork. This brickwork maintenance in some plants ran into startling figures.

The water-cooled furnace has reduced boiler outage, due to wall maintenance, almost to a negligible amount, especially in the case of pulverized-fuel boilers.

It may be of interest to know that the older boilers in the East River Station operated at 400 per cent of rating or over for a period of one year up to the time that the new 800,000-lb. boilers were placed in operation. Every one of these boilers was available every working day on the peak, and the only trouble on the walls was the replacement of one sidewall tube, which was due to a condition which can be eliminated in the future.

One boiler in this station has been available every day for the last year and eight months. These boilers are entirely water cooled, with the exception of a small amount of refractory around the headers and air lanes.

When the late Thomas E. Murray suggested 100 per cent water-cooled surfaces in these boilers, the idea was frowned upon by almost every one, as it was thought that an all-water-cooled furnace would materially affect the efficiency of the boilers. However, he took the position that this might possibly occur, but if it did the remedy should not be through covering the water-wall surface, but by improving the design of pulverized-fuel burners.

The efficiency was low on starting these boilers, but the burners were improved, and the result has been that these boilers compare very favorably in efficiency with any other boilers operated under the same conditions.

From the experience at East River Station, the writer would say that the statement of the authors on the second page no longer holds true, as the availability factor of a boiler unit so designed should be well over 90 per cent. This statement reads as follows:

> It is the hope of every central-station designer and operator that some day the steam-generating equipment may be brought to such a stage of perfection of design and construction that, when properly operated, it will have the same degree of availability for service as characterizes the modern steam turbine attached generator and subsidiary or auxiliary equipment.

The National Electric Light Association Prime Movers Report for 1926 and 1927 shows a weighted availability factor for units in service from one to ten years of 86 per cent. This figure is even lower than the availability factor reported by all the pulverized-fuel units which are given in this paper.

It is safe to say that the modern boiler unit on an average has an availability factor considerably in excess of the turbine.

The authors comment concerning the availability of boilers at high ratings being no lower than at low ratings. In fact, if anything, the availability is slightly higher at high ratings. This is contrary to the experience of the time before the introduction of the water walls. At that time it was felt that boilers could not be operated at high ratings without causing considerable punishment to the walls. There is no doubt that high rating on a refractory furnace is very injurious, but there were no grounds for the argument that the boiler itself might be damaged. Recent practice has shown that the boiler is capable of withstanding almost any load to which it might be subjected—at least so far as it has been possible to operate boilers up to the present time.

Table 6 shows the comparison of availability factors of different pressure classes. Perhaps the evidence is not sufficiently complete so that it is conclusive. However, if it justifiably indicates that availability factor decreases with increase of pressure, the statements which have been made concerning high-pressure plants are not in accord with the facts presented in this paper.

The margin of gain in high pressures is so slight that if the availability factor does decrease with pressure, as shown, the probable savings which have been estimated will be wiped out by the extra expense due to more equipment being necessary to carry the load on account of decreased availability.

NEVIN E. FUNK.[9] The efforts that the authors have spent in compiling the comparative data on the availability of boiler units are to be very much commended by the engineers of all industries whose work involves them in boiler-plant design or operation. To the writer's knowledge this is the first time comparative data of this kind for boiler units have been made available to the industry.

The writer desires to emphasize the fact, which the authors have brought out in their paper, that these data should be used with a great deal of caution in making comparisons. This cautionary advice is no reflection upon the data presented by the authors, but is merely a warning based on personal experience in gathering data from many sources.

No matter now concisely or how carefully methods of collecting data are prepared, they are always subject to misinterpretation by those responsible for collecting the data at their source. This misinterpretation may be due to misunderstanding the intent of the person requesting the information or may be due to differences of opinion as to the location of the dividing line between various elements of the data sought.

As more concrete examples of this thought in which differences of interpretation might easily exist, the writer believes it is not out of place to point out that even what might appear as so simple a thing as determining the actual steaming and banking periods of boilers in the field is subject to error, for unless the pressure of an individual boiler is definitely below the line pressure, it cannot with surety be determined that this boiler is not steaming. No instruments have yet been evolved which are delicate enough to measure extremely low steam outputs if they are capable of measuring high steam outputs. The concrete

[8] Vice-President and Engineering Manager, Thomas E. Murray, Inc., New York, N. Y. Mem. A.S.M.E.

[9] Assistant Chief Engineer, Philadelphia Electric Company Philadelphia, Pa. Mem. A.S.M.E.

proof of this is the fact that it is possible to carry a 10,000-kw. output on a station with a 180,000-kw. capacity without any apparent output from any of the boilers shown on the steam-flow meters and with no draft under the fires.

And again, it is quite possible when a boiler is in reserve that the operating force may elect to do some repair work on the unit which actually puts it out of service, but which repair work could be at least temporarily completed by the time that the fire in the furnace had been built up to a point where the boiler could be utilized. This boiler would not be classified as crippled.

These examples indicate that even though the authors have made very definite demarcations between the four classes of conditions of boiler operations, nevertheless the men in the field who are making the subdivision of time of these various classes of service might very readily err on one or the other side of a correct line, which in the total might change the results obtained materially.

Quite obviously, boilers S 29 to 35, inclusive, have such an extremely low period of crippled time that it seems almost impossible to believe that the data submitted on these units can possibly conform to that submitted on the remaining boiler units.

This type of inconsistency always exists in collecting data of this kind, and it is only after data have been collected several years from the same units that these inconsistencies can gradually be eliminated due to the bulk of the data that are obtained.

This, then, is another reason for caution in using the statistics presented; that is, they are not of sufficient duration to permit the authors to eliminate obvious inconsistencies in the reports of some or all of the field forces. Furthermore, the short period of time over which these data have been compiled does not allow for the variation in maintenance that takes place from year to year, and therefore may not give the possible trend in the performance of the units listed.

The authors have shown in Table 2 a comparison of the various availability factors, and it is interesting to note from these data the definite increase in percentage of pulverized-fuel units above stoker-fired units and the corresponding increase of oil- and gas-fired units over pulverized-fuel-fired units. However, the weighted average age of the stoker-fired units is greater than that of the pulverized-coal-fired units or the oil-fired units. This may or may not have some effect on the factors involved, but is one of the reasons why the relative difference in availability factors should not be used with too greater certainty.

Figs. 5 and 6 in the authors' paper indicate that the furnaces of the stoker-fired units seem to be less responsible for outage of the boiler than the furnaces of the powdered-coal-served units.

This, however, is not indicative of the possibilities of more furnace trouble with powdered coal, since the powdered-coal boiler appears on the average to have less crippled time than the stoker-fired boilers. This in itself would have a tendency to increase the percentage of outage time due to the furnace if the furnace troubles were of the same order in actual hours outage as the stoker furnaces.

The writer is sorry that the authors did not make a comparison of the subdivision of the percentage outage as a function of the total hours included in the period under observation rather than a percentage of the crippled period, since this would give a more ready comparison between the different types of firing so far as the effect on availability is concerned.

It is interesting to note the relative necessity for correction in the initial design of the different type units and the effect of this on the crippled time of the particular units involved. This can be given as follows:

Type of firing	Per cent of total units involved	Average per cent increase of crippled time due to design changes on units involved
Stokers	0	0
Pulverized fuel	11.3	70
Oil and gas	11.8	139

The foregoing comparison is subject to the same limitations noted by the writer in regard to some of the comparisons made by the authors.

The authors have presented two extremely interesting curves in Figs. 9 and 10, which show that crippled time is much more a function of design or careful operation than of rate of combustion or type of firing equipment. Of course, data for several additional years may indicate a very definite trend for each type of firing, and it is to be hoped that it is not asking too much of the authors to continue the good work they have started.

These data extended over a longer period will be extremely valuable.

Ollison Craig.[10] At the conclusion of this paper are two figures; one, Fig. 9, shows boiler availability plotted against average evaporation per unit of saturated surface per hour; and the other, Fig. 10, shows boiler availability plotted against evaporation per unit of furnace volume per hour. Triangles represent data from powdered-fuel-fired units, squares represent data from stoker-fired units, and circles represent data from units fired by other means. Abscissae used in either case are a function of coal-burning rates. Inspection of the scattered points indicates that increase of evaporation does not decrease availability, but might even be considered to increase availability, as pointed out by the authors. It might be expected that, if availability was affected by method of firing, the difference would most probably be found in outage on account of firing equipment or furnace. In the case of stoker firing certain experiences bear out the conclusion quite definitely. This writer recalls one case in which two stokers of same type and size, set alike, under the same size and kind of boilers, were fired for a period of one week, one stoker being carried on bank and the other operated to limit of capacity. The stoker which was pushed to limit of capacity suffered no damage, while the banked stoker required renewal of a number of parts due to burns.

If coal is burned on a stoker at a greater rate, an average thicker fuel bed will usually be maintained, providing more protection to the iron parts from furnace radiation; and more air and raw coal will carry more heat away from the iron. If the fuel-burning rate is low, fire can burn down into the fuel bed at certain points and come in contact with the iron.

In the case of powdered-coal firing an increase of availability could be shown due to the fact that installations operating at the greater rates of heat absorption per square foot of saturated surfaces are more probably equipped with water walls, and outage due to furnace repairs is reduced. A study of data presented, in the case of powdered-coal firing, indicates furnace outage to be about 24 per cent of total outage with refractory walls and about 15 per cent with water-cooled walls.

This writer wishes to raise a point in connection with Figs. 9 and 10. Fig. 9 compares powdered-coal firing and stoker firing on the same basis, but Fig. 10 does not. Heat absorbed by steam may be considered approximately proportional to heat released in the furnace. So, heat released in the furnace could have been used as abscissae in Fig. 10, and the relative position of the points would remain about the same. However, it must be remembered that in the case of powdered-coal firing the heat released in the furnace is the result of burning the coal, both solids

[10] Consulting Engineer, Riley Stoker Corporation, Worcester, Mass. Mem. A.S.M.E.

and gases, as the coal passes through the furnace space, while in the case of stoker firing the greater portion of the fuel is burned on a practically stationary fuel bed, and the heat released in the furnace space by burning combustible material in this space represents the smaller portion of the total heat in the coal burned. The two conditions are in nowise comparable.

Table 5 indicates that of the three methods of firing, stoker-fired installations have on the average the greatest outage, powdered-coal firing is next, and oil and gas firing the smallest outage. A knowledge of the reasons would be interesting. It might be considered as being largely due to the firing equipment and the effect on the furnaces. Averages taken from data in Table 3 indicate that, in the case of stoker installations, the stokers and furnaces are responsible for 27.5 per cent of the outage time, 15 per cent being due to the furnaces and 12.5 per cent due to the stokers. Also, in those installations having refractory walls, furnaces are the cause of 16.2 per cent of the outage, while on the case of water-wall installations this figure becomes 10.4 per cent.

In the case of powdered-coal installations, powdered-coal equipment and furnaces are responsible for 24.8 per cent of the outage time, this being divided 19.2 per cent for the furnaces and 5.6 per cent for the powdered-coal equipment. Outage due to furnaces is 24.1 per cent in the case of refractory walls and 14.9 per cent in the case of water walls.

The data would then indicate that powdered-coal equipment is more dependable than stoker equipment, as indicated by comparison of the outage percentages of 5.6 and 12.5, but that the percentages 19.2 and 15 indicate that there is room for development along the lines of furnace design in the case of powdered-coal firing. This development will probably occur through better designs of water walls for high capacity units and application of better knowledge of furnace design, together with better designs of burners in the case of refractory walls.

A decided difference in availability is shown with and without air preheaters. A knowledge of the reason would be interesting. Is it due to repairs required to preheaters? Does any type of preheater cause less outage than others? Is the outage due to the effect of hot air on other parts of the equipment, and if so, how does it vary with the different methods of firing?

J. G. Worker.[11] There have been read two very interesting papers. Mr. Clark tells the best kind of a boiler plant that can be obtained and the amount of money that can be invested wisely, and Messrs. Hirshfeld and Moran show how easy it is, after spending this money, through detailed weaknesses and cripples to destroy the proper return on the investment.

Both papers state that we are in the midst of a business consideration of the utility of boiler units, and the information and data in these papers should not, as the authors caution, be used to draw final conclusions, but rather to help correct weaknesses and give more skilled attention to those elements of design that affect reliability.

Particularly interesting are the stoker data in these papers. A few years ago stoker and boiler units were of small size as compared to the modern units of today. More coal per foot of furnace width is being fed than ever before, which means the generating of more steam per foot of furnace width.

It will be noted from Table 9 that the newer stations now under construction, such as Delray No. 3 of the Detroit Edison Company and the Bewag Station of Berlin, Germany, are designed to give very high steam outputs per foot of furnace width.

Control of Stoker Fuel Beds

The control of the fuel bed, and especially the big fuel beds required for large steam units, introduces forces that must be opposed by the mechanical design of the fuel-feeding structure of a stoker mechanism. When stokers were only eight or ten feet long, as of a few years ago, the fuel-feeding mechanism of older stokers was ample to withstand any forces introduced in handling the fuel to the grates. With larger stokers a completely new development—although following the same principle, but really a new departure in design—has taken place, as shown in

TABLE 9

Station	Pounds of steam per hour per foot of furnace width, stoker-fired plants		Pounds of steam per square foot of furnace area, stoker-fired plants	
	Maximum	Continuous	Maximum	Continuous
Delray No. 3	17,200	12,500	735	530
Bewag	17,000	11,400	835	555
Beacon Street	16,700	6,600	640	250
Hudson Avenue	15,200	9,000	770	450
Edgar	12,500	9,000	650	465
Cons. Htg. (Balt.)	8,000	4,900	420	260
Richmond	7,000	4,200	535	330

Fig. 12. An example of how the design of different parts has changed, due to this influence of controlling all parts of a big fuel bed, is shown in Fig. 13. This shows how much more

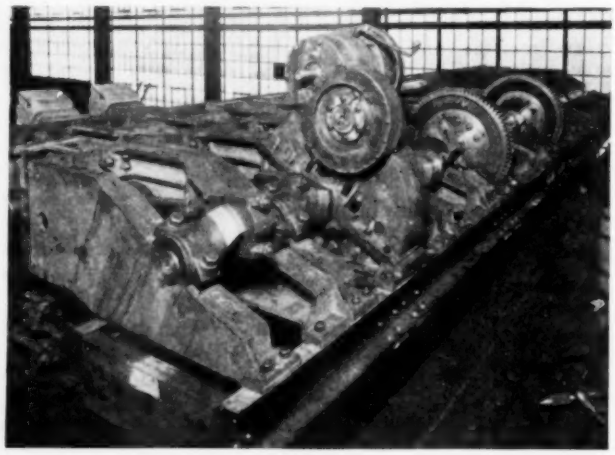


Fig. 12

strength is being put into the high-capacity stoker than into the smaller stoker of a few years ago.

Starting Up Boiler Units

A boiler unit today has many new things attached to it; and it has been found in the starting of new plants that one thing or other happens, making it necessary to take the boiler unit out of service until changes are made. It seems that we can help this situation by extending the same engineering skill in the starting up of plants as has been manifested in the creation of its design.

There is no question that the availability factor of boiler units has been considerably less in the starting of new plants than after that period of changes and modification. We therefore can look for a vast improvement in this particular situation if more thought and effort are put into the unit in the beginning.

It is interesting in reading a foreign contract, covering boiler and stoker units, to find that it not only included specifications of design, but also specifications covering the method by which the boilers and stokers would be put in operation. In part the specifications covered the following:

In a modern boiler setting, mechanical defects may show up in starting, such as defective high-pressure fittings, boiler tubes,

[11] General Sales Manager, American Engineering Company, Philadelphia, Pa. Mem. A.S.M.E.

superheater tubes, water walls, stoker parts, etc., so that by a process of elimination all elements will be tried out. It is proposed that in starting no combustion problems will be involved. A very good grade of coal will be selected of which its action in firing is well known, and even though it is a better grade of coal than will ultimately be used at the plant, a knowledge

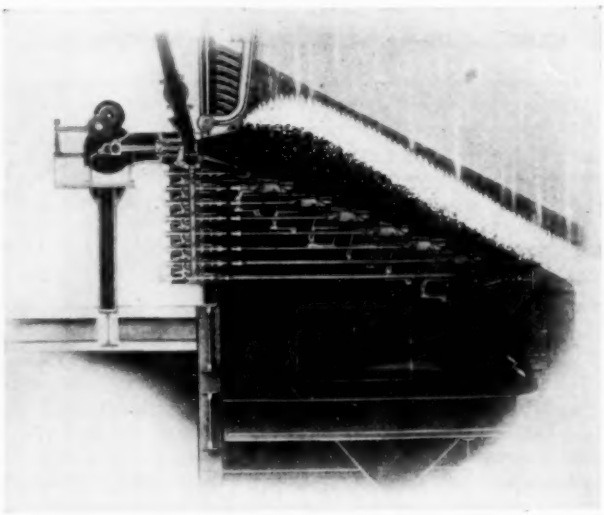


FIG. 13

of this good coal and its use will assure on the start no problems in its burning. If a plant is designed for high preheat, the start will be on cold air planned for a definite period of operation; then a small amount of preheated air will be run for a certain established period. At the start the unit will not be put to test for

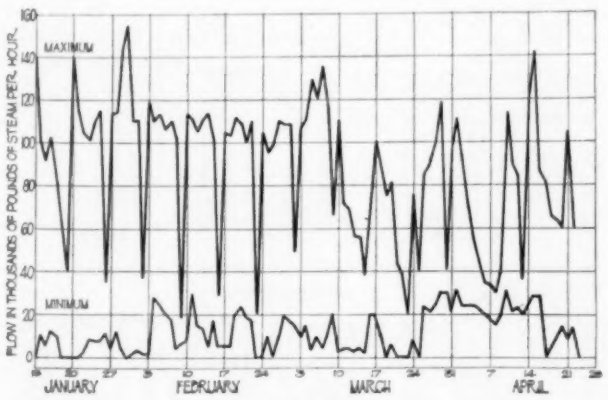


FIG. 14

its ultimate capacity, but rather a normal rating will be rigorously adhered to until it is definitely established that all mechanical parts have stood the test of normal rating. Then the unit will be gradually tested for high capacity. After the unit has stood the test for all mechanical defects, the burning of the various grades of coal that normally will be used at the plant will be started, with the assurance that no mechanical interruptions will interfere with the problems of burning the coal to the best advantage both as to efficiency and rate of burning.

EXAMPLE OF USE FACTOR

The authors make this statement: "The records of availability reported herein for many stoker-fired plants certainly show that such equipment need not have a low availability factor." The writer would like to cite an actual example to confirm this statement of a carefully planned boiler unit that was started and held on the line about four months. This was a modern 45 underfeed stoker applied to a 1000-hp. boiler. It was the one dependable boiler unit in a heating plant. It was absolutely necessary that the unit stay on the line, and it did for about four months. The maximum and minimum steam

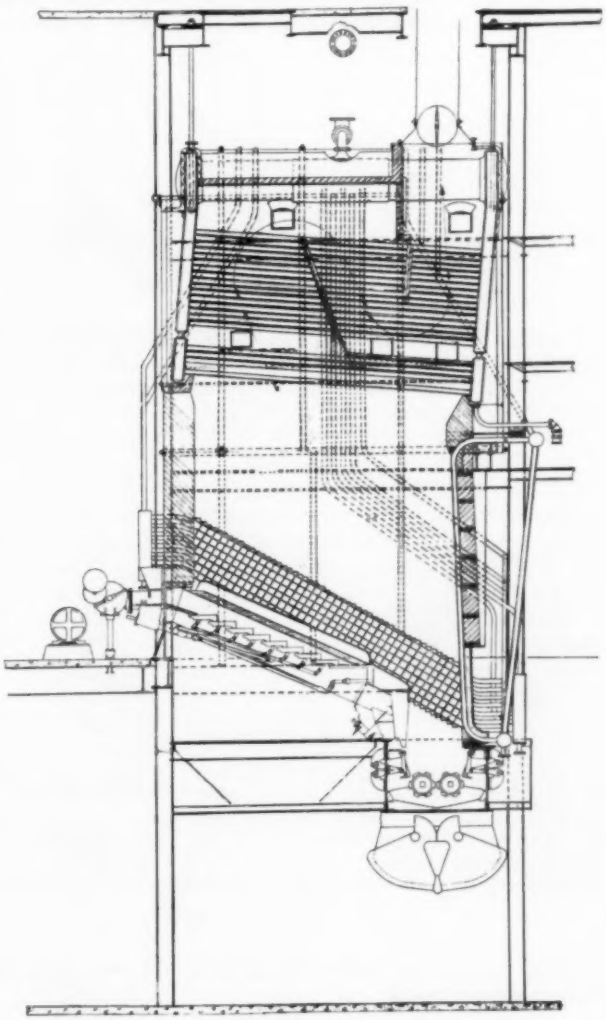


FIG. 15

outputs during this period are shown in Fig. 14. The furnace is equipped with partial water cooling on the two side walls and rear walls, as shown in Fig. 15.

After the unit was taken off the line, a careful inspection was made, and the indications were that the unit could have run much longer, as there were no apparent immediate repairs required for the stoker, water wall, or boiler.

WATER-COOLED WALLS

It is noticed in the paper that, out of 101 stoker-fired plants reported, only 24 plants have water-cooled walls. Of the 62 pulverized-fuel plants reported, 34 have water-cooled walls. In other words, 55 per cent of the pulverized-fuel furnaces are water cooled against 25 per cent of the stoker-fired furnaces.

Many of the new stoker-fired plants now going in have water-cooled furnaces, so that one can look for an improvement in the availability of stoker-fired furnaces in so far as the furnace outages are concerned, especially when reports come in to show at least 50 to 75 per cent water-cooled furnaces for stoker-fired plants.

Even older plants have already sensed the value of the modern furnace construction and are shaping their expansions to take advantage of these new elements by removing old stokers and putting in modern water-cooled walls and modern and larger stokers. It is being shown that not only higher availability

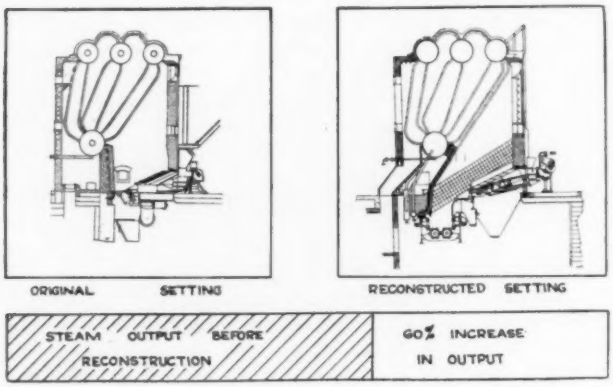


Fig. 16

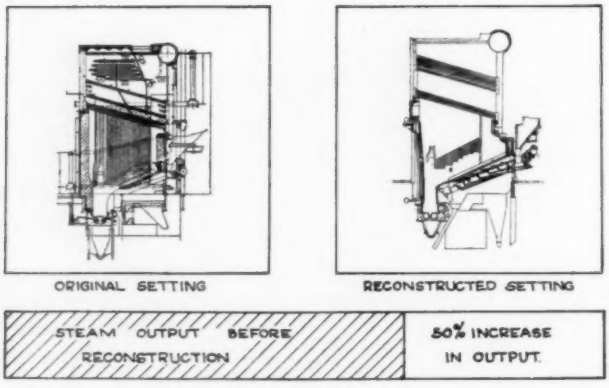


Fig. 17

is being obtained but higher thermal efficiency and greatly increased steam outputs.

Two Plants Typical of the Future

The writer would like to cite one or two important modern jobs that are typical of what we may look forward to as a means of increasing the availability factor of stoker-fired plants.

Plant "A," a steel plant remodeled, had three boilers equipped with old dump-grate-type underfeed stokers. With the old plant, with the three boilers in operation, it was possible to obtain continuously 61,500 lb. of steam per hour, and on a peak 81,600 lb. of steam per hour. It was decided to remodel the boiler plant and revamp the old stokers, changing them to a crusher-type stoker and with increased grate surface. The setting before and after remodeling is shown in Fig. 16. The new boilers are designed to give continuously about 100,000 lb. of steam per hour, and on a peak 130,500 lb. of steam per hour. This will give a continuous increase of about 30,000 lb. of steam per hour, and a peak increase of about 50,000 lb. of steam per hour.

The complete cost of revamping the old stoker-fired boilers was about $65,000. This included new stokers, side and rear water walls, brickwork, excavation, complete erection, and everything ready for steam generation. On this basis of cost the additional steam figures $2.17 per pound of steam continuously and $1.30 per pound of steam for peak loads.

Plant "B," a central-station plant remodeled, is an installation

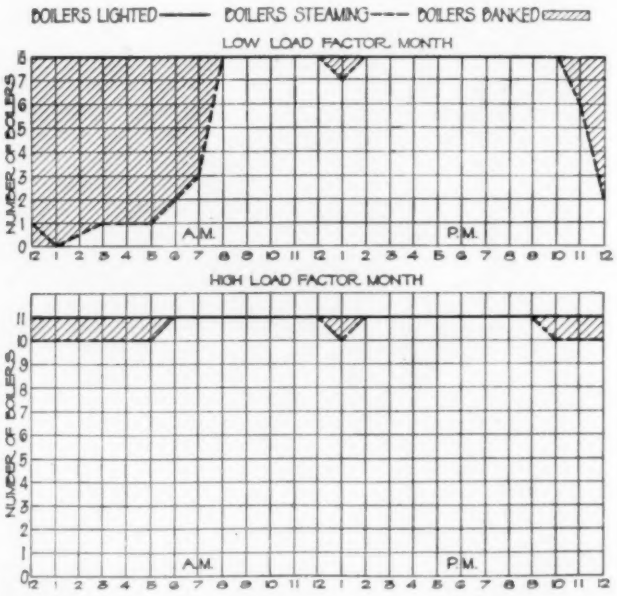


Fig. 18

(Legends: Heavy solid line, boilers lighted; heavy broken line, boilers steaming; single cross-hatching, boilers banked.)

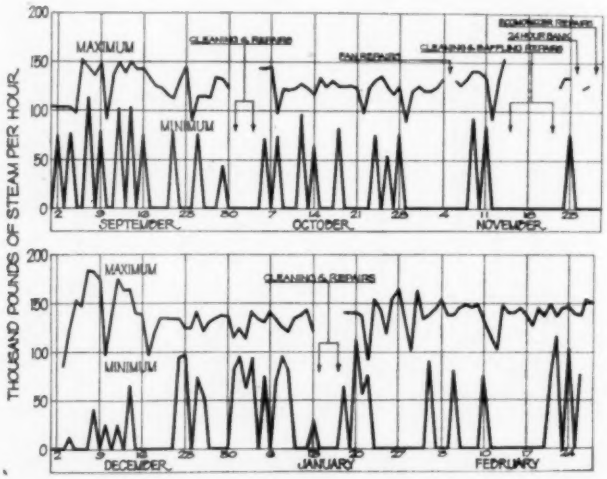


Fig. 19

having 14-retort 33-tuyère underfeed stokers installed under 1710-hp. boilers. One of the settings is being remodeled by removing the 33-tuyère stoker and installing a 53-tuyère stoker, as shown in Fig. 17. The 53-tuyère stoker will have a grate surface of 493-sq. ft. and additional grate area of 115 sq. ft. With the old installation, 200,000 lb. of steam per hour was obtained by burning 78 lb. of coal per square foot of grate surface per hour. With the remodeled setting an evaporation of 300,000 lb. continuously per hour will be obtained at the rate of 72 lb. of coal

per square foot per hour. On a basis of burning 78 lb. of coal per square foot per hour, the new evaporation will be 330,000 lb. of steam per hour. In the new stoker all of the furnace changes will cost approximately $150,000. Taking the 330,000 lb. of steam that the remodeled setting will give over the old 200,000 lb. of steam, the cost of this remodeling job will amount to $1.15 per pound of steam increase.

Banking of Stoker-Fired Boilers

The ability to bank an equipment over long periods with a minimum fuel loss, the authors show, has a marked effect on the "availability factor."

If it is possible to have an active fire on the fuel bed ready to respond to any operating condition, in a great number of plants this has considerably more of a value as to its availability than if the boiler is out.

The large Richmond Station of the Philadelphia Electric Company has recently had an opportunity to gage the full value of this factor, because it had been necessary to operate with a very high load factor while at other times it was necessary to operate with a low load factor. Fig. 18 is a typical boiler schedule for both the high- and low-load factor periods.

Fig. 19 shows a steam output chart for one of the typical boilers of the plant.

Some very interesting figures covering the operation of the Richmond Station show a very high efficiency for both the low- and high-load factor periods.

Results of Richmond Station

Preheated Plants. The authors have given a comparison of the availability factors for units having air preheaters and those operating on cold air.

The operation of the Beacon Street Plant of the Detroit Edison Company is interesting because these units do not have any so-called heat-recovery devices. It has no economizers, no preheaters, no water walls. A typical output chart is shown in Fig. 20. It will be noted that over 350,000 lb. of steam is taken out of the unit on peaks.

The data in Fig. 11 were taken from the operating records kept at the Beacon Street Plant. They illustrate the characteristic operation of the plant.

The boiler efficiencies shown in Table 10 are those derived from the weekly computation and are based on the amount of make-up water used as determined by an accurately calibrated V-notch meter.

The writer has the history of the boiler operation of this plant from the time it started on September 27, 1928, up to the end of the heating season, May 31, 1929. A total of seven days of boiler outage may be charged to stoker repairs during the whole season of approximately 720 boiler days. This amounts to about 0.97 per cent.

This record and similar analyses indicated by the authors show the suprisingly large number of elements affecting boiler outage. Engineers may well devote a great deal of study to the causes of these outages and means for reducing them.

TABLE 10

Period	Steam output as shown by boiler flow meter in thousands of pounds: No. 1 boiler	No. 2 boiler	No. 3 boiler	Total boiler steam flow	Plant steam output, V-notch meter	Quantity coal burned tons	Boiler efficiency for month
Sept.....	....	....	9,753	9,758	8,209	471	78
Oct.....	36,846	10,809	48,916	96,771	80,002	4,427	79.6
Nov.....	42,676	100,261	60,150	203,087	164,789	8,870	81.4
Dec.....	128,106	122,942	8,721	263,769	218,204	12,167	80.5
Jan.....	122,257	114,009	131,189	367,455	203,365	16,323	80.2
Feb.....	89,031	86,245	97,844	273,120	220,679	11,992	80.7
Mar.....	96,430		92,164	188,594	156,940	8,469	82.7
Apr.....	33,710	85,519	31,676	150,905	129,210	6,806	883.2
May.....	11,298	103,941	....	115,239	99,561	5,320	83.5
Total.....	560,354	623,726	480,418	1,164,498	1,370,959	74,845	

NOTE: From 15 to 20 per cent of all the steam leaving the boilers is used for feedwater heating. The balance is the steam output.

Dr. D. S. Jacobus.[12] These papers[13] are of great value in recording progress in the improvement of steam-generating

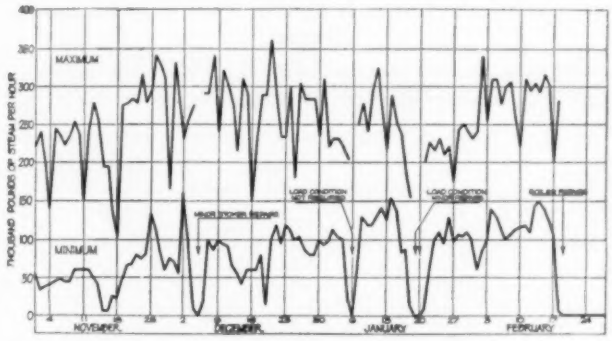


Fig. 20

plants. The trend, as brought out in the papers, has been toward increased size and output, improved firing methods, improved arrangement of the boiler and heating surfaces, improved auxiliary equipment, and in employing greater operating skill.

The higher steam pressures and temperatures which are in use today, and the higher rates of fuel burning and consequent increase in the rates of evaporation, demand a correspondingly higher grade of workmanship and materials and have taken the construction of a steam-generating equipment out of the old-time boiler shop and put it into a high-grade machine shop. All this has resulted in securing a higher reliability as well as an increase in the thermal efficiency of the plant as a whole. It is gratifying to note that the availability for service of large boiler units has become as high as that of the turbine units and that the added efficiency has not been purchased by a counterbalancing increase of investment, so that the advance has been made in an economic sense as well as a purely technical one. This is the result of developments which have been made step by step in reaching the degree of perfection obtained today, and it represents progress made through the cooperation of many men and minds.

The results secured are of interest to all and deserve a most careful study. The continuation of the investigation should have most hearty support.

Authors' Closure

Mr. Baker's mention of the apparent effect of use on availability suggests the presentation of the data as shown in Fig. 21. Use factors are plotted against availability factors without allowance for severity of use. It is interesting to note how closely many of the units hug the limiting diagonal. The predominance of points representing stoker-fired units near the diagonal may possibly be explained by variation in interpretation of definitions. A stoker-fired boiler on bank would undoubtedly be interpreted as "in use," whereas a pulverized-fuel-fired or oil-fired boiler might equally well be interpreted as "in reserve."

Another point raised by Mr. Baker is dealt with in Fig. 22, which attempts to show the effect of complica-

[12] Advisory Engineer, Babcock & Wilcox Company, New York, N. Y. Mem. A.S.M.E.

[13] The discussion also includes the paper by Frank S. Clark on "Effect of Large Boilers at High Capacities on Operating Characteristics and Investment in Boiler Plant."

tion in design on availability. Units having unprotected walls are compared with others which have air or water cooling. The latter class ranges from complete protection to only partial protection of vulnerable spots. Each class is divided into groups which vary from the simple boiler to the more elaborate combination of boiler, economizer, and air preheater.

Improvement in Design Is Keeping Pace With Increased Driving

At first glance the chart seems to favor the more simple designs with unprotected walls, although the difference is very

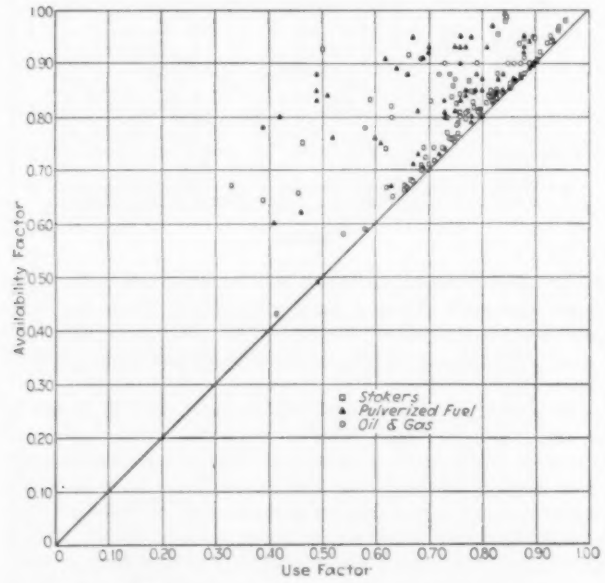


Fig. 21 Effect of Use Factor on Availability

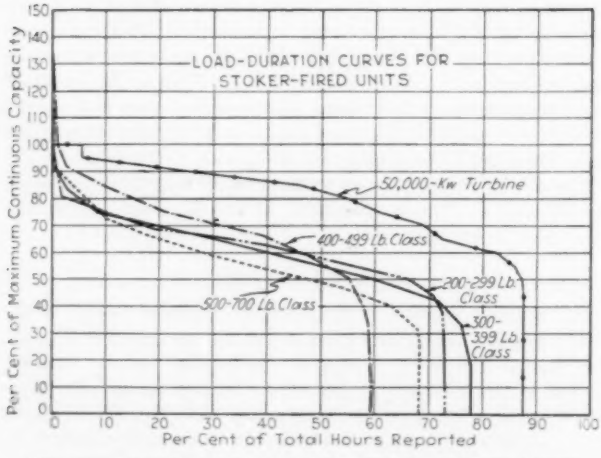


Fig. 23 Load-Duration Curves for Stoker-Fired Units

slight. It seems probable that if severity of use could be taken into consideration the more complicated designs would be shown to be the better from the standpoint of use of investment. At any rate improvement in design seems to be keeping pace with increased driving.

At Mr. Houghton's suggestion the authors have constructed load-duration curves as shown in Figs. 23 and 24, for the various classes of steam-generating units. These are composite curves and represent averages of each class and not extremes. The turbine curve represents the average load duration of the machines in one typical high-load factor, but not base-load station. The maximum continuous capacity for each boiler as used in plotting these curves was reported by the station operators.

Mr. Funk suggests that, if the causes of outage were expressed as percentages of the total time reported, the troubles caused by the various parts of the units would be more readily com-

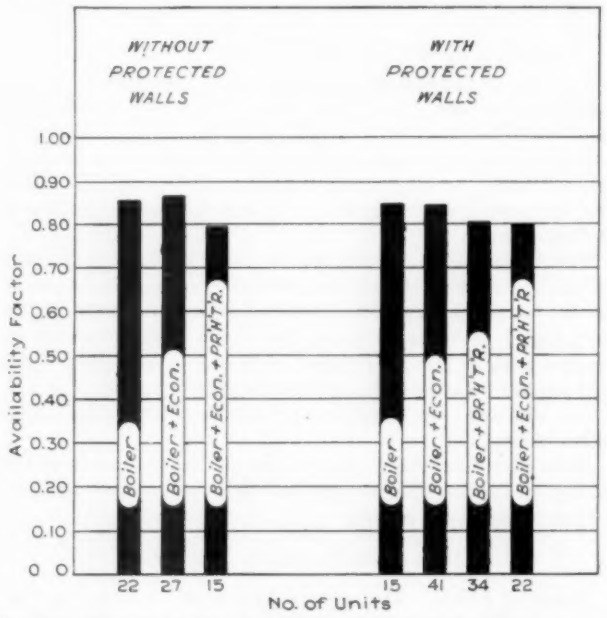


Fig. 22 Effect of Complication of Design on Availability

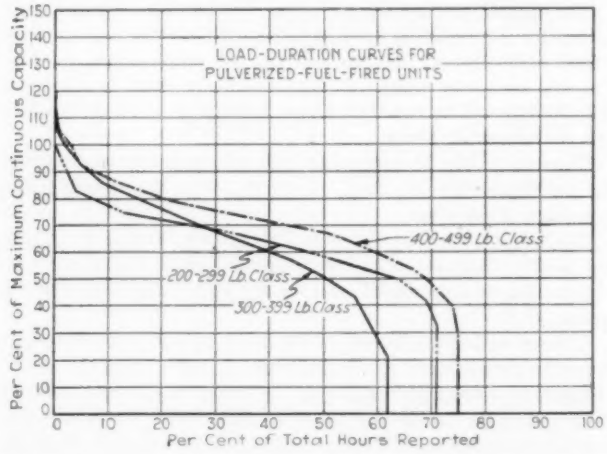


Fig. 24 Load-Duration Curves for Pulverized-Fuel-Fired Units

parable. This change has been made and is presented in Table 11.

The averages by classes are given in Table 12 so as to make them readily comparable.

Impossible to Determine Actual Heat Released

Mr. Craig calls the attention of the authors to the unit used as the abscissa in Fig. 10, pointing out that, if rates of driving with respect to furnace volume are to be compared, the unit used should be based on the heat released in the furnace. This is perfectly true, and such a unit would have been used had the necessary information been available. The data, as collected, did not include quantities of coal burned or efficiencies, so it is impossible to determine the actual heat released in each fur-

TABLE 11 ANALYSIS OF TIME UTILIZATION

OUTAGES CLASSIFIED AS TO CAUSE; EXPRESSED AS PERCENTAGE OF TOTAL PERIOD

Unit number	Scheduled cleaning and overhaul	Changes in equipment	Boiler and superheater	Furnace	Stoker	Pulverizing equipment, burners	Economizers and air preheaters	Ash equipment	Draft equipment	Not chargeable to unit	Total outage
Stoker-Fired Units, 500 to 700 Lb. per Sq. In. Pressure											
S-1	9.7	..	0.8	6.2	0.1	..	4.3	..	..	..	21.1
S-2	11.0	..	0.5	0.3	0.2	..	6.3	..	..	..	18.3
S-3	7.6	..	0.3	..	0.8	..	11.4	..	..	2.9	23.0
S-4	11.2	..	1.2	1.2	0.1	..	9.7	..	..	2.7	26.1
S-5	5.4	..	0.8	1.2	0.1	..	11.5	..	..	..	19.0
S-6	7.3	..	0.3	..	..	..	11.5	..	..	..	19.1
S-7	2.3	..	3.8	11.5	..	..	1.2	..	..	..	18.8
S-8	4.2	..	2.6	7.2	..	..	1.0	..	..	..	15.0
S-9	11.7	..	2.8	6.8	0.3	..	0.7	..	0.8	0.2	23.3
S-10	1.6	..	3.6	7.6	..	..	2.2	..	..	0.2	15.2
S-11	15.1	..	3.1	0.3	7.5	..	5.7	..	1.4	..	33.1
S-12	6.4	..	0.4	8.7	1.4	..	0.8	..	0.6	..	18.3
S-13	5.1	..	0.9	8.5	1.0	..	0.6	..	0.4	..	16.5
S-14	7.6	1.9	1.0	0.3	0.2	..	20.7	..	0.3	..	32.0
S-15	1.2	..	4.9	3.5	2.8	..	1.2	..	0.6	1.0	15.2
S-16	0.9	0.1	6.0	7.0	3.7	..	0.2	1.2	..	0.1	19.2
S-17	3.3	..	5.1	3.0	2.1	..	0.1	0.2	1.5	..	15.3
S-18	3.7	..	5.8	3.5	2.0	..	0.2	..	0.2	0.6	16.0
S-19	1.9	0.1	4.5	2.6	3.2	..	0.1	0.4	..	0.6	13.4
S-20	2.9	..	4.5	2.3	5.4	..	0.6	..	0.1	0.5	16.3
Stoker-Fired Units, 400 to 499 Lb. per Sq. In. Pressure											
S-21	1.4	..	4.8	4.1	1.2	..	4.1	..	1.1	0.8	17.5
S-22	23.6	..	..	0.3	..	..	..	..	..	..	23.9
S-23	25.8	..	0.1	..	..	..	..	..	..	..	25.9
S-24	24.2	..	..	..	..	..	..	..	..	..	24.2
S-25	29.1	..	..	..	..	..	..	..	..	..	29.1
S-26	0.2	..	33.3	12.0	8.1	..	..	..	0.5	2.9	57.0
S-27	2.0	..	7.2	1.3	5.7	..	0.7	..	..	0.4	17.3
S-28	..	..	16.2	1.1	8.1	..	0.3	..	2.7	4.9	33.3
Stoker-Fired Units, 300 to 399 Lb. per Sq. In. Pressure											
S-29	(No analysis of outage)										1.1
S-30	(No analysis of outage)										..
S-31	(No analysis of outage)										0.7
S-32	(No analysis of outage)										1.2
S-33	(No analysis of outage)										1.8
S-34	(No analysis of outage)										1.7
S-35	(No analysis of outage)										1.2
S-36	1.2	..	1.3	0.1	3.1	..	0.7	..	..	..	6.4
S-37	11.8	..	3.0	1.1	13.3	..	..	0.5	0.3	..	30.0
S-38	8.3	..	10.6	1.1	8.8	..	..	0.3	..	..	29.1
S-39	7.2	..	0.3	1.6	6.4	..	10.7	..	..	..	26.2
S-40	18.6	0.9	1.8	0.4	1.6	..	0.7	..	..	..	24.0
S-41	17.3	..	1.9	0.1	5.6	..	2.2	..	..	0.5	27.6
S-42	0.1	..	10.5	4.5	3.3	..	..	..	0.8	0.8	20.0
S-43	2.4	..	..	0.9	..	..	..	..	..	..	3.3
S-44	10.0	5.9	1.3	9.1	..	..	0.3	5.2	0.4	..	32.2
S-45	7.8	5.3	2.4	10.0	..	..	0.5	2.8	0.3	..	29.1
S-46	..	..	0.1	1.8	1.0	..	2.4	..	..	..	5.3
S-47	3.0	..	0.1	0.5	..	..	..	..	..	..	3.6
S-48	3.8	..	7.4	2.1	7.0	..	0.6	2.9	0.9	..	24.7
S-49	14.9	0.5	2.1	0.8	1.6	..	13.3	..	0.5	..	33.7
S-50	1.9	..	0.4	1.3	..	..	0.1	..	..	..	3.7
S-51	2.1	..	0.3	0.9	..	..	..	..	..	..	3.3
S-52	0.3	..	6.4	4.3	3.3	..	..	0.2	0.1	..	14.6
S-53	9.0	..	0.9	2.3	2.1	..	..	..	..	..	14.3
S-54	0.1	..	4.8	2.8	2.5	..	..	..	0.1	0.8	11.1
S-55	0.3	..	4.9	1.6	2.6	..	..	0.1	0.3	..	9.8
S-56	0.8	..	5.9	2.0	2.4	..	..	0.1	0.3	0.3	11.8
S-57	0.9	..	6.1	5.3	2.0	..	..	..	0.2	..	14.5
S-58	0.4	..	7.0	3.1	3.2	..	..	0.2	0.2	0.2	14.3
S-59	1.1	..	6.0	5.9	3.0	..	..	0.1	0.4	..	16.5
S-60	..	..	7.1	4.9	4.2	..	..	..	..	0.8	17.0
S-61	0.9	..	5.5	3.4	3.0	..	..	..	0.1	0.3	13.2
S-62	0.5	..	9.0	3.2	2.6	..	..	..	0.1	0.1	15.5
S-63	0.7	..	7.1	4.4	2.8	..	..	..	0.2	0.9	16.1
S-64	1.8	..	1.8	1.3	..	..	..	..	..	..	4.9
S-65	25.1	..	1.9	0.5	0.9	..	..	..	..	..	28.4
S-66	7.0	..	8.2	6.3	..	..	..	..	..	..	21.5
S-67	25.9	..	..	0.1	..	..	..	..	..	..	26.0
S-68	9.7	..	0.2	..	0.2	..	..	..	..	..	10.1
S-69	9.9	..	1.2	0.6	0.1	..	..	..	..	..	11.8
S-70	2.1	1.0	7.3	6.4	7.9	..	..	0.5	0.5	..	25.7
S-71	4.7	..	9.3	7.1	2.4	..	0.1	0.3	..	3.8	27.7
S-72	24.8	..	1.9	1.9	3.5	..	..	0.7	..	..	32.8
S-73	2.4	..	8.4	3.6	5.6	..	0.1	..	0.1	..	20.2
S-74	2.5	..	3.1	1.2	2.2	..	..	0.5	0.7	..	10.2
S-75	1.5	..	6.5	10.7	13.0	..	..	2.2	1.3	0.4	35.6
S-76	3.5	..	6.3	14.0	6.7	..	..	3.7	..	..	34.2
S-77	2.2	..	1.2	0.3	1.8	..	..	..	0.1	..	5.6
S-78	2.4	..	2.6	1.6	1.2	..	..	..	..	..	7.8
S-79	5.4	..	0.9	1.0	1.8	..	..	0.2	0.2	..	9.5
S-80	4.7	0.6	10.8	1.5	7.4	..	..	..	..	..	25.0
S-81	9.4	..	4.2	0.7	2.5	..	..	..	..	0.4	17.2
Stoker-Fired Units, 200 to 299 Lb. per Sq. In. Pressure											
S-82	12.6	..	0.8	1.4	..	..	..	0.6	0.2	..	15.6
S-83	..	..	2.1	3.2	4.3	..	1.7	2.1	2.1	..	15.5
S-84	..	..	3.2	0.2	5.0	..	..	..	..	..	8.4
S-85	..	..	0.4	0.1	1.2	..	..	0.3	..	..	2.0
S-86	2.3	..	0.8	1.1	0.6	..	..	..	1.5	..	6.3
S-87	1.6	..	2.1	3.7	3.1	..	..	..	1.4	0.3	12.2
S-88	13.6	..	0.8	0.1	..	..	..	0.3	..	..	14.8
S-89	..	..	2.1	0.6	..	..	1.8	..	..	..	4.5
S-90	31.0	..	1.5	..	2.5	..	..	..	..	..	35.0
S-91	2.2	..	0.6	0.5	1.2	..	..	..	..	..	4.5
S-92	4.3	..	2.9	3.1	3.3	..	0.1	..	1.5	2.2	17.4
S-93	4.5	..	2.5	3.5	1.6	..	..	0.3	1.3	0.2	13.9
S-94	23.1	..	0.3	..	6.4	..	..	..	..	..	29.8
S-95	(No analysis of outage)										19.3

TABLE 11 ANALYSIS OF TIME UTILIZATION (*Continued from page 291*)

Unit number	Outages Classified as to Cause; Expressed as Percentage of Total Period: Scheduled cleaning and overhaul	Changes in equipment	Boiler and superheater	Furnace	Stoker	Pulverizing equipment, burners	Economizers and air preheaters	Ash equipment	Draft equipment	Not chargeable to unit	Total outage
Stokes-Fired Units, 200 to 299 Lb. per Sq. In. Pressure (Continued)											
S-96	(No analysis of outage)										14.4
S-97	(No analysis of outage)										7.3
S-98	(No analysis of outage)										10.4
S-99	21.6	..	2.8	..	..	..	8.9	..	..	..	33.3
S-100	11.4	..	3.2	..	0.6	..	0.8	..	..	0.6	16.6
S-101	19.8	..	..	..	0.6	..	..	..	..	..	20.4
Pulverized-Fuel-Fired Units, 400 to 499 Lb. per Sq. In. Pressure											
P-1	0.8	..	4.3	7.0	..	2.5	..	0.8	..	..	15.4
P-2	..	..	1.4	9.9	..	0.9	..	..	..	..	12.2
P-3	1.9	1.5	12.1	1.8	..	1.3	..	0.8	..	..	19.4
P-4	1.1	0.6	2.6	5.2	..	2.2	..	..	0.5	..	12.2
P-5	10.3	..	1.6	2.8	..	..	..	0.5	..	..	15.2
P-6	..	..	8.9	9.9	..	..	..	..	..	..	18.8
P-7	4.5	0.5	7.0	0.7	..	1.1	0.3	0.3	0.2	..	14.6
P-8	2.8	..	10.7	9.2	..	..	0.7	0.9	..	..	24.3
P-9	2.1	..	7.8	..	..	0.7	0.2	4.2	0.1	..	15.1
P-10	4.9	..	0.7	1.3	..	..	0.4	..	..	1.1	8.4
P-11	..	..	22.0	4.5	..	..	..	..	0.2	..	26.7
P-12	3.3	21.7	..	6.9	..	..	1.3	..	..	..	33.2
P-13	0.3	..	0.6	3.1	..	..	..	..	..	1.1	5.1
P-14	5.7	0.1	1.0	0.6	..	..	..	..	4.9	0.3	12.6
P-15	3.7	..	5.3	2.1	..	1.1	..	0.5	0.4	0.3	13.4
P-16	4.1	..	8.8	1.7	..	0.3	..	1.0	0.3	..	16.2
P-17	4.1	..	2.1	..	..	..	..	0.5	..	..	6.7
P-18	2.4	3.5	0.4	..	..	6.1	..	..	..	..	12.4
P-19	0.3	2.1	..	..	..	0.3	..	..	..	..	2.7
P-20	0.1	3.5	0.4	..	..	5.3	..	..	..	..	9.3
Pulverized-Fuel-Fired Units, 300 to 399 Lb. per Sq. In. Pressure											
P-21	5.1	7.8	0.1	0.9	..	5.2	0.6	..	..	0.3	20.0
P-22	1.4	14.0	..	0.4	..	2.6	1.0	..	2.6	..	22.0
P-23	0.5	8.9	4.3	5.2	..	..	0.7	..	4.3	..	23.9
P-24	15.7	..	..	1.3	..	..	3.7	..	..	..	20.7
P-25	0.9	7.2	0.6	..	..	..	6.0	..	..	..	14.7
P-26	0.6	10.8	4.4	2.5	..	0.3	1.6	..	0.3	..	20.5
P-27	7.5	10.5	4.9	1.9	..	..	7.2	..	5.6	..	37.6
P-28	1.4	11.0	2.8	..	..	..	1.7	0.5	..	..	17.4
P-29	6.1	..	0.7	0.6	..	..	..	0.2	1.1	..	8.7
P-30	12.9	..	1.4	..	..	..	0.9	..	..	..	15.2
P-31	2.0	..	1.3	1.4	..	..	0.1	0.1	0.1	3.0	8.0
P-32	0.3	..	0.7	4.8	..	0.5	..	..	0.3	..	6.6
P-33	1.3	..	4.8	2.2	..	0.4	0.2	0.4	0.9	..	10.2
P-34	0.1	..	0.5	4.5	..	..	..	..	0.3	..	5.4
P-35	0.5	..	1.2	4.8	..	..	..	0.8	..	..	7.3
P-36	0.2	..	7.0	7.7	..	1.3	..	1.0	1.5	..	18.7
P-37	13.7	..	5.8	..	..	..	0.5	..	..	..	20.0
P-38	4.3	4.6	3.0	2.4	..	..	..	0.3	0.3	..	14.9
P-39	2.1	..	6.4	4.0	..	2.2	..	0.2	2.0	..	16.9
P-40	0.4	..	2.6	14.2	..	9.1	0.2	..	..	0.6	27.1
P-41	1.5	..	7.9	17.5	..	0.9	..	1.2	..	..	29.0
P-42	3.3	0.1	0.8	0.8	..	..	5.3	..	1.4	..	11.7
P-43	10.3	..	1.4	0.8	..	..	..	0.3	0.1	..	12.9
P-44	..	..	50.0	..	..	..	0.6	..	..	..	50.6
P-45	11.1	..	2.8	3.8	..	0.1	1.6	0.2	0.8	..	20.4
P-46	0.8	4.1	..	1.4	..	..	..	..	..	..	6.3
P-47	2.0	..	2.8	..	..	..	0.5	..	..	..	5.3
P-48	2.2	..	2.6	..	..	..	..	..	..	..	4.8
P-49	3.6	..	2.5	1.1	..	..	..	..	..	..	7.2
P-50	3.3	..	3.7	..	..	..	..	..	..	..	7.0
P-51	1.9	..	8.7	..	..	..	..	..	..	..	10.6
P-52	3.7	..	2.8	..	..	..	..	..	..	..	6.5
P-53	..	..	11.7	..	..	..	1.7	0.9	2.0	0.7	17.0
P-54	..	..	7.8	..	..	11.8	5.3	..	0.6	..	25.5
P-55	..	8.9	2.4	5.0	..	..	0.2	..	..	..	16.5
P-56	2.9	18.0	..	18.8	..	..	..	..	..	0.4	40.1
P-57	8.2	..	0.8	..	..	..	0.3	..	0.1	0.3	9.7
P-58	1.5	0.1	4.4	2.9	..	..	0.6	..	..	..	9.5
P-59	0.1	..	8.3	2.9	..	0.1	1.1	..	..	..	12.5
P-60	1.4	..	5.6	2.8	..	..	..	..	0.4	..	10.2
P-61	2.0	..	3.8	0.2	..	0.1	..	..	4.4	..	10.5
P-62	5.1	0.1	5.6	..	..	..	..	..	0.3	..	11.1
Oil- and Gas-Fired Units											
OG-1	9.0	..	..	..	..	..	..	..	..	..	9.0
OG-2	1.5	..	3.2	0.1	..	..	..	..	0.3	..	5.1
OG-3	7.7	..	3.6	0.5	..	..	..	..	..	..	11.8
OG-4	7.4	..	2.7	1.4	..	..	0.1	..	..	..	11.6
OG-5	3.9	5.7	0.8	0.1	..	..	..	..	..	..	10.5
OG-6	6.4	1.3	0.3	1.6	..	..	..	..	..	..	9.6
OG-7	3.6	..	17.1	0.5	..	..	..	..	..	0.6	21.8
OG-8	8.0	..	7.4	4.4	..	..	..	..	..	0.4	20.2
OG-9	6.9	..	3.3	3.3	..	..	..	..	..	..	13.5
OG-10	9.0	..	..	..	..	..	..	..	..	..	9.0
OG-11	8.0	2.3	..	..	..	..	..	..	..	..	10.3
OG-12	9.7	..	..	..	..	..	..	..	..	..	9.7
OG-13	..	..	9.5	..	..	..	..	..	..	32.4	41.9
OG-14	11.8	..	6.1	..	..	..	..	..	..	22.7	40.6
OG-15	5.4	..	..	..	..	..	..	..	..	..	5.4
OG-16	1.7	..	2.5	1.9	..	..	..	..	..	..	6.1
OG-17	11.1	..	0.1	8.7	..	..	..	..	..	..	19.9
Miscellaneous Combinations											
M-1	6.5	..	1.5	0.3	..	0.3	0.3	..	1.1	..	10.0
M-2	4.8	..	0.3	0.4	..	1.0	0.3	..	0.1	..	6.9
M-3	9.4	..	0.5	0.6	..	0.2	0.6	..	0.5	..	11.8
M-4	1.6	..	1.7	..	..	..	..	..	..	..	3.3
M-5	0.9	7.9	1.1	6.2	..	..	..	..	..	..	16.1
M-6	..	..	1.1	4.3	10.1	..	..	..	..	..	15.5

nace. It was therefore necessary to use a unit which represents only approximately that which would have been used if available.

The authors regret that insufficiency of data prevents them from going further into the various lines of study suggested by other discussers. The discussion as a whole has been very gratifying, and it has shown an interest in the study sufficiently great to warrant its continuation. The authors have therefore resumed the collection of boiler-outage data. They hope to be able to continue for at least two more years, and have added more power stations to the list, including several industrial plants.

With this increase in scope, and profiting by the experiences of the first year and the suggestions made in the discussion, it is believed that the results will become increasingly conclusive.

TABLE 12 OUTAGES CLASSIFIED AS TO CAUSE AND METHOD OF FIRING

(Average Outages Expressed as Percentages of Total Period)

	Stokers	Pulverized coal	Oil and gas
Scheduled cleaning	7.8	3.6	6.9
Changes in equipment	1.8	6.3	4.3
Boiler and superheater	3.9	5.1	3.5
Furnace	3.2	4.3	2.3
Stoker	3.3	...	..
Pulverized equipment and burners	...	2.4	..
Economizers and preheaters	3.5	1.5	0.1
Ash equipment	1.0	0.7	..
Draft equipment	0.7	1.3	0.6
Not chargeable to unit	1.1	0.8	14.0

FSP-52-35

Working Stresses for Steel at High Temperatures

By D. S. JACOBUS,[1] NEW YORK, N. Y.

Certain of the published results for the stress to produce creep at higher temperatures have led to apprehension in the employment of stresses that have been used, and it is the object of the paper to dispel such apprehension for the particular stresses which are discussed.

The results secured in practical experience are used for setting the maximum working stresses for drums and superheater headers up to temperatures of 800 deg. fahr., and for temperatures above 800 deg. fahr., the working stresses are scaled down in proportion to the falling off of the stresses that produce creep. The first curve of creep stresses published by Mr. H. J. French of the Bureau of Standards was used, and the working stresses for a given temperature are taken at 2/3 the creep stress. The effect on the stress of piercing a boiler shell with holes and of transmitting heat through the shell is discussed, as well as the tendency toward equalization of stress through extension of the most-strained fiber. The tendency of the creep to equalize the stresses is also brought out.

The method of extrapolation used by different authorities is discussed, and the conclusion is reached that more exact results are obtained by using the method employed by Prof. F. H. Norton in his book on the "Creep of Steel at High Temperatures" than by some other methods.

THE safe working stresses for the shells of boilers and pressure vessels have been determined to a greater extent by experience in the use of such vessels under actual operating conditions than by any theoretical rules. These stresses are ordinarily based on the average stress in the ligaments between tube holes or the like. This is an approximate method. On the one hand the stresses in the metal at the sides of a hole in a drum which is stressed within its elastic limit may be twice or over the average stress in the ligaments or in the solid plate where a single hole is placed in a seamless shell. On the other hand the ductility of the steel causes the stresses to be different after a drum is placed in service than when subjected to the initial pressure, the general effect being to cause the stresses to be more uniform under working conditions than on the initial application of pressure.

That the service conditions have much to do with the pressure at which a vessel can be safely operated may be seen on considering a seamless cylindrical drum having a hole drilled in its shell. If such a drum is subjected to a gradually increasing pressure, the stress at the sides of the hole of, say, twice the average stress in the plate will cause the stress in the metal near the hole to exceed the elastic limit when a certain pressure is reached. On increasing the pressure still further, the yielding of the metal will result in a permanent set in the metal and will cause a redistribution of the stresses in the metal. If the pressure is relieved, there will be stresses near the hole due to the set in the material, and if the pressure be high enough, these stresses are such that on applying and relieving the pressure a number of times the material will fail at the hole through a fatigue action. The safe working pressure for a drum of this sort would therefore be much higher for service conditions where it would be subjected to a reasonably steady pressure or where the pressure would be only occasionally applied and released than it would be if used under a continually fluctuating pressure.

Another question to consider is the distribution of strains due to the thickness of the walls of a vessel which causes the stresses on the inside of the shell in a thick-walled vessel to be greater than those on the outside when the stresses are within the elastic range of the material. With ductile material the stresses will be more or less equalized from the inside to the outside of a thick drum on increasing the pressure to a point where the proportional limit of the most highly strained parts of the material is exceeded. On relieving the pressure there would be tensile stresses on the outside of the shell and compressive stresses on the inside of the shell. An overstrained drum of the sort could be operated more safely at a steady or at a reasonably steady pressure than at a continually fluctuating pressure, and in this respect the influence tending to metal fatigue would be similar to the action at the sides of a hole which has been referred to.

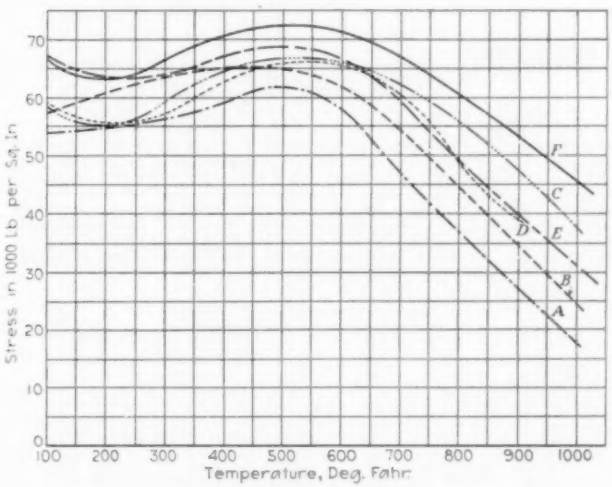


FIG. 1 TENSILE STRENGTHS FOR CARBON STEELS

(*A*, 0.13-C steel, White, Eng. Res. Bull., Univ. of Mich., 1928; *B*, 0.17-C rolled bar, *Valve World*, August, 1928, p. 269; *C*, 0.17-C firebox steel, French, B. S. Tech. Papers, 219, p. 692; *D*, 0.19-C firebox steel, French, B. S. Tech. Papers, 219, p. 691; *E*, 0.24-C steel, Proc. A.S.T.M., vol. 25, p. 39, fig. 3; *F*, 0.25-C marine steel, French, B. S. Tech. Papers, 219, p. 690.)

Another factor which results in strain in the material of a shell which must be considered in establishing the working stress is the effect of heat applied to the outside of the shell and transmitted to the inside of the shell. This is important in boiler tubes, especially at the higher working pressures, and in any vessel which is subjected to heat while under pressure. In this case, as in the others, straining the material results in equalization of the stress from the outside to the inside when subjected to heat, and on allowing the tube or vessel to cool down counter-stresses will be set up, so that a fluctuating tempera-

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

ture has a similar effect on reducing the safe working stress as a fluctuating pressure.

All this shows that the safe working stresses should be based on the results secured through experience and practical service rather than on any theoretical ground.

Experiments of a comparatively recent date have shown the

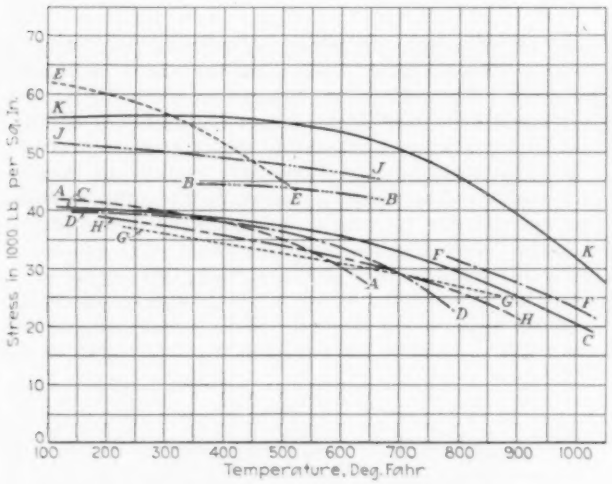


FIG. 2 YIELD POINTS FOR CARBON STEELS

(*A*, 0.10-C steel bar, MacPherran, Proc. A.S.T.M., 1921, p. 855; *B*, 0.14-C rolled bar, Lea, Proc. Inst. Mech. Engrs., December, 1924, p. 1064; *C*, 0.20-C steel, Inberg and Sale, Proc. A.S.T.M., vol. 26, p. 37; *D*, 0.24-C boiler plate, French, B. S. Tech. Papers, no. 362, p. 250; *E*, 0.32-C rolled bar, Lea, Proc. Inst. Mech. Engrs., December, 1924, p. 1064; *F*, 0.35-C steel normalized, *Valve World*, August, 1928; *G*, 0.37-C steel normalized, *Engineering*, 1925, p. 518; *H*, 0.37-C steel, Lynch et al., Proc. A.S.T.M., vol. 25, p. 14; *J*, 0.41-C steel unannealed, MacPherran, Proc. A.S.T.M., 1921; *K*, 67,000 t.-s. steel, Lea, *Engineering*, 1920, p. 295, Fig. 17.)

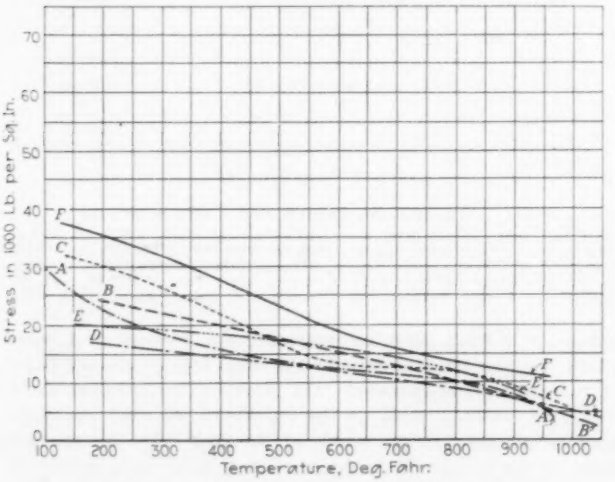


FIG. 3 PROPORTIONAL LIMITS FOR CARBON STEELS

(*A*, 0.013-C steel tube, White and Clark, Eng. Res. Bull, Univ. of Mich., 1928; *B*, 0.18-C steel, French, Proc. A.S.T.M., vol. 24, p. 62; *C*, 0.20-C steel, Inberg and Sale, Proc. A.S.T.M., vol. 26, p. 37; *D*, 0.24-C boiler plate, French, B. S. Tech. Papers, no. 362, p. 269; *E*, 0.24-C boiler plate, French, Proc. A.S.T.M., vol. 25, p. 39; *F*, 0.37-C steel normalized, Lynch and McVetty, Proc. A.S.T.M., vol. 25, p. 14.)

importance of considering the plastic flow of steel when under stress at high temperatures. Much work has been done in determining the rate of extension or "creep" at different temperatures. Certain of the published results derived from these experiments have led to some apprehension in the employment of stresses that have been considered safe, and it is the object of this paper to overcome such apprehension for the particular stresses which are discussed.

It must be remembered that it was necessary to set working stresses for higher temperatures long before the results of creep tests were available. The working stresses given in Table 1 for temperatures ranging from ordinary room temperatures up to 800 deg. fahr. were so set and the results secured in practice have shown that the stresses which were so set are safe.

Over ten years ago it was necessary to set the working stress for an oil still to be made by The Babcock & Wilcox Co. with riveted drums of $1^3/_4$-in. plate for a temperature of 825 deg. fahr. The steel selected was a boiler steel in conformity with the specifications of the A.S.M.E. Boiler Code, having a minimum tensile strength of 55,000 lb. per sq. in. After a study of the results obtained in oil-still and superheater practice and of all available data on the ultimate strength, yield point, and proportional limits of the steel determined by quick time tests, the working stress was set at 7180 lb. per sq. in., based on the working pressure which was carefully regulated, and 7950 lb. per sq. in., based on the setting of the safety valves. A number of stills were built for these stresses and operated satisfactorily.

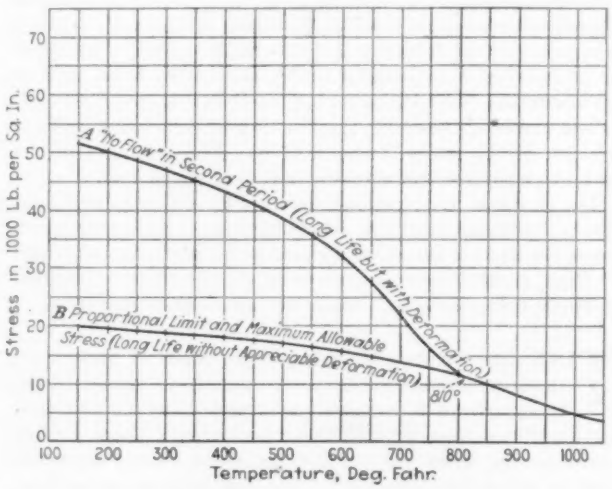


FIG. 4 CREEP AND PROPORTIONAL LIMIT VALUES FOR CARBON STEEL

(*A* and *B*, 0.24-C steel, H. J. French, B. S. Tech. Papers, no. 296.)

From the results of this experience and that secured in superheater and boiler practice a working stress of 8000 lb. per sq. in. at 800 deg. fahr. was established, and also the stresses for temperatures lower than this.

In establishing the working stresses for higher temperatures than 800 deg. fahr., which was done at a later date, the creep-test results then available were made use of, and the figure of 8000 lb. per sq. in. for a temperature of 800 deg. fahr. was scaled down in proportion to the falling off of the stresses to produce creep. The first curve of creep stresses published by Mr. H. J. French of the Bureau of Standards was used, and the working stress for a given temperature taken at $^2/_3$ of the creep stress shown by Mr. French's curve.

It should be borne in mind that the effect of creep is to equalize the stresses and to make the conditions better in this respect.

The stresses determined in the way described are given in Table 1, together with the stresses used in the design of superheater headers for an A.S.M.E. Boiler Code steel having a minimum tensile strength of 62,000 lb. per sq. in.

Fig. 1 gives curves showing the ultimate strength of various steels determined by quick time tests at different temperatures. Fig. 2 gives the yield points by quick time tests, Fig. 3 the proportional-limit points by quick time tests, and Fig. 4 the first curve of creep stresses published by Mr. French.

Much has been done in establishing creep stresses since we set the stresses given in Table 1. Professor F. H. Norton of the Massachusetts Institute of Technology conducted a most accurate and painstaking investigation, the results of which are published in his book entitled, "Creep of Steel at High Temperatures." He made a great many tests of long-time duration on 17 different alloy steels at temperatures ranging from 1000 to 1500 deg. fahr. The chemical analyses, physical properties, and photomicrographs to show the structure of the different steels are given in his book.

TABLE 1 WORKING STRESSES FOR DRUMS OF BOILERS AND PRESSURE VESSELS AND FOR SUPERHEATER HEADERS

	Stresses in lb. per sq. in. based on the ultimate strength of the steel and the creep stress for A.S.M.E. boiler code steel having a tensile strength of 55,000 to 65,000 lb. per sq. in.					Working stress in round figures in lb. sq. in.	
Temp., deg. fahr.	Ultimate strength by quick time test	Creep stress	1/5 Ultimate strength	2/3 Creep stress	Working stress	For drums, min. t.s. 55,000	For superheater headers, min. t.s. 62,000
650	Over 55,000	27,000	Over 11,000	18,400	11,000	11,000	12,400
700	Over 55,000	21,000	Over 11,000	14,000	11,000	11,000	12,400
750	52,000	16,000	10,400	10,700	10,000[a]	10,000	11,500
800	45,000	12,000	9,000	8,000	8,000	8,000	10,000
850	40,000	10,000	8,000	6,700	6,700	6,750	7,500
900	38,000	8,000	7,600	5,300	5,300	5,500	5,500
950	30,000	6,000	6,000	4,000	4,000	4,000	4,000

[a] Shaded down to allow for the falling off of the yield point and the proportional limit.

Fig. 5 and Table 2 are taken from Professor Norton's book, showing the results of his experiments on three carbon steels at temperatures of 900 and 1000 deg. fahr., and some tests that he made as a continuation of his research on the alloy steels.

The results obtained by a number of experiments for carbon steel are shown in Figs. 6 and 7. These figures and the names of the experimenters and references and brief description of their work given in Table 3 and in the Bibliography are taken from Professor Norton's book. There is a wide discrepancy in some of the results obtained by different experimenters, depending on the test methods, duration of the tests, and the method of interpreting results. There is need of more exact data for temperatures below 900 deg. fahr. It is hoped that correspondingly accurate experiments to those of Professor Norton will be made for a lower range of temperature.

It will be well to consider the accuracy of the figures given in Table 1 in view of the data which became available after it was prepared. A paper by Messrs. French, Cross, and Peterson published by the Bureau of Standards, No. 362, contains a curve for 0.24-C steel which gives lower values for the creep stresses than the first curve published by French. The curve is on the basis of the stress producing 0.1 per cent or less elongation in 1000 hours or 1 per cent elongation or less in 10,000 hours. The statement is made that it was only possible to estimate the maximum stresses which will permit certain selected life with certain deformation, and that this explanation was made to avoid misunderstanding considering the accuracy of the data or the method of arriving at the numerical values. The method of extrapolation used in obtaining the results was different from that employed by Professor Norton, and if Professor Norton's method is applied to the experimental results shown for a 0.24-C steel at 810 deg. fahr. in Fig. 10 of the paper, the stress to produce 1 per cent of creep in 10,000 hours would be 12,500 lb. per sq. in., whereas curve No. 5 in Fig. 21 of the paper gives a stress of about 9000 lb. per sq. in. for this temperature. The figure of 12,500 lb. per sq. in. for the stress to produce 1 per cent of creep in 10,000 hours corresponds closely to the creep stress of 12,000 lb. per sq. in. given in Table 1.

The results of Professor Norton's tests given in Fig. 5 and the data contained in Table 2 give a stress of 10,500 lb. per sq. in. to produce a creep of 1 per cent in 10,000 hours for a 0.2 per cent carbon steel at 900 deg. fahr., which is considerably higher than the creep stress given by Mr. French's first curve as contained in Table 1 and shows that the creep stresses given in Table 1 may be considered as conservative.

The method of extrapolation used by Professor Norton is shown in Fig. 5. The stresses to produce a given rate of creep at a certain temperature are plotted against the rate of creep on logarithmic paper, the results in Professor Norton's experiments giving nearly a straight line. This method of extrapolation is more accurate than one employing a widely curved line. The many tests and the wide range and consistency of his experiments show that the results obtained by his method of extrapolation to the extent that he used it are reliable.

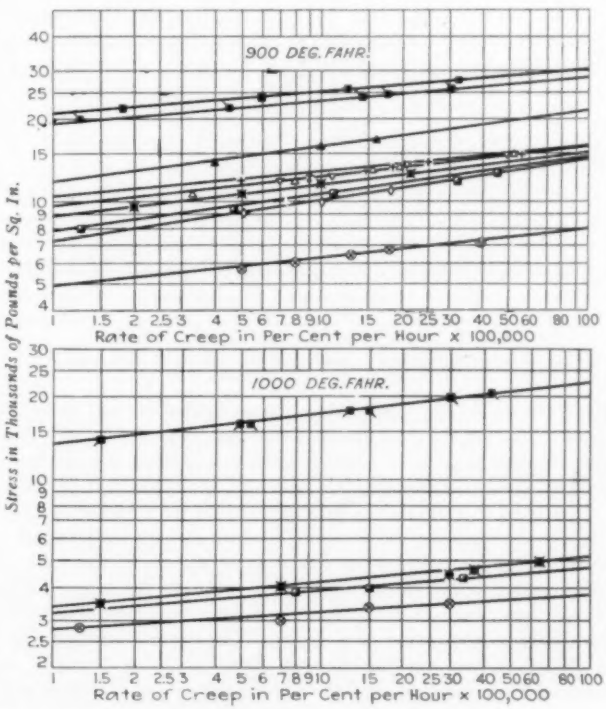


Fig. 5 Runs at 900 and 1000 Deg. Fahr.

TABLE 2 CREEP STRESS[2]

Steel	Symbol	Life with 1% elongation 100,000 hr. 900°F.	1000°F.	10,000 hr. 900°F.	1000°F.
E	■	20,500	....	25,000	....
E (rerun at 1000°F.)	◥	19,500	....	23,500	....
E (rerun at 1100°F.)	◧	....	13,500	....	17,500
E (rerun at 1200°F.)	◪	....	13,500	....	17,500
G	▲	12,000	....	16,000	....
M	+	10,500	....	13,000	....
P	▽	9,600	....	12,500	....
O	△	9,600	....	12,500	....
0.42% carbon steel	✖	8,800	3,500	11,500	4,200
0.20% carbon steel	◘	7,800	3,200	10,500	3,900
0.08% carbon steel	⊕	4,900	2,800	6,300	3,300
I	◊	7,200	....	10,000	....

The results obtained by Professor Norton, employing his method of extrapolation, would indicate that a stress of 2/3 that required to produce 1 per cent of creep in 10,000 hours would be equivalent to from 80 to 90 per cent of the stress required to produce a rate of creep of 1 per cent in 100,000 hours and that the stress would correspond to 1 per cent of elongation

[2] For composition of steels designated by the letters in this table see Table 4.

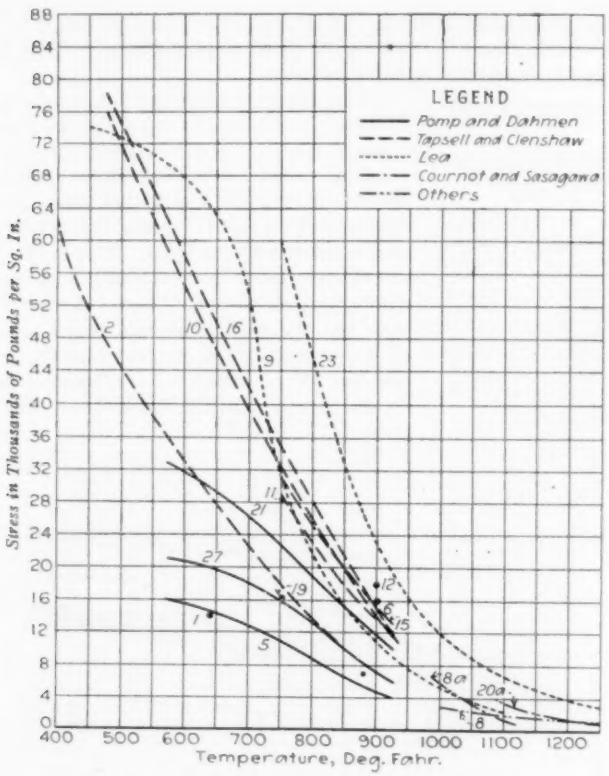


FIG. 6 CREEP VALUES FOR CARBON STEELS
(For names of experimenters and brief description of their work, see Table 3 and Bibliography.)

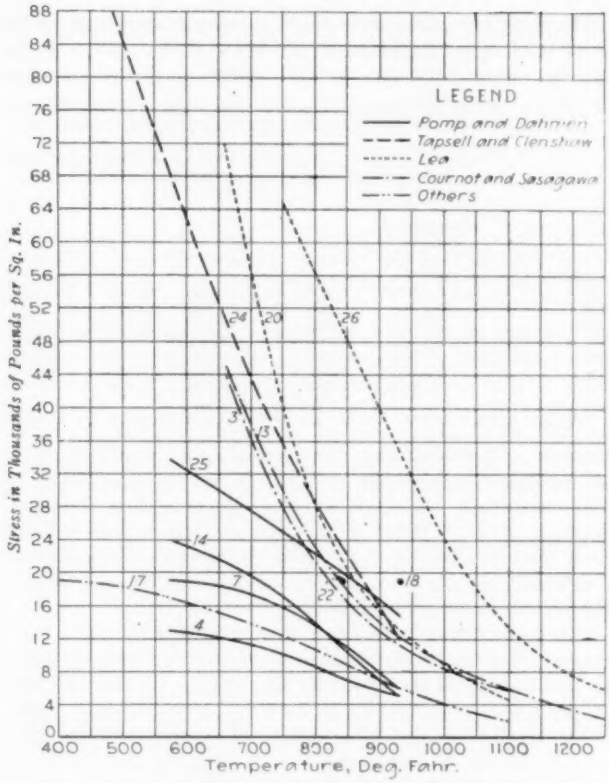


FIG. 7 CREEP VALUES FOR CARBON STEELS
(For names of experimenters and brief description of their work, see Table 3 and Bibliography.)

due to creep in 25 to 50 years. These figures are not given as absolute values, but to show that a reduction in the stress below that required to produce 1 per cent of creep in 100,000 hours has a marked effect in increasing the time for a given amount of creep.

It is hard to formulate a rule that should be generally applicable for determining the working stresses at high temperatures. A variation in temperature has a great effect on the rate of creep, and in some cases it is hard to estimate the temperature of the metal, such as in those where heat is transmitted through the metal to the inside of a vessel. The general method used in obtaining the working stresses given in Table 1 might be followed for conditions similar to those for drums of boilers and pressure vessels and for superheater headers, viz.:

TABLE 3

No.	Composition	Authority	Bibliography
1	Electrolytic iron	Chevenard	1
2	Armco iron	Tapsell	11
3	0.04-C steel	Cournot & Sasagawa	4
4	0.05	Pomp & Dahmen	7
5	0.06	Pomp & Dahmen	7
6	0.08–0.12	Jasper	10
7	0.11	Pomp & Dahmen	7
8	0.13	White & Clark	6
8a	0.13	Clark & White	12
9	0.14	Lea	3
10	0.17	Tapsell & Clenshaw	8
11	0.20	Tapsell & Clenshaw	8
12	0.20–0.30	Jasper	10
13	0.22	Cournot & Sasagawa	4
14	0.23	Pomp & Dahmen	7
15	0.23	Tapsell & Clenshaw	8
16	0.24	Tapsell & Clenshaw	8
17	0.24	French, etal.	9
18	0.30	Dickenson	2
19	0.32	Lea	3
20	0.37	Lynch, et al.	5
20a	0.38	Clark & White	12
21	0.40	Pomp & Dahmen	7
22	0.45	Dickenson	2
23	0.45	Lea	3
24	0.51	Tapsell & Clenshaw	8
25	0.58	Pomp & Dahmen	7
26	0.75	Lea	3
27	1.00	Pomp & Dahmen	7

For the higher temperatures where the steel is distinctly plastic, use $^2/_3$ of the stress required to produce a rate of creep not to exceed 1 per cent in 100,000 hours. In case of doubt, use the most conservative figure in view of the many uncertain elements.

For the lower temperatures use the same working stress as at ordinary room temperatures, provided the ultimate strength determined by a quick time test at the particular temperature divided by the factor of safety is as great as the working stress for ordinary room temperatures, and also provided the stress does not exceed $^1/_3$ of that required to produce a rate of creep not to exceed 1 per cent in 100,000 hours.

In the figures given in Table 1 the working stresses were shaded down for 750 deg. fahr., which comes at a point where a horizontal straight line on plotting the working stresses would meet a curve which decreases as the temperature increases. This shading down was done in order to allow for the falling off of the yield point and the proportional limit at these points, with the idea that this feature is important in a riveted structure.

It may be claimed that the yield points and proportional-limit points should enter any rule for establishing the working stresses. For ordinary temperatures it is common practice to base the working stresses on the ultimate strength of the material rather than on the yield point or the proportional-limit point and to use the yield point only in specifying the physical properties of the material to be employed. There have been proposals to base working stresses on the yield point, or on the so-called elastic limit or proportional-limit points (in fact in some cases the working stresses are obtained in this way), but it still remains common practice to base the stresses on the ultimate strength of the material. An objection to basing the working stresses on the yield point, or what for many years was known as the elastic limit, or on the proportional-limit point, is that there are apt to be wide variations in the results for these points determined by different laboratories for the identical steel,

whereas there are not as wide variations in the results for the ultimate strength.

The many creep tests that have been and are being made are of great value to the industry. Higher and higher metal temperatures are being employed in some fields of work, and the results secured by creep tests assume more and more importance. A greater fund of experience is also becoming available as a guide to the dependability of working stresses at high temperatures, and combining this experience with the results secured by creep tests will do much to rationalize the problem.

It is hoped that the working stresses used for steel at high temperatures by different parties will be submitted in order that we may all get together in establishing a rule that will represent good practice.

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1 P. Chevenard, "Comptes Rendus," 175, p. 486. This short article gives creep data on nickel and its alloys at one temperature.

2 J. H. S. Dickenson, "The Flow of Steels at a Red Heat," *Engineering*, 114, p. 326, p. 378. The paper considers the creep of steels, giving a description of the apparatus used and data at one stress on a number of steels.

3 F. C. Lea, "Effect of Low and High Temperatures on Materials." *Proc. Inst. Mech. Engrs.*, 2, p. 1053. Very complete tests are made on the various mechanical properties of steel over a wide range of temperatures. This is one of the first investigations where the value of long-time tests is realized. The influence of strain hardening is also discussed.

4 J. Cournot and K. Sasagawa, "Contribution à l'Étude de la Viscosite des Alliages à Temperature Elevée." *Rev. Met. Mem.*, 22, p. 753, also "Comptes Rendus," 181, 0. 661. The flow of several steels is studied at various temperatures. The specimens are in the form of wires, and the duration of the test is not long, but the extension is measured precisely. The apparatus and method of test are fully described.

5 T. D. Lynch, N. L. Mochel, and P. G. McVetty, "The Tensile Properties of Metals at High Temperatures." Proc. A.S.T.M., 25, 11, p. 5. Data on the tensile properties of various materials at temperatures up to 930 deg. fahr. are collected. Some data on long-time tests are given.

6 A. E. White and C. L. Clark, "Properties of Boiler Tubing at Elevated Temperatures Determined by Expansion." Trans. A.S.S.T., 11, No. 1, p. 73. The object of this investigation was to determine the safe working loads for low-carbon steel tubes from 900 deg. to 1500 deg. fahr., but the data are rather incomplete. It is pointed out that at the higher temperatures creep occurs much below the proportional limit.

7 A. Pomp and A. Dahmen, "Entwicklung eines Abgekurtzten Prufverfahrens zur Ermittlung der Dauerstandtfestigheit von Stahl bei Erhöhten Temperaturen." Mitt. a. d. Kaiser-Wilhelm-Inst. f. Eisenforchung zu Dussel, IX, no. 3. This is an extensive article on creep measurements. The work of others is reviewed, with a description of their testing apparatus. Creep data on several steels are given, although the length of test at one load was only a few hours. The limiting creep stress is compared with the elastic limit.

8 H. J. Tapsell and W. J. Clenshaw, "Properties of Metals at High Temperatures." 1, Mechanical properties of Armco iron, 0.17 per cent carbon steel, and 0.24 per cent carbon steel with special reference to creep. Department of Scientific and Industrial Research, Engineering Research, special report No. 1, published under authority of His Majesty's Stationery Office. Comprehensive tests are described on the mechanical properties of low-carbon steels. The value of creep tests is discussed, and considerable data are given. A good list of references and creep results by other experimenters are given.

9 H. J. French, H. C. Cross, and A. A. Peterson, "Creep in Five Steels at Different Temperatures." B. S. Tech. Papers, No. 362. Creep tests are made on five types of steel up to 1350 deg. fahr, with more than the usual understanding of the difficulties involved.

10 T. McLean Jasper, "Strength of Steels at Elevated Temperatures, With Particular Reference to Factor of Safety." *Power*, 67, p. 446. This article discusses safety factors and design for high-temperature steel structures. Creep values for two low-carbon steels are given.

11 H. J. Tapsell, "Properties of Materials at High Temperatures," 3, Note on the "creep" of Armco iron. Department of Scientific and Industrial Research, Engineering Research, special report No. 6 published under authority of His Majesty's Stationery Office. The creep of iron is studied particularly at temperatures below 400 deg. fahr. Strain hardening and temperature softening are discussed in relation to creep.

12 C. L. Clark and A. E. White, "The Stability of Metals at Elevated Temperatures." Eng. Res. Bull., Univ. of Michigan. The authors compare the creep stress in long-time tests with the proportional limit in short tests. At low temperatures the agreement is good, but at high temperatures they find no definite limiting creep value. They believe that at a certain temperature the strength of the crystals is equal to that of the boundaries. Above this temperature the amorphous boundaries are the weaker and below it the crystals yield first.

Discussion

A. E. White.[3] Much has been written on this subject, but most of it has been of a character such that it has been necessary for engineers and designers to use their own judgment as to what stresses to employ. One finds in this paper very definite recommendations as to working stresses.

The entire question of the characteristics of metals at high temperatures is exceedingly involved. The results of the findings of many investigators are widely divergent, as evidenced by the variation in values obtained by various investigators as given in Figs. 6 and 7 of the paper. When one recognizes that creep values for carbon steels as given by various investigators in Fig. 6 vary at 800 deg. fahr. from a little above 8000 lb. per sq. in. to 44,000 lb. per sq. in., or, as shown in Fig. 7, at the same temperature from 8000 lb. to 56,000 lb., one can realize the discrepancies which exist. These differences are, without doubt, due in slight degree to differences in composition, but to a greater degree to the different methods employed in the testing of the metals and to the methods employed in arriving at the results.

The writer is in general accord with the working stresses given in Table 1. This conclusion he has reached because of the following:

1 The creep values given in the table would be among the most conservative of the creep values given in Figs. 6 and 7.

2 Though Professor Norton, who did the work of procuring the creep values, did not carry his findings beyond 400 hours, yet the writer feels they represent a high degree of accuracy because of the many tests made and the wide range and consistency of the experiments.

3 To date the writer knows of no failures which have resulted when working stresses given in Table 1 have been used.

C. L. Clark.[4] A paper of this type is very valuable in that it allows results obtained in actual commercial practice to be compared with those obtained in the laboratory. Since the stresses to which the metals in actual use are subjected are not simple tension, as is the case in the laboratory test, and since the temperature and stress variations in actual use are much greater than those encountered in the refined laboratory test, it is surprising that there is such a close agreement between the creep values reported by Professor Norton and the safe working stresses previously adopted by the Babcock & Wilcox Company.

Care must be exercised in the selection of safe working loads for periods of 10,000 or 100,000 hours on the basis of laboratory tests lasting only a relatively few hours. It is obvious that with the large number of steels to be investigated and the wide temperature range to be covered, each test cannot be continued for thousands of hours. They should be continued for a sufficient length of time, however, so that the true shape of the time-elongation curve can be determined. The values reported in this paper were obtained as the result of tests lasting 400 hours, and accordingly should be more representative of actual conditions than those obtained over a period of a few days.

The creep apparatus at the University of Michigan has been recently modified so that the elongation of the test specimen can be determined directly to 2.8 millionths of an inch. Results obtained from a test on a certain ferrous alloy at 1000 deg. fahr. show that, for at least the first 300 hours, the elongation is not directly proportional to the time, but that the time-elongation curve assumes more the form of a parabola.

It is the writer's belief that at least one representative type of metal (pearlitic, austenitic, austenitic + carbides, etc.) should be tested at each temperature in question, that the test should be continued for several thousand hours, and the elongation determined accurately to a few millionths of an inch so that the true shape of the time-elongation curve can be definitely settled for once and for all. Also in this work the results obtained should be plotted to a scale comparable with the accuracy of the readings. Nothing is to be gained if readings accurate to a few millionths are plotted to a scale suitable for readings accurate to a few thousandths.

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[4] Department of Engineering Research, University of Michigan, Ann Arbor, Mich.

The above series of tests is believed to be of the utmost importance because practically every limiting creep value reported in literature is based on the assumption that the elongation is directly proportional to the time. If this can be definitely proved either one way or the other, greater reliance could be placed on reported limiting creep values.

GEO. A. ORROK.[5] The author has presented the kind of a paper which best represents the ideals of the Society: a statement of present-day practice and its difficulties, and an account of carefully conducted sets of experiments, followed by a set of definite recommendations for future practice. Most engineers will agree with him that the results secured by rules of boiler construction based on experience rather than theory have justified his statement. But an early engineer once said that where experience and theory did not agree, either the practice was poor or the theory imperfect. To the writer it would appear that the author was using both experience and theory and that little conflict was obtained.

In every construction exposed to pressure and temperature strains we have the main loading, steady or fluctuating within limits more or less determinate, and the additional loadings due to eccentricity, type of construction, and the conditions of operation. The main loading is taken care of by calculating the stresses, and the indefinite loadings are covered by a margin of strength; the ratio of main stress to ultimate stress is miscalled the "factor of safety."

For the sake of simplicity, expediency, or legal definiteness, much clearness has been sacrificed. The widespread use of factors of safety referred to main stresses has fostered an ignorance of actual stress conditions, and the legalization of these factors, however necessary, is nevertheless a decided deterrent in the general progress toward the knowledge of structures.

The steam boiler is probably the most representative construction where the magnitude of indefinite additional stresses reaches or surpasses that of the stresses due to the main load. We may freely admit that our theoretical knowledge of the strength of steel at elevated temperature is not large; furthermore our calculations of stresses are only conventional forms of approximation, miraculously working together with the factor of safety—but still we must design new boilers, for pressures and temperatures beyond previous practice. The remedy for this unfortunate condition must be sought in a series of successive steps undertaken to evaluate the additional stresses in order of their magnitude and to incorporate them in the regular boiler-strength calculations. The investigation of thermal stresses in tubes and drums and the localized stresses around holes should be the first step.

In the writer's paper on high-pressure boilers (1927)[6] the high values of thermal stresses in boiler tubes were indicated, but probably very few are aware of the fact that in most boiler tubes exposed to radiation the external fibers of the metal are in a state of compression in spite of inner steam pressure and that the thermal stresses due to heat transfer exceed those due to the pressure of steam. The question of concentrated stresses around holes and edges has been dealt with theoretically by many mathematicians, but their work is neglected in boiler practice.

Some years ago the phenomena of creep in steel became known, and now with the 1000-deg. fahr. steam temperatures entering the power-plant field it requires our immediate attention. In the presence of this phenomenon we are apparently losing our criteria and margins of stability. Dr. Jacobus advises the use of two-thirds of the creep limit stress as determined by experiment for long duration; 100,000 hours, for instance. The conception of stress margin is deeply rooted in the engineering mind, due to the long years spent on studies of cold structures. The range of temperature fluctuation in boiler parts is much wider than the range of variable stresses, and the stability of the structure, be it a drum, a tube, or a valve, depends to a very much greater extent on temperature than on the stress.

The principal stresses in boiler parts are not likely to fluctuate to a great extent, but the temperatures are, and the 100 deg. fahr. nominal temperature margin if acceptable for a drum is quite low for a boiler tube or superheated-steam circuit. Cases are known of very great variations of temperature. At Langerbrugge (Belgium) the 850-deg. fahr. normal steam temperature rose for $1^1/_2$ hours to 1056 deg. fahr., and usually fluctuates between 800 and 950 deg. fahr. At Gennevilliers one turbine designed for 750 deg. was getting steam up to 1020 deg. fahr. for prolonged periods. In American stations 100 deg. fahr. variations in total steam temperature have been common. Adding here the indefinite increment of temperature due to heat transfer at varying cleanliness, we see that the temperature of the metal will be the factor controlling the stability of the structure.

If all the failures and troubles are taken by and large, the majority of them will be found due to local stresses and hence two-thirds of the creep limit for the main conventional stress cannot be considered as equivalent to one-fifth of the ultimate strength cold nor to one-third of the elastic limit cold. However, one-third of the creep limit might be equivalent for stresses of long duration. For accidental short-duration stresses, two-thirds of the creep limit appears to be sufficient. This reasoning is valid at constant temperature, but in actual conditions of boiler operation many parts are subject to local overheating, and the average working temperature may vary within such limits that the stresses as a criterion of metal condition become quite misleading, their effect far exceeding the design limits.

Dr. Jacobus' conception of a design for a definite period of time with a given ultimate creep deserves serious attention. This idea was first put forward by Bailey in England. During this period the average working temperature will be increased by the integrated increments due to the variation in operation and by a constant fraction due to heat transfer. For instance, a 750-deg. fahr. steam plant during 100,000 hours must be considered as working at $750° + 30° + (850° - 759°)/n$, where 30° is added to heat transfer, 850° is the accidental maximum, and n is a factor of unevenness of operation; the total, at $n = 5$, will be 800 deg. fahr. This increase of average working temperature by 50 deg. fahr. lowers the creep limit of 12,000 lb. per sq. in. to 10,000 lb. per sq. in., using up half of the reserve stability calculated at two-thirds creep limit stress at 750 deg. fahr. In the range 900–950 deg. fahr. the additional 50 deg. use up all the reserve. It is clear thus that the stress margin of stability represented by one-third of creep limit stress vanishes quickly at a slight unavoidable increase of temperature, and cannot be taken as a criterion of long-period stability.

The logical method to use in the selection of working stresses, first proposed by Mellanby and Kerr, that of temperature margins, should be preferred. But whatever system of securing the reserve strength may be selected, the question of thermal and localized stresses must be fully investigated, and the final margin in the form of stress or temperature chosen in respect to the total calculable actual stress.

N. ARTSAY.[7] There are very few parts in boilers and engines that are not subject to combined stresses. The refinement of calculations which introduces thermal stresses and creep limits

[5] Consulting Engineer, New York, N. Y. Mem. A.S.M.E.

[6] Trans. A.S.M.E., vol. 49–50 (1927–1928), paper FSP-50-32.

[7] Principal Assistant of Geo. A. Orrok, Consulting Engineer, New York City.

certainly should be preceded by estimation of the combined effect of principal apparent stresses. For structures working below, say, 600 deg. fahr. there are three theories of failure under load. The first, that of exceeded principal apparent stresses, failed utterly. The second, that of exceeded true stress accompanying the maximum principal deformation, is being superseded now by the third theory, that of exceeded maximum shearing stress. Considering the creep phenomena in structures with combined stresses, we are without any theory of strength whatever.

The experimental data on creep at pure tension are already available in sufficient quantities, but the application of these results is hazardous or, at the other extreme, may involve much waste of material. For instance, if we have a cylindrical closed vessel under cold pressure the true stress will be $a\ (1 - {}^1/_2 m)$, where s is the principal hoop stress and m is Poisson's ratio. It seems thus that in a closed vessel a higher principal apparent stress could be allowed. But, while in an open cylinder subject to hoop stress the increment of volume is $x\ (1 - 2m)$, where x is the deformation, the increment of volume in the case of a closed cylindrical vessel is ${}^3/_2 x - 3\ xm$, which is greater. There are many considerations to support the view that the ability of metals to creep or to redistribute the molecular clusters is proportional to the increment of volume of the metal due to stresses, and hence in the case of two principal tensions the true measure of stress for the creep estimation will be a stress larger than the principal apparent stress.

An experiment to determine creep in the metal of a closed cylindrical vessel containing a neutral gas under pressure—nitrogen, for instance—made parallel with a creep determination in pure tension, will shed some light on this important question.

J. J. KANTER.[8] There can be little doubt that the distribution of creep stress in steel pressure vessels under operating conditions can by no manner or means be considered the same as in a tension-test specimen under constant load. While it is true that creep can redistribute stresses, it must be borne in mind that under fiber-stress conditions the ligaments of metal will be subjected to a range of creep-stress moments.

In view of this rather complicated stress condition, it would appear worth while to turn more attention to the problem of conducting creep tests on metal under fiber stresses as well as straight tension. Such tests might serve to throw some light on the discrepancies known to exist between the so-called creep limits and what has been found feasible for successful operation and service.

All of the difference between test data and service should not be attributed to thermal gradients alone. Until actual studies are made on fiber-stress creep, attempts at explanation remain highly speculative. A very important factor entering into creep phenomena, seemingly neglected in the paper, is creep arrest, or what French terms a transition from "primary" to "secondary" flow. Further studies upon this effect, made at Crane Co. laboratories, show continuous deceleration of creep rates on some tests of as much as one year's duration, load and temperature constant.

The classification of the rapid early creep and the subsequent slower rate now appears somewhat arbitrary. Fiber-stress creep is accompanied perhaps by an effect somewhat similar or analogous to the retardation found in tension. From a speculative standpoint, it even appears reasonable that transition from "primary" to "secondary" may be more pronounced in tube walls than for bars under tension. If the stress on the most strained fiber tends toward equalization, and therefore more creep than in less strained fibers, the outer fiber of a tube section under internal pressure should pass beyond the primary creep stage soonest.

[8] Crane Co., Chicago, Ill.

The result of such a condition, it is readily seen, is to bolster up the outer portion of the section to a certain extent, and the net rate of radial creep should retard much faster than it does in tension. This conception perhaps affords a slight clew to the reason why actual service proves a much greater load-carrying ability for tubes than appears possible from the results of long-time testing.

It is to be regretted that Professor F. H. Norton did not consider creep deceleration in his excellent series of tests. The method pursued in his tests, it appears, was to increase the testing load periodically, after several hundred hours of observation, until a suitable range of stresses and creep rates had been tried. While this procedure is somewhat economical of time and requires but one specimen for each temperature investigated, it leaves one of the natural and fundamental creep conditions entirely out of the picture.

Tests made at temperatures of 900 and 1000 deg. fahr. show that deceleration at "creep-limit" stresses is of a very considerable order. Moreover the total creep found for bars brought to load by the step method leads to much different values for extrapolated rates of creep than do the tests made, each using a separate specimen. (This was proved in the tests by the Crane Company.)

In view of these facts, a more rational evaluation of creep data would be had if total creep over a convenient period, and inclusive of the rapid initial flow, were considered.

It hardly need be said that the closer and more completely we adhere to observed facts, the greater will be the practical worth of the tests.

H. F. MOORE.[9] The use of Norton's logarithmic-plot method for determining the stress corresponding to a definite and very slow rate of "creep" under high temperature, instead of attempting to determine the stress limit for zero creep, is a very interesting feature of Dr. Jacobus' paper. The attempt to determine an absolute "creep limit" seems as unsatisfactory as the attempt to determine an absolute elastic limit.

The situation we face in attempting to choose a criterion for allowable stress at high temperatures seems to be analogous to that existing some ten years ago in the field of failure of metals under repeated stress. At that time there was not available any considerable amount of test data for long-time fatigue tests, and it was not certain that there was any "fatigue limit" below which an indefinitely large number of repetitions of stress could be withstood without fracture. In 1910 Professor Basquin, of Northwestern University, proposed for repeated stress a method of determination of safe stress for a given "life" of material which was very similar to that proposed by Professor Norton for "creep" under high temperature. Professor Basquin's method gave safe results, but long-time fatigue tests have given clear indications that there is a well-defined "fatigue limit" for most, if not all, metals.

Whether "creep" tests extending over longer periods than have been used for any tests up to the present will indicate a "creep limit" corresponding to the fatigue limit under repeated stress is a matter for the future to decide, and the time required for such a decision will probably be even longer than that needed to establish the existence of a fatigue limit. Meanwhile Professor Norton's method seems to be one which may be made to yield safe and reliable results for determining limiting stresses for metal parts in danger of failure by "creep" under high temperatures, and Dr. Jacobus' use of that method is to be regarded as a step in advance.

[9] Research Professor of Engineering Materials, University of Illinois, Urbana, Ill. Mem. A.S.M.E.

ALBERT SAUVEUR.[10] Dr. Jacobus deals with a problem that must command the attention of engineers and metallurgists until it is satisfactorily solved. His treatment of the subject is rational and illuminating. The writer finds himself in full agreement with him when he expresses the opinion that a closer relation exists between the creep value of a steel and its tensile

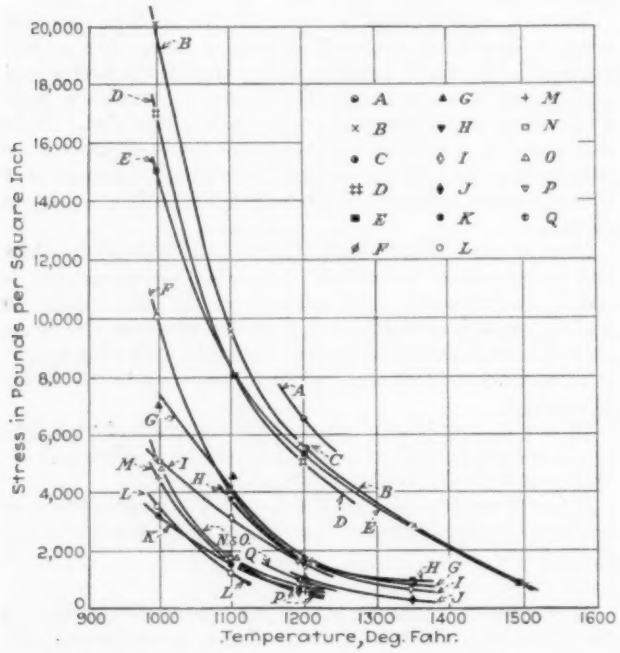


FIG. 8 CREEP STRESS FOR A LIFE OF 100,000 HOURS WITH 1 PER CENT ELONGATION

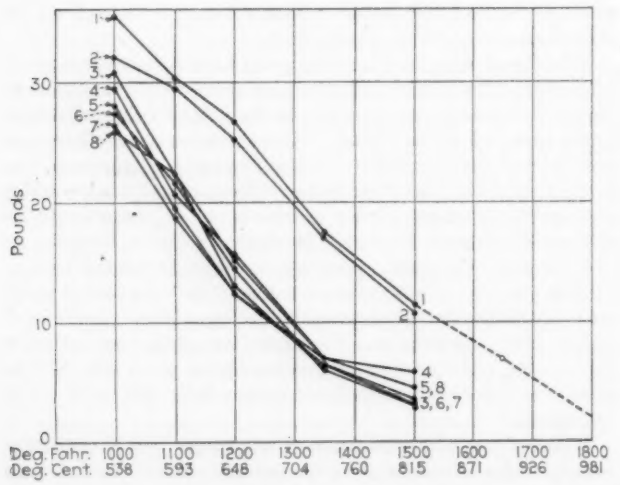


FIG. 9 RESULTS OF TORSION TEST AT TEMPERATURES SELECTED BY PROFESSOR NORTON

strength than between the former and the elastic limit or yield point.

The writer has had the opportunity to submit to torsional stress at elevated temperatures some of the steels used by Prof. F. H. Norton in his study of the creep of steel.[11] These steels included five nickel-chromium austenitic steels and twelve non-austenitic alloy steels and carbon steels. The results obtained by Professor Norton are shown graphically in Fig. 8 for a creep stress for a life of 100,000 hours with 1 per cent elongation. It will be seen that the austenitic steels (A, B, C, D, and E) are distinctly separated from all other steels, and that their resistance to creep is considerably greater at all temperatures between 1000 and 1400 deg. fahr.

Seven of the steels used by Professor Norton—namely, two nickel-chromium austenitic steels, three non-austenitic chromium steels, one nickel steel, and one low-carbon steel—were subjected to our torsion test at the temperatures selected by Professor Norton, namely, 1000, 1100, 1200, 1350, and 1500 deg. fahr. The actual torsional strength of the test bars in pounds[12] are plotted against temperatures in Fig. 9. The similarity between this diagram and Professor Norton's cannot fail to attract attention. The austenitic steels are again distinctly in a class by themselves. Had we used this simple and rapid twisting test as a means of ascertaining the relative creep values of these seven steels, our conclusions would have been quite similar to those reached by Professor Norton after many months of delicate testing. From the aspect of both sets of curves, one might wonder whether at temperatures in excess of 1650 deg. fahr., when all steels are austenitic, they might not also have substantially the same creep values.

F. H. NORTON.[13] The writer is sure that this interesting paper will be of great assistance to designers, as it summarizes and puts into workable form the scattered data on the properties of steel at high temperatures. Attention is called to the difference in stresses caused by plastic rather than elastic deformation. We greatly need more information on the distribution of stress under plastic conditions, and better theories for computing plastic flow in irregular objects. It seems evident from the scanty data at hand that the stress concentration usually found in irregular elastic bodies are much reduced in plastic objects.

We also need more data on the creep of steels at lower rates in order to determine how the rate of creep falls off with the load, and whether there is any limit below which plastic flow ceases. It has recently been noticed that the pendulums in astronomical clocks show a very slight yearly elongation, which has been attributed to plastic flow, even under these conditions of low load and low temperature. Perhaps in no other way could such delicate measurements of length change be made. At the Massachusetts Institute of Technology we are now conducting some creep tests which will run for a long enough time to detect creep under conditions of comparatively low loadings, so that we may be able to extend our data in this direction.

JOHN M. LESSELLS.[14] The paper in attempting to put on a working basis for design the data obtained on the tests on steel at high temperatures is to be highly commended.

The writer fully agrees with Mr. Orrok that it is more dangerous to have an increase in temperature than an increase in pressure. In this respect, the paper by Mr. H. L. Guy[15] is worthy of attention. The question of temperature variation is an important phase of choice of working stresses.

As to whether steel which has apparently ceased to creep while

[10] Gordon McKay Professor of Metallurgy and Metallography, Harvard University, Cambridge, Mass.

[11] "The Creep of Steel at High Temperatures," F. H. Norton. McGraw-Hill Book Co., 1929.

[12] "The bars were 1/2 in. square, grooved near the middle to a diameter of 1/4 in. over a width of 1/2 in.

[13] Professor, Massachusetts Institute of Technology, Cambridge, Mass.

[14] Engineer, Research Department, Westinghouse Elec. & Mfg. Co., East Pittsburgh, Pa. Mem. A.S.M.E.

[15] "Tendencies in Steam Turbine Development," read before the North Western Branch of the Institution of Mechanical Engineers, January, 1929.

subjected to high temperature will ever show later tendencies to creep, will depend on whether the temperature is sufficient to overcome the cold-working effects due to stressing. We can consider creep as the predominance of annealing over strain hardening.

The author has directed attention to working stresses for parts where comparatively large deformation can be allowed. There are other machine parts where the working stress must be chosen on the basis of permissible deformation, and in high-temperature testing these two phases of the work must always be considered.[16] This thought suggests that the title should be restricted to working stresses for steel at high temperatures as applied to drums and boilers.

Jos. Kaye Wood.[17] The writer feels that the publication of this paper is well timed and that the words of advice regarding greater activity and cooperation in this field of research should be given serious consideration. In this respect the author voices the opinions expressed by the writer in an article[18] published by him just prior to the presentation of the paper. In addition to stressing the inadvisability of using the yield point or proportional limit as the basic physical property for controlling the safe working stress, the writer also advocated a factor of safety equal to 1.5 when using the creep limit as the basic physical property. Using a factor of safety of 1.5 is equivalent to taking two-thirds of the creep limit as advised by Dr. Jacobus. The tensile strength is not as satisfactory as the creep limit for determining safe working stresses unless the factor of safety is based upon long experience as in boiler practice. Furthermore a basic physical property that requires a larger factor of safety is less reliable than one that requires a smaller factor of safety, provided both physical properties are determined accurately.

In his article previously quoted, the writer reports a creep limit at 932 deg. fahr., equal to 7000 lb. per sq. in. for firebox steel, which agrees closely with Dr. Jacobus' results. In the same article a creep limit of 14,000 lb. per sq. in. at 932 deg. fahr. is reported for vanadium steel. For these same steels at the same temperatures the proportional limits are given as 8000 lb. per sq. in. for the firebox steel and 6000 lb. per sq. in. for the vanadium steel. These comparative figures show how misleading the use of the proportional limit may be when employed as a basic physical property, even when this property is carefully determined, as was done in this case.

F. P. Coffin.[19] The question has been raised, by one of the previous speakers, as to the effect of different percentages of carbon on the strength of steel at elevated temperatures.

Some tests have been made in the Research Laboratory of the General Electric Company to determine this point, using steel with carbon contents of 0.16 and 0.40 per cent, respectively, over the temperature range from 400 to 650 deg. cent. (752 to 1202 deg. fahr.). Both steels were annealed at 800 deg. cent. (1472 deg. fahr.), and were tested at 400, 500, 600, and 650 deg. cent. The flow limits were almost identical at temperatures from 500 to 600 deg. cent. with the curve heading toward zero strength at the mean critical point of 723 deg. cent. (1334 deg. fahr.). At 400 deg. cent. the 0.40 carbon steel had about 50 per cent greater strength than the 0.16 carbon steel. The approximate flow limits for 1 per cent elongation in 10 years were 18,000 and 12,000 lb. per sq. in., respectively.

Another specimen of the same 0.40 carbon steel is now being tested at 450 deg. cent. (842 deg. fahr.) after a preliminary normalizing treatment in which it was air-cooled from 900 deg. cent. (1652 deg. fahr.) and drawn at 600 deg. cent. (1112 deg. fahr.). The test is still in progress, but the indications are that this specimen will fall on the same flow-limit curve as the annealed 0.16 carbon steel.

So far as we have carried this investigation, the indications are that the flow-limit curves of the two types of steel converge a little below 450 deg. cent. and that the effects of different heat treatments are equalized at 450 deg. by the annealing effect incident to holding the steel at this temperature during the test which has been under way nearly four weeks.

We have also found that steels containing around 1 per cent of manganese show a similar advantage at the lower temperatures, but that as the temperature is raised the flow-limit curve converges with that of carbon steel with a lower manganese content. In the case of 1 per cent manganese, however, the advantage is retained to a higher temperature than in the case of a 0.40 carbon steel. The two curves appear to converge somewhere between 550 and 600 deg. cent.

The figure mentioned by Dr. Jacobus as a safe working stress for carbon steels of 8000 lb. per sq. in. at 800 deg. fahr. (427 deg. cent.), which has been in practical use for a number of years, falls within the flow-limit curve which we have determined by long-time tests.

Author's Closure

The author wishes to thank those who discussed his paper for the consideration they have given it and for the valuable information submitted.

It is gratifying to note that Professor White, who is an authority on the subject, is in general accord with the working stresses given in Table 1. The work on this subject that is being done at the University of Michigan, as described by Mr. Clark, is of a most refined and exact nature and much useful information will be forthcoming.

Mr. Orrok emphasizes an important point in saying that whatever system of securing reserve strength may be selected, the question of thermal and localized stresses must be investigated and the final margin in the form of stress chosen in respect to the total calculable actual stress. He also emphasizes the importance of considering the effect of an increase in the average working temperature. As stated in the paper a variation in temperature has a great effect on the rate of creep, and in the case of any doubt it is well to err on the high rather than on the low side in assigning the working temperature.

Mr. Kanter refers to creep arrest, or what French terms a transition from "primary" to "secondary" flow, and says it is to be regretted that Professor Norton did not consider creep deceleration in his excellent series of tests. He also says that a more rational evaluation of creep data would be had if total creep over a convenient period, and inclusion of the rapid initial flow, were considered. The rate of creep determined by Professor Norton and as used in the paper is the rate after it becomes more or less uniform and does not include the effect of the initial elongation on the first application of load. It is well known that a slight amount of permanent set may occur in steel at comparatively light loads and that after a redistribution of the stresses in the steel there may be little or no permament set at the same stresses which initially caused a permanent set. This action takes place in drums and other working parts of boilers, and steel is used which has sufficient ductility to allow for a redistribution of the stresses. The most useful factor in creep tests in determining the stresses for boiler drums is therefore the rate of creep which occurs after the steel is submitted to a particular stress for a long period of time without including the initial distortion.

[16] P. G. McVetty, "Testing of Materials at Elevated Temperatures." Proc. A.S.T.M., vol. 28 (1928), part II, p. 60.

[17] Consulting Engineer, New York, N. Y.

[18] "Creep Limit Alters Technique in Design of Large Tanks," J. K. Wood. *Chem. & Met. Eng.*, vol. 36, pp. 610–613.

[19] Research Laboratory, General Electric Co., Schenectady, N. Y.

Professor Norton's, results are given in this way. For constructions other than boilers and pressure vessels the initial rate of creep may be a controlling factor, and it must not be assumed that the working stresses given in Table 1 are universally applicable.

Professor Norton refers in his discussion to additional tests he is making to determine how the rate of creep falls off with the load and whether there is any limit below which plastic flow ceases; these tests combined with others he is making will serve to throw additional light on what Mr. Kanter calls creep deceleration.

Mr. Lessells calls attention to the fact that the working stresses given are for parts where comparatively large deformation can be allowed and that there are other machine parts where the working stress must be chosen on the basis of permissible deformation. While the paper bears particularly on the selection of working stresses for drums built of A.S.M.E. Code steel having a minimum tensile strength of 55,000 lb. per sq. in., and for superheater headers of steel having a minimum tensile strength of 62,000 lb. per sq. in., the subject is considered from a broader viewpoint. As already stated, however, it must not be assumed that the working stresses in Table 1 are universally applicable."

It is interesting to note that Mr. Wood proposed a factor of safety of 1.5 based on the creep stress prior to the presentation of the paper.

Mr. Coffin states that the experiments made at the Research Laboratory of the General Electric Company show that 8000 lb. per sq. in. at 800 deg. fahr., which is given as the safe working stress in Table 1, falls within the flow limit curve determined by long-time tests, which checks the accuracy of this figure.

Mr. Howard C. Cross, of the Battelle Memorial Institute, Columbus, Ohio, called attention to an inaccuracy in Fig. 4 as published in the preliminary copy of the paper. This figure has been corrected. He also pointed out that it would have contributed to clearness if the compositions and treatments of the steel designated by code letters in Table 2 had been given. Table 4 was added to the paper to give the compositions of the steel in line with Mr. Cross' suggestion. The treatment of the steels is not given, as any close study would require further details of Professor Norton's experiments, which can be obtained from his book.

As brought out in the paper and in the discussions, there is a wide variation between different experimental results, and pending the time when there will be a closer agreement it will be well to lean toward the side of safety.

TABLE 4 COMPOSITION OF STEELS DESIGNATED BY LETTERS IN TABLE 2

Per cent of:	E	G	M	P	O	I
Carbon	0.09	0.10	0.40	0.40	0.34	0.39
Silicon	0.43	0.86	0.20	0.25	0.17	3.51
Manganese	0.38	0.31	0.59	2.22	0.46	0.35
Phosphorus	..	..	0.025	0.020	0.015	..
Sulphur	..	..	0.031	0.021	0.022	..
Nickel	8.12	0.23	1.30	..	..	0.22
Chromium	18.11	17.60	0.66	..	..	2.25
Tungsten	..	..	..	..	0.74	..

FSP-52-36

The Bin System of Pulverized Coal

By HENRY KREISINGER,[1] NEW YORK, N. Y.

The advantages of the bin system for handling pulverized coal are given as follows: Dependable feed of coal to furnace within a wide range of rating of the boiler. Easy adjustment of primary and secondary air. Coal can be pulverized in the off-peak period. Mills can always be operated at full rating. Uniform fineness of coal. Breakdown of mill does not interfere with boiler operation. Increased moisture or ash content does not affect maximum steam output of boilers. Wear on mill does not affect fineness of coal nor capacity of boiler. Clean and noiseless boiler room. High thermal efficiency of boiler plant.

IN THE bin or storage system the pulverized coal is first put through a preparation plant, from which it is conveyed to the storage bin located at the place of its use. From this storage bin the coal is fed to the furnaces and is burned as load conditions require. In the bin system the process of preparation of pulverized coal is independent of the process of burning. The difficulties that may exist in the preparation of the coal do not interfere with the process of burning. In like manner the coal may be burned with frequent changes from high to low rates without any inconvenient changes in the rate of milling.

The bin or storage system has been applied to large central stations as well as to industrial plants. One of the smallest industrial installations consists of two 320-boiler hp. boilers supplied with pulverized coal by one 6-ton roller mill. The largest central stations now burning pulverized coal use the bin or storage system. They include the plants with the highest yearly operating efficiency. The Avon Station of the Cleveland Electric Illuminating Company is operating with a yearly boiler efficiency of 88 per cent.

The bin system is also used extensively in industrial furnaces such as for heating billets and annealing. It is particularly well adapted to cases where close regulation of temperatures and furnace atmospheres is necessary.

The bin-system plants vary in design as to the location of the coal-preparation equipment. Some have a separate coal-preparation building located some distance from the boiler plant, others have the coal-preparation equipment placed near the boiler room, and in a few cases the mills are placed even in the boiler room. The location of the coal-preparation equipment depends on the plant site, the means of delivery of raw coal to the plant, and the tastes and prejudices of the designer and the operator.

The kind, the amount, and the arrangement of coal-preparation equipment vary with its location. When the coal-preparation plant is located some distance from the main plant, air transfer is generally used for conveying the pulverized coal from the preparation plant to the feeder bins in the boiler room, although the Fordson Plant has used screw conveyors about 600-ft. long for a number of years with entire satisfaction.

The use of air conveyance generally requires more equipment in the preparation plant than does the screw conveyor. There must be a complete air-compressing plant with air compressors and air-storage tanks. The coal must be dried in order that it may be transported successfully with air conveyance. This drying may require coal driers and means for removing coal dust from the drier gases before they are discharged into the atmosphere.

When the coal preparation plant is placed near or in the boiler room, the problem of preparing and conveying the pulverized coal is simplified. The air conveyance can be disposed of and the coal can be moved by short screw conveyors. The coal need not be dried to a low percentage of moisture, and in many cases the driers can be entirely omitted. What drying needs to be done can be easily accomplished in the mills during the process of pulverization.

Reason for Placing Preparation Plant at a Distance From Boiler Plant

When the coal-preparation plant is located some distance away from the main plant, it is placed at the coal-unloading plant or at the coal-storage pile. The unloading of the raw coal from railroad cars or boats or the taking of coal from the storage pile is the dustiest process around a power plant. A large amount of dust is always raised whenever the coal-unloading equipment is in operation. Therefore, wherever possible the coal-unloading equipment and the coal-storage pile are placed some distance from the main plant. The logical place for the coal-preparation plant is, then, near this coal-unloading equipment or near the coal-storage plant. The raw coal is taken directly into the coal-preparation plant, is pulverized, and then transported by air to the bins at the boilers. The transportation of pulverized coal by air conveyance is a much cleaner process than the method of conveying the raw coal from the coal-unloading equipment or storage pile to the main plant. All the dusty work is done some distance from the main plant, and the boiler room, turbine room, and switch house are kept clean and free from coal dust.

Most power plants have a coal storage in order to be assured of coal supply for power generation in case of strikes or any other cause of irregular coal shipment to the plant. Some plants receive their coal supply in large boat cargoes which must be unloaded in a specified time. Such large shipments of coal are usually unloaded into the storage coal pile, from which the coal is taken as it is required by the boiler plant. Since some methods of conveying the coal from the storage pile to the boiler plant would have to be provided, powdered coal offers a clean way of transporting this coal supply.

When the first pulverized-coal plants were designed, there was much agitation against pulverized coal, and the danger of explosion and fire hazards was much talked about. Comparatively little was known about the handling of pulverized coal and its behavior under various conditions. At that time a move was started to make rules for the safe installation and operation of pulverized-coal plants, and a separate preparation plant was at once adopted as the safest design. Little thought was given that such rules might handicap the development of the use of this new form of fuel. The designer of the plant played safe and

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Contributed by the Fuels Division and presented at the meeting, May 15, 1929, of the Boston Section, The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

placed the preparation plant some distance from the boiler room, generally at the coal-storage pile or at the coal-unloading plant.

During the last few years engineers have learned more about pulverized coal and the way to handle it. They know under what conditions it is dangerous and how to avoid such conditions. They feel that the design with the separate preparation plant far from the boiler room may entail greater expense and present many complications. They also have realized that a preparation plant can be so designed that it can be kept clean and can therefore be placed near or even in the boiler room if other considerations make such location desirable.

It would seem that a separate coal-preparation plant would make a power plant expensive. The author was once under such an impression, but closer investigation of the cost of the various types of plants designed and built within the last eight or ten years will show that such need not be the case. Once when talking with a designing engineer who was well informed on the cost of different power stations, the author expressed his opinion that separate coal-preparation plants were not a good design because they made the power plant too expensive. As examples of such designs were mentioned the Lakeside Station of the Milwaukee Electric Railway and Light Company, the Trenton Channel Plant of the Detroit Edison Company, and the Middletown Plant of the Metropolitan Edison Company. The designing engineer smiled and said this: "Do you know that the three plants you have just named are among the plants with the lowest cost per kilowatt capacity of any plants in the United States?" An examination of the cost shows that the Trenton Channel Plant costs about $103 and the Lakeside Plant and the Middletown Plant less than $90 per kilowatt capacity. Whether these low costs are due to economic design or economic construction can be determined only by careful study of the itemized cost. The high costs which appear obvious may be found not to exist on close examination. A careful, impartial study of the location of the plant, the available coal supply, the method of shipment of coal to the plant, the load factor, and the variation of the load on the plant must be made before the most economical design can be determined.

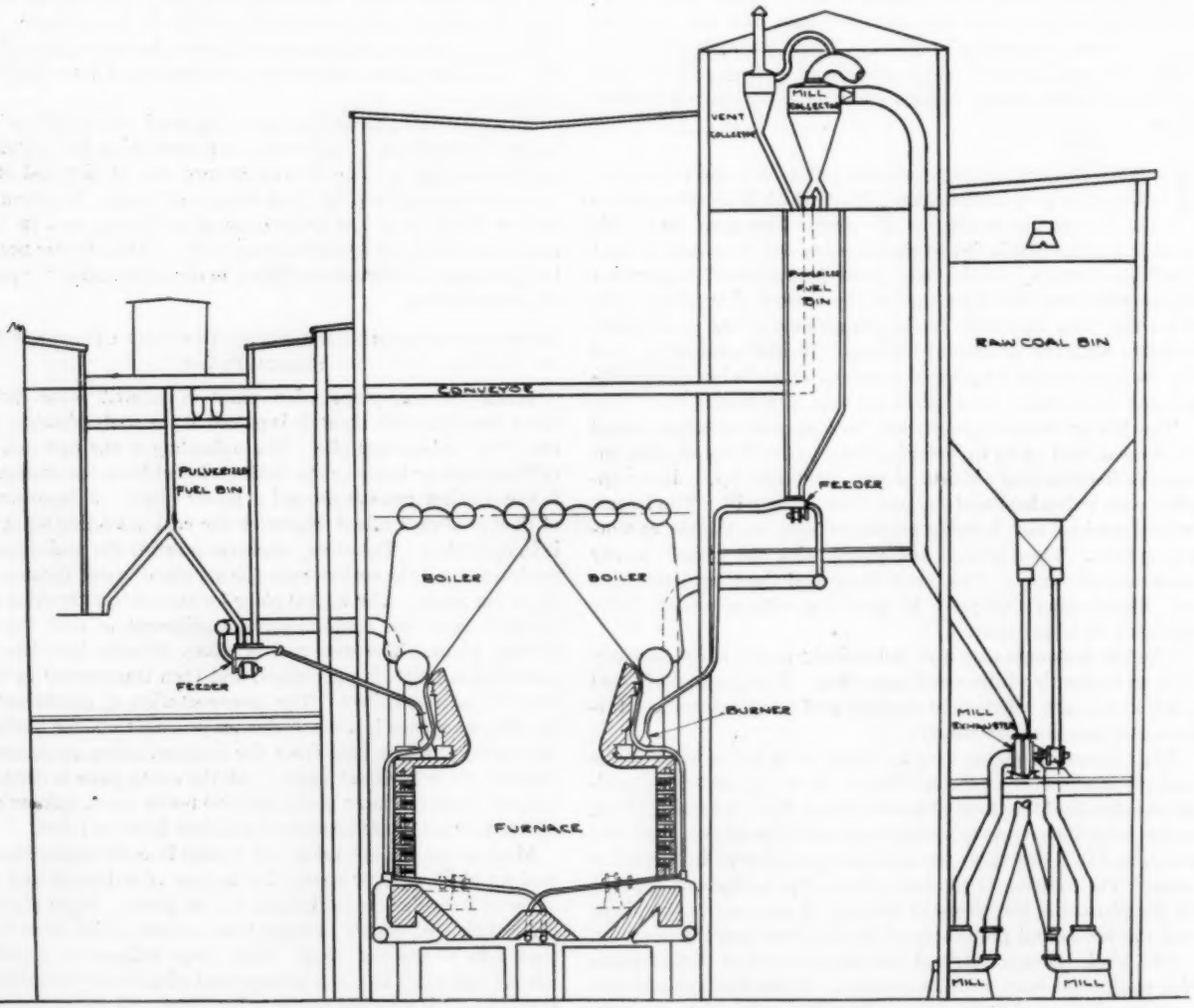


FIG. 1 LAKESHORE PLANT OF CLEVELAND ELECTRIC ILLUMINATING COMPANY
(Coal-preparation plant is located in a separate room adjacent to the boiler room.)

Fig. 1 shows the plant of the Cleveland Electric Illuminating Company. Coal-preparation equipment is located in a separate room adjacent to the boiler room. Screw conveyors are used to convey pulverized coal to the feeder bins.

Fig. 2 shows the plant of the United Railways Company, Providence, R. I. The mills are on a platform in front of the boilers.

Sequence of Processes in the Bin System

The following is the sequence of processes in the bin or the storage system:

(*a*) Drying raw coal
(*b*) Pulverizing coal
(*c*) Transporting pulverized coal to feeder bins
(*d*) Feeding pulverized coal to burners
(*e*) Burning pulverized coal.

Drying

Ten years ago it was thought necessary to dry all coals below 1 per cent moisture in order that the coal could be successfully pulverized and burned. Separately fired horizontal rotary driers of the kiln type were then in general use. As experience in preparing and burning pulverized coal accumulated, it became evident that such extreme drying of coal is needed only in very few cases. In most cases the moisture in the dried coal may be considerably higher. Inasmuch as the separately fired horizontal driers were costly to install and to operate, and took too much space, the possibility of using successfully coals with higher percentages of moisture introduced other types of driers which were more compact and did not require separate firing. Thus flue gas and steam driers with gravity feed were made use of, and later a vertical rotary drier heated with steam was developed.

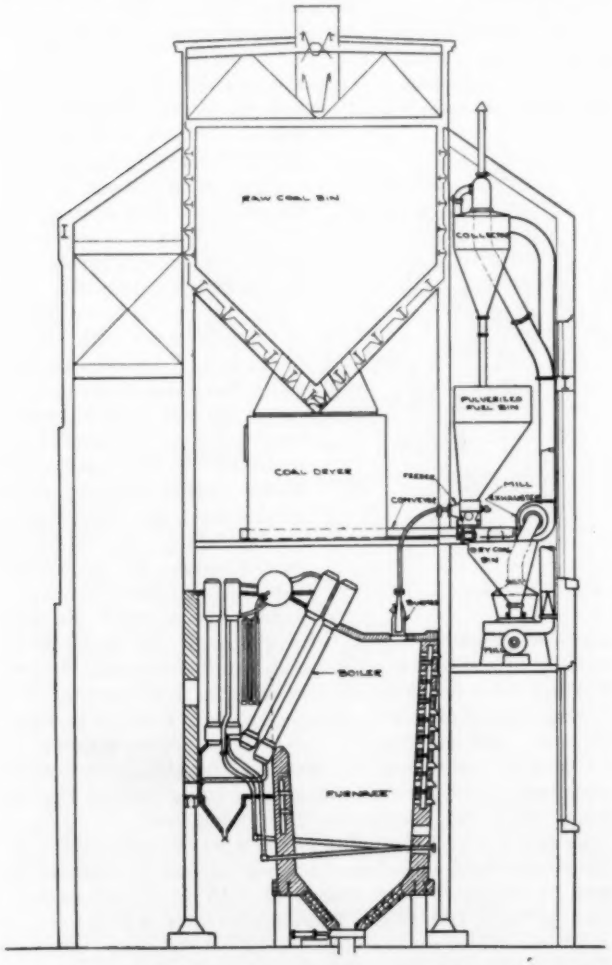


Fig. 2 Plant of United Railways Company, Providence, R. I. (Mills are located in the boiler room in front of the boilers.)

Still more recently drying coal in mills was applied with considerable success, and many of the new powdered-coal plants are installed with this method of drying.

Gravity-Feed Driers. The gravity-feed-type driers did not come up to full expectations, and very few are now being installed. The chief objection against them is that the gravity feed is not reliable; the coal hangs in them and does not flow through them at a uniform rate. When the coal is very wet and fine, the flow may stop altogether. Thus the drier is least effective when most needed.

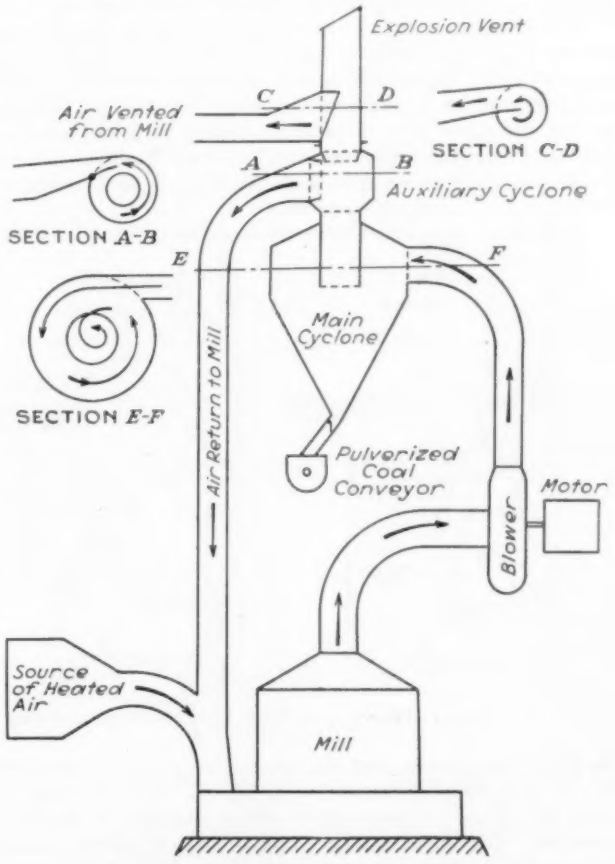


Fig. 3 Mill System Arranged for Drying

Vertical Rotary Steam Drier. The rotary steam drier gives the coal a positive flow, and therefore the drier is very useful for very fine and wet coals which need drying the most. The vertical rotary steam drier takes very little floor space, but requires considerable head room. It is probably the best drier for power-plant purposes.

Dust Emission From Driers. All driers emit a certain amount of dust with the air or gas which is used in drying. The horizontal rotary drier is the worst offender in this respect because the coal cascades in the drier, and much dust is thereby put in suspension in the gases and carried out. It is necessary that the air or gas from the driers be sent through a cyclone separator to remove most of the dust, and in some cases the gas or air must be washed before it is discharged into the atmosphere.

Mill Drying. The mill drying has much to recommend it. It is capable of doing as much drying as is necessary, with very little additional apparatus. Fig. 3 shows diagrammatically the mill system arranged for drying. In the air-separation mill 2 to 3 lb. of air per pound of coal are circulated through the mill system to lift the pulverized coal out of the mill and deposit it

in the cyclone separator. When coal is dried in the mill, heated air is continually introduced into the mill system and an equal weight of air taken out of it. The added air supplies the heat needed for drying, and the air vented from the mill carries away the moisture that has been driven from the coal. For drying purposes $^1/_2$ to $1^1/_2$ lb. of heated air per pound of coal is supplied. By the mill-drying method 2 to 6 per cent of moisture can be removed from the coal. The heated air for drying can be supplied from the air heaters for heating air for combustion with the flue gases or it can be supplied with a special steam air heater located near the mill and heating the air with high-pressure steam.

If the mills are located near the boiler room, the air vented from the mill can be used either as primary or secondary air, or otherwise blown into the furnace, so that the dust which it may carry can be burned, and no special devices are required for cleaning the air. In cases where the mills are located far from the boiler room and it is impractical to carry the vents from the mill to the furnaces, it is necessary to wash this air in air washers before discharging it into the atmosphere.

Air Washer. In the air washer the dusty air is first humidified by low-pressure steam, and the mixture of steam and dusty air is then subjected to very fine water sprays. The water sprays cool the mixture and cause the steam to condense on the dust particles. The condensation of steam on the dust particles causes them to become heavy and more easily thrown out of suspension. The wetted and coagulated dust particles are collected in the hopper-shaped chamber in which the air makes a turn of 180 deg. In order to facilitate the settling of the wetted particles, eliminators in the form of vanes are placed in the stream of the gas at the point of turning. Water sprays are used to keep the eliminators free of dust.

The amount of dust carried from the mill into the air washer depends upon the volume of the air used for drying and the percentage of moisture to which the coal is reduced by the drying process. The larger the volume of air vented from the mill, the greater the quantity of dust carried into the air washer. Also the lower the percentage of moisture in the dried coal, the greater the dust loss. If the percentage of moisture in the dried coal is to be low, the humidity of the air passing through the mill must be low in order that the moisture would leave the coal and be absorbed by the air. On the average the dust lost in the air washer is about 0.5 per cent.

Disposal of Mill Vents. Fig. 4 shows the method of disposing of the mill vents without an air washer. The mill vents are used as primary or as secondary air, or a combination of the two. The diagram on the left of the figure shows the mill vent used as part of the secondary air introduced directly into the horizontal burner. The diagram on the right shows the mill vents used as secondary air with vertical firing, in which case the mill vent is blown into the furnace through air ports in the front wall. The diagram in the middle shows the mill vents used as a combination of primary and secondary air. Such use of the mill vents is practicable only when the mills are located near the boiler room so that the ducts need not be excessively long.

Location of Cyclone Separator. The cyclone separator can be under either pressure or suction. In the older installations the cyclone separator was always under a pressure. That is, the mill blower was located at the outlet from the mill and forced the mixture of coal into the cyclone. With such arrangement the mill blower handled all the coal, and there was a considerable abrasion of the mill blower due to the high concentration of dust. The blades of the blower wheel suffered especially from such abrasion and had to be renewed after the mill had pulverized 10,000 to 30,000 tons of coal.

In order to eliminate this abrasion, the mill blower in more recent installations is located between the cyclone separator and the mill. The mill blower then handles the air after the coal dust has been separated out of it by the cyclone. With such an arrangement there is practically no wear on the mill blower, and the blades last indefinitely.

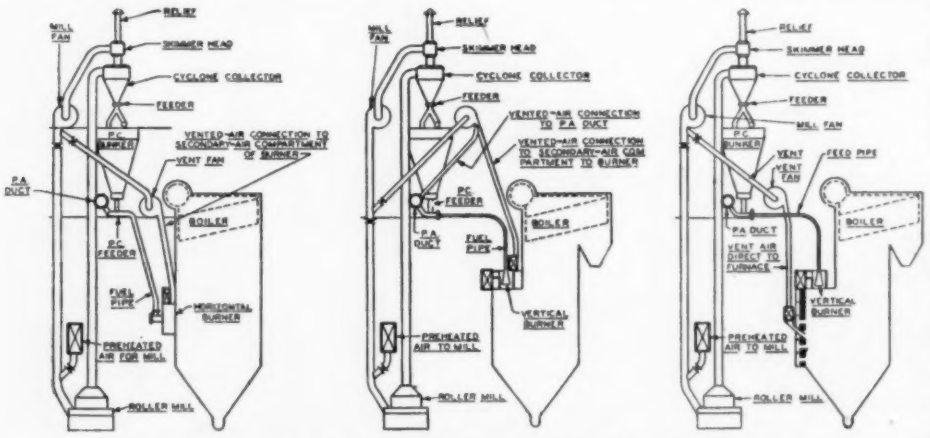


FIG. 4 DISPOSAL OF MILL VENTS BY UTILIZING THE AIR FOR COMBUSTION

Fig. 5 shows the two locations of the mill blower with respect to the cyclone separator and the mill. The diagram on the left shows the mill blower located between the mill and the cyclone separator. In this case the mill blower handles the mixture of air and coal dust. The diagram on the right shows the blower located between the cyclone and the mill. The mill blower handles the air after the dust has been separated from it. In both cases the heated air is supplied by a special small steam air heater located near the mill. The mill vents are shown to discharge into an air washer.

Percentage of Moisture. The percentage of moisture to which the coal must be dried for successful operation depends on the kind of coal. Generally speaking, the high-grade Eastern coals must have a lower percentage of moisture than the Western coals. Thus the Pocahontas, New River, and Pennsylvania coals may require drying down to 2 or 3 per cent of moisture, the Illinois and Iowa coals down to 6 or 8 per cent of moisture, the semi-bituminous Western coals to 12 or 15 per cent of moisture, and the North Dakota and Texas lignites to 24 or 28 per cent of moisture. There are two large central stations using pulverized lignite where the lignite is dried in gravity-feed steam driers to an average of about 26 per cent moisture. With this apparently high moisture the lignite pulverizes, handles, and burns satisfactorily. In both of these plants the pulverized lignite is conveyed to the feeder bin by screw conveyors.

PULVERIZATION OF COAL

The fineness to which coal must be pulverized in order to be successfully burned varies with the kind of coal. Generally speaking, the high-fixed-carbon coal must be pulverized to a higher degree of fineness than the high-volatile-matter coals. The volatile matter leaves the particles of coal upon heating and burns largely as gas. It is therefore easy to burn. The fixed carbon,

however, stays as coke residue in the coal particle and burns slowly. The smaller the original coal particle, the smaller will be the coke residue of the fixed carbon and the less time will be required to burn it completely. If the original coal particles are large, the coke residue left after the distillation of the volatile matter may be too large to burn completely while they remain in the furnace. The high-fixed-carbon Eastern coals should therefore be pulverized to a fineness of 65 to 75 per cent through a 200-mesh screen, the high-volatile coal to a fineness of 55 to 65 per cent through a 200-mesh screen, and the lignites and peats to 40 to 50 per cent through a 200-mesh screen. On the other hand, the anthracite coal may have to be pulverized to 80 or 85 per cent through a 200-mesh screen to get equally as good combustion as with the other coals.

The degree of fineness should also vary according to the furnace in which it is to be burned. If the furnace is small, the coal must be pulverized very fine in order that complete combustion may be obtained. In the same way a large furnace will burn fairly coarse coal without high losses from incomplete combustion.

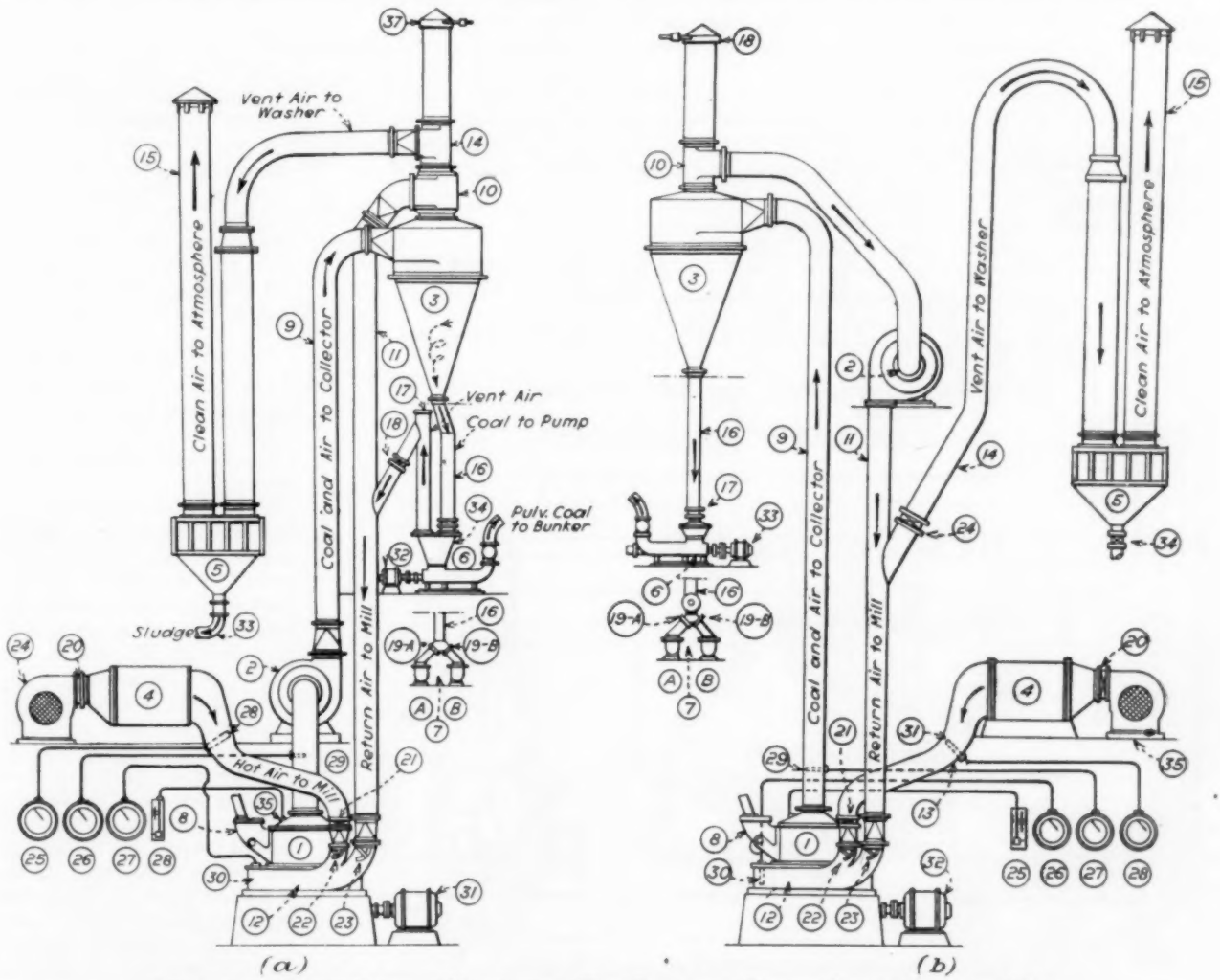


FIG. 5 TWO WAYS OF LOCATING MILL BLOWER WITH RESPECT TO CYCLONE SEPARATOR AND THE MILL

(The diagram on the left shows the mill blower between the mill and the cyclone separator; the blower handles pulverized coal. The diagram on the right shows the mill blower between the cyclone and the mill; the mill blower handles the air after the pulverized coal has been separated from it.)

TRANSPORTATION OF PULVERIZED COAL

The pulverized coal is transported either by screw conveyors or air conveyance. The screw conveyors are cheaper to install and to operate. They are simple in construction and reliable in their operation. If the distance over which the coal is to be conveyed is not too long, the screw conveyor generally proves the more satisfactory means of moving pulverized coal.

Air conveyors are used in practically all cases where the preparation plant is located some distance from the boiler room. They lend themselves easily to central control of the distribution of coal.

That is, coal can be delivered to any bin in the boiler room by means of electrically operated valves which can be opened or closed from a centrally located switchboard. Several air conveyors can feed one main line, and the coal can be discharged to any bunker in the boiler room. Fig. 6 is a diagram of a coal-distribution system by air conveyance. The amount of air and the pressure required for the transportation of powdered coal vary with the distance, and generally 20 to 60 lb. of coal can be transported with 1 lb. of air. The pressure at the transporter varies from 20 to 60 lb. per sq. in., depending on the length of the transport line and the number of turns in it. The velocity of the mixture of coal and air through the transport line is about 20 ft. per sec.

Feeding of Pulverized Coal

The pulverized coal is fed from the storage or feeder bins to the furnaces at any desirable rate by means of feeders. Fig. 7 shows four representative types of feeders. Of the four types the two with the vertical shafts shown at the top of the figure give the most satisfactory all-around results. These feeders have agitators attached to the top end of the vertical shaft which prevent bridging of the coal and cause it to flow without interruption into the feeder. The feeder at the right is a duplex feeder; that is, it feeds two streams of coal of equal size and therefore can supply two burners with coal.

The lower half of the figure shows two screw feeders. The one on the right is a single-screw feeder and is one of the oldest devices for feeding pulverized coal. If the coal is very dry the coal may flood through the screw and cause irregular feedings.

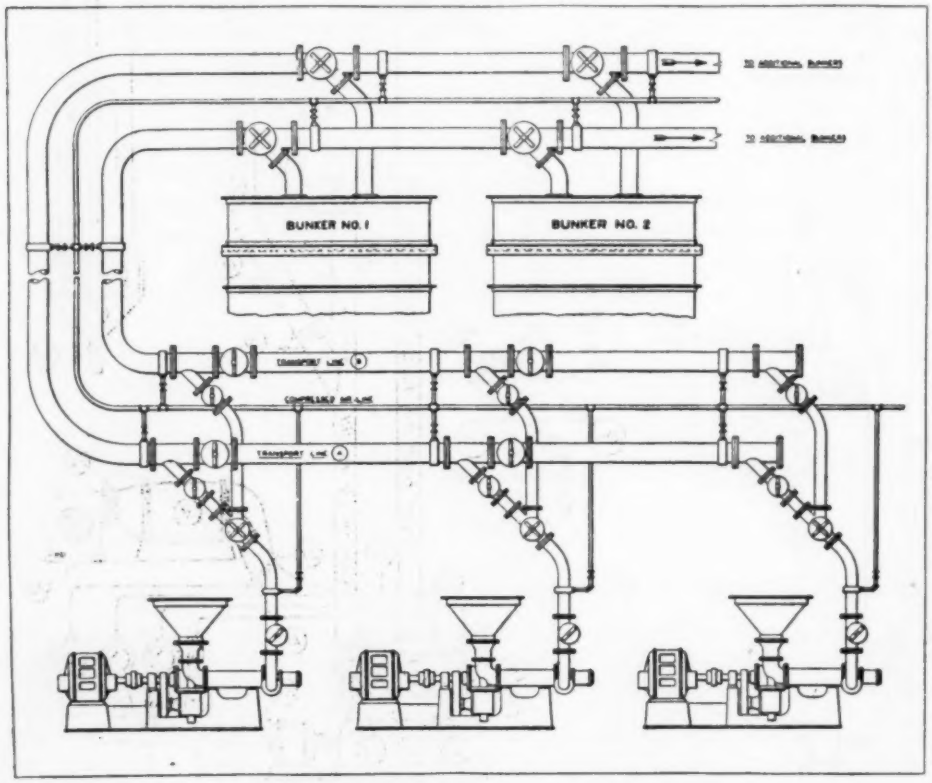


Fig. 6 Coal-Distribution System by Air Conveyance

In order to stop this flooding a special check valve is placed at the discharge end of the feeder screw. This check valve consists of a disk which is gently pressed by a helical spring against the barrel of the feeder. The check valve keeps the barrel of the feeder full of coal and allows the coal to pass only as fast as the feeder screw moves it. The flooding does not exert sufficient pressure on the check valve to open it and permit the coal to go by. A rotary agitator is placed above the feeder screw to stir the coal, thus making it flow into the feeder screw. There are a great many of these feeder screws in operation, and in many cases they are entirely satisfactory. In fact, some of the users are reluctant to change to the newer types of feeders. In other cases considerable difficulty is encountered with the coal sticking in the feeder and causing irregular feeding and excessive wear on the screw.

The picture at the lower left-hand corner shows a double-screw feeder. The double-screw feature is intended to eliminate flooding. Observations on this type of feeder show that in this respect the feeder is successful.

Rate of Feeding. The rate of feeding the coal is determined by the speed of the feeder. The feeders are driven either by variable-speed motors or by constant-speed motors through a variable-speed drive. In general, the variable-speed motor is more desirable. It takes less space, and the control of the feeding is more easily obtained. It is also more reliable, and the maintenance is low.

Usually the variation in speed in the ratio of 1 to 4 is employed. Generally, there is more than one feeder on a boiler, so that by shutting down some of the feeders a further reduction in the rate of feeding can be obtained. This is one of the advantages of the storage system and permits very wide variation in the rate of working the boiler. In one installation a ratio as high as 1 to 12 in the rate of feeding coal is used with good combustion results. The power required to drive the feeders is about one-half of a horsepower per ton of coal per hour.

Firing Pulverized Coal

There are three ways of firing pulverized coal in general use—namely, the vertical firing, the horizontal firing, and the tangential firing.

Vertical Firing. In the vertical-firing method the mixture of coal and primary air is fired vertically downward through burners located in the arch of the furnace. Secondary air is supplied through air ports in the front wall. The mixture of primary air and of coal and the secondary air meet at right-angles, thus producing mixing. The stream of coal and air turns near the bottom of the furnace and flows upward, making the path of the coal and air through the furnace of U-shape. In installations where air heaters are used, the secondary air is preheated and supplied to the furnace under a pressure varying from about 1 to 2 in. of water. When the secondary air is supplied under pressure, the air ports in the front wall are small and the air enters the furnace with high velocity, and more turbulent mixing is produced.

The rate of combustion is usually limited by the ability of the furnace walls to withstand the slagging action of the ash. With refractory walls the rate of heat liberation is 10,000 to 18,000 B.t.u. per cubic foot of combustion space per hour. With water-cooled furnaces the rate of heat liberation is often 25,000 B.t.u. per cubic foot of total combustion space per hour. With water-cooled furnaces the limit to the rate of heat liberation is the slagging of the boiler tubes. If the rate of heat liberation is too high, molten ash is carried into the boiler tubes and clogs the gas passages.

Horizontal Firing. In the horizontal method of firing the secondary air is usually supplied under pressure. This is particularly true where air heaters are used. The secondary air, as well as the primary air is given a rotative motion as it leaves

the burner, which motion produces very turbulent mixing, resulting in good combustion.

Tangential Firing. In the tangential method of firing the burners are located in the corners of a nearly square furnace. The streams of mixture of coal and air are directed toward the center, tangent to a small circle. The burners are located either near the top of the furnace, with the gases moving in the general downward direction, or at the bottom of the furnace, in which case the gases are moving in a general upward direction. The impingement of the four streams of coal and air against each other near the center of the furnace produces very intensive mixing, and combustion takes place at very high rates. Inasmuch as there is also considerable impingement of the flame against the side walls, this method of firing is used only with water-cooled furnaces. The usual rate of heat liberation is 20,000 to 30,000 B.t.u. per cubic foot of total furnace volume.

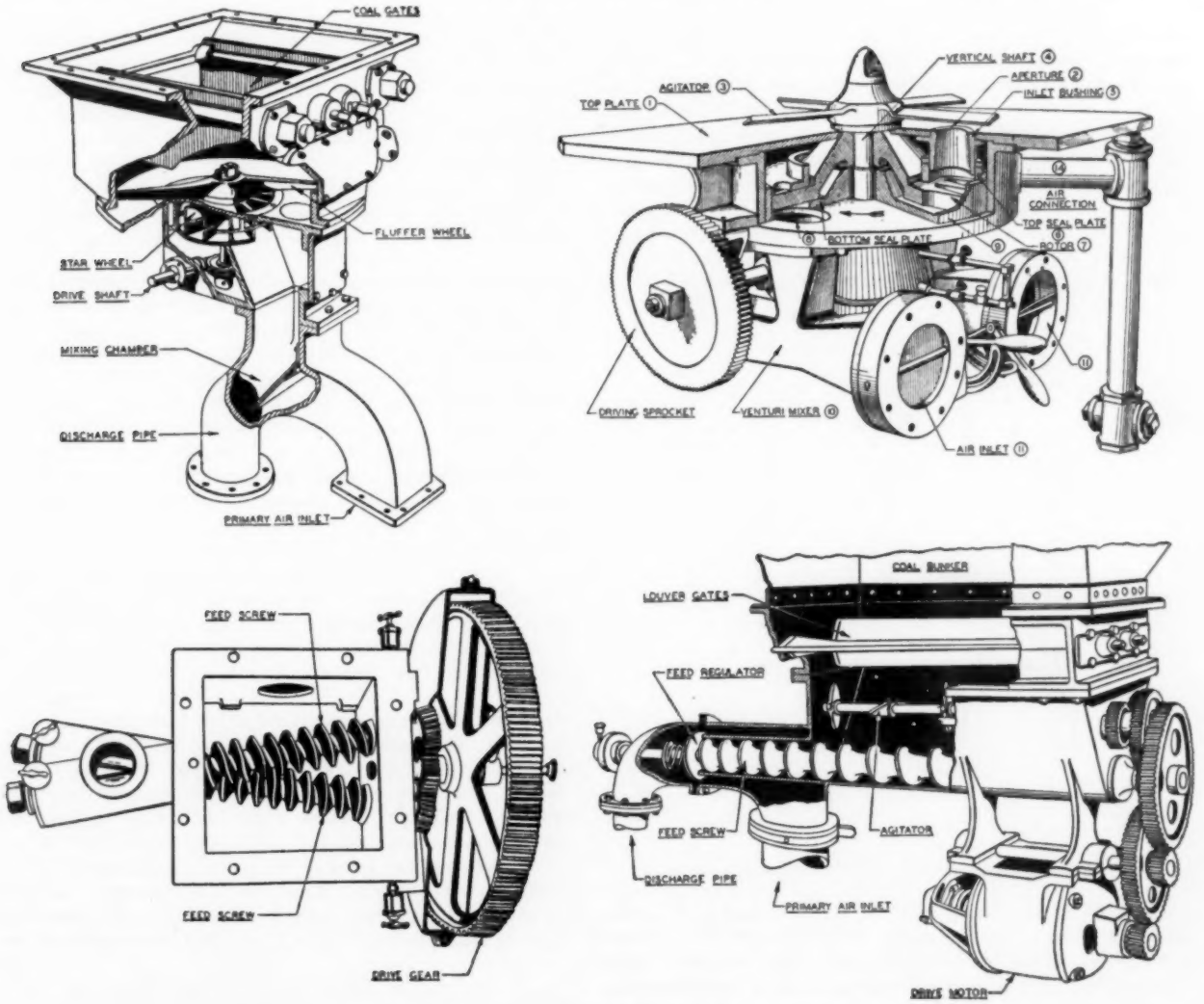


FIG. 7 FOUR TYPES OF PULVERIZED-COAL FEEDERS

These features make the bin system especially desirable for a power plant supplying power to railroads where the load varies over a very wide range and for steel plants where pulverized coal is burned as a supplementary fuel to blast-furnace gas.

In one railroad plant the load on the turbo-generator is increased from 5000 to 20,000 kw. within one minute and is frequently dropped within the same short time. The boilers must be able to pick up the load without appreciable steam-pressure drop and must drop it without the blowing off of the safety valves.

At the plant of the Youngstown Sheet and Tube Company powdered coal is used as auxiliary fuel to blast-furnace gas. Whenever the gas supply is reduced below the quantity necessary to hold the pressure at a given point, coal feeders and burners are automatically turned on and kept in operation, holding the pressure constant until the blast-furnace gas supply is increased, when the feeders and burners are again automatically turned off.

(3) Coal can be pulverized in the off-peak period, making it possible for the plant to generate more salable power during the peak-load periods. Thus the storage plant has higher peak capacity, which, when capitalized, reduces the first cost of the station.

ADVANTAGES OF THE BIN SYSTEM

(1) Uniform and dependable feed of coal to furnace within a wide range of rating of the boiler.

(2) Easy adjustment of primary and secondary air to any coal and rate of feeding.

(4) Mills can be always operated at full rating, which gives the lowest power consumption per ton of coal pulverized.

(5) Uniform fineness of coal.

(6) Breakdown of a mill does not interfere with the boiler operation.

(7) Increased moisture or ash content of the coal, though it may affect the rate of milling, does not affect the maximum steam output of the boilers.

(8) Wear on the mill does not affect the fineness of coal and the capacity of the boiler.

(9) Clean and noiseless boiler room.

(10) High thermal efficiency of boiler plant.

FSP-52-37

The Purpose and Characteristics of the Air-Flow Mechanism of a Boiler Meter

By W. G. LAUFFER,[1] NEW YORK, N. Y.

Stack loss in a modern steam-generating station is assumed to be a controllable item effected by the installation and use of combustion guides. The paper reveals the probability of error in such an assumption. The air-flow mechanism of a well-known boiler meter, the proper operation and calibration of which have probably not been understood, is given a detailed study in this report, and the result is described as resulting in an indicator of combustion efficiency that is reliable within close limits.

THE largest controllable loss in a modern steam-generating station is the stack loss. In the design of a station, it is usually assumed that control of the stack loss will be satisfactorily effected by the installation and use of combustion guides. This report reveals the probability of error in such an assumption. The air-flow mechanism of a well-known boiler meter is used at the Hudson Avenue Station for control and indication of the stack loss. The proper operation and calibration of this meter had not apparently been properly understood nor completely developed. However, the detailed study of the purpose and characteristics of the mechanism, which is described in this report, has resulted in an indicator of combustion efficiency that is reliable within very close limits.

Excepting a brief description of the meter, included for clarity and completeness, only the additional developments of the theory and use of the air-flow mechanism are here discussed. Detailed description has been left to the manufacturer's bulletins and instructions.

The work was conducted by the Hudson Avenue Station betterment crew of the Test Bureau, Operating Department, Brooklyn Edison Company. The author wishes to express his recognition of the spirit in which was carried out the routine of innumerable Orsat analyses and "deadweight checks," on which rests the proof of the theories presented. Indebtedness is also acknowledged to the engineers directly in charge of the operation of the station as a whole, and of the boiler room in particular, who, appreciating the importance of the problem, rendered every possible assistance to arrive at a satisfactory and prompt solution.

The report of the work is divided into four parts:

Part I relates the difficulties that had been encountered in setting the boiler meters for predetermined air requirements and the first steps that were taken to remedy these difficulties. This part also describes the development of a method of calibrating the meter against differential, the "deadweight check," and presents the computations involved.

Part II includes the fundamental mathematics for the derivation of the proper displacer contour. This is expanded in the Appendix.

Part III is a discussion of the determination of the desired characteristic of total air with respect to rating. This part also includes the development of the proper technique of flue-gas sampling and the demonstration of the necessity for gas-sampling lines on both sides of the boilers.

Part IV describes in detail the routine that has been developed to insure continuous maintenance in calibration of the air-flow mechanism, the necessity of which is a fundamental prerequisite to a consistently minimum stack loss.

The text has been written with the technical theme interwoven with the actual progression of the practical work involved in the determination of the correct functioning and use of the instrument. It is believed that this treatment is not only most interesting, but will awaken an appreciation of some of the basic reasons why the application of instrument guides for combustion have been only too commonly discounted by the men who use them.

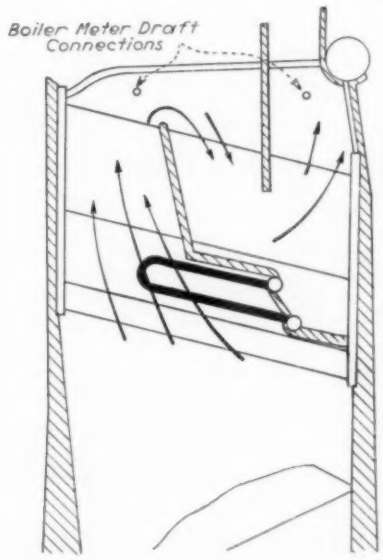


FIG. 1 POINTS AT WHICH DRAFT IS TAPPED OFF

DESCRIPTION OF THE BOILER-METER MECHANISM

The boiler-meter-flow elements in question measure the flow of steam from the boiler and the flow of products of combustion through the boiler. The air-flow pen is to be so adjusted that, when the fuel is being burned with the proper amount of air, the two pens, indicating air flow and steam flow, show the same reading on the boiler-meter chart.

The steam-flow element consists of a bell actuated by the difference in pressure occurring just before and immediately after an orifice or flow nozzle in the steam line from the superheater outlet to the header. This bell is so shaped that the movement is proportional to the square root of the actuating differential, and consequently directly proportional to the flow causing the differential.

The air-flow element measures the flow of flue gas through the

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Presented at a meeting, New York, N. Y., May 9, 1929, of the Metropolitan Section of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

boiler and is actuated by the differential draft between two points in the boiler in the gas path. The boilers at the Hudson Avenue Station are of the B. & W. cross-drum three-pass type. The points at which the draft is tapped off are at the top of the first pass and at the top of the third pass. (See Fig. 1.) The difference in draft between these two points represents a friction loss of head due to the gas flow through the boiler. This head loss has the characteristic of all friction-head losses—i.e., the loss is proportional to the square of the flow. Since, however, in this meter the steam and air pens are to be coincident, the air-flow element must, like the steam-flow element, give a deflection that is proportional to the flow rather than one proportional to the square of the flow or to the head. This is accomplished by counterweighting the arm from which the draft-actuated bells hang with a weight which is immersed in mercury. This weight or displacer is so shaped that, in withdrawing it from the mercury, the force exerted by it on the arm will cause the meter reading to be proportional to the air flow.

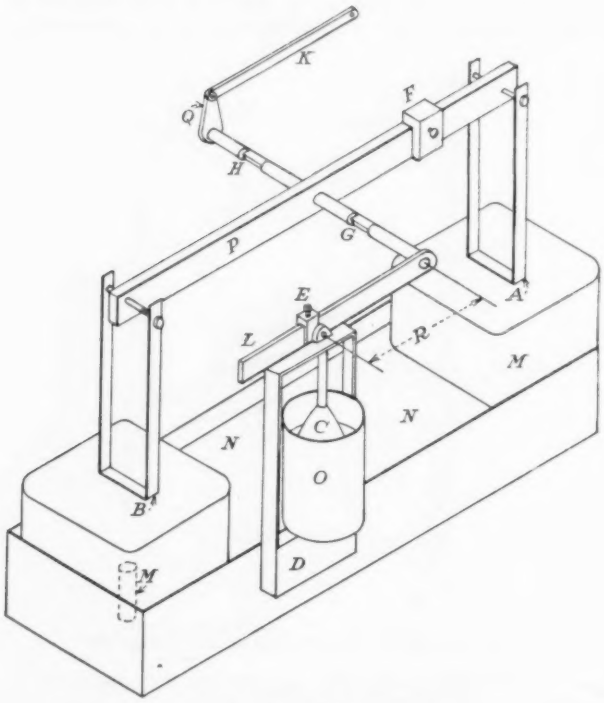


FIG. 2 AIR-FLOW MECHANISM

The air-flow mechanism is shown in Fig. 2. *A* and *B* are the draft bells suspended in a bath of oil *N*. The draft is introduced under the bells through the $^1/_8$-in. pipes *M*. *C* is the displacer, fixed in its yoke weight *D*, hanging in the cup of mercury *O*, from the movable clamp *E*. The air bells, the displacer, and its yoke weight all act on the lever *P*, which, balanced on the knife edges *G* and *H*, operates the air pen through crank *Q* and arm *K*. The weight *F* is an additional adjustment, the function of which is explained later in detail. The meter is set for any desired excess air at a given boiler rating by changing the distance of the clamp *E* from the fulcrum—i.e., by changing the distance *R*. This distance is hereafter referred to as the "radius" of the meter, while the weight *F* is referred to as the "counterweight" or "balance."

When the meter is installed, the higher draft (in this case the draft at the top of the third pass) is introduced under bell *A* and the lower draft under bell *B*. When the beams *P*, and consequently *L*, are in zero position against a stop which is not shown in the sketch, the mercury in the pot *O* is carried at the level of the joint between the displacer and the rod from which it hangs. The zero position is assumed by the mechanism when there is no differential between the bells *A* and *B*.

As the difference in draft between bells *A* and *B* increases, a torque is exerted in a clockwise direction. This tilts the arm *P*, lifting the displacer *C* out of the mercury. The downward force exerted on the lever *L* by the displacer and its yoke is thus increased, since the buoyant effect of the displaced mercury is being decreased. The turning of *P* continues until a state of equilibrium is reached. Thus, for any given radius setting and counterweight position, there is a definite relation between the displacement of the arm *P*, and consequently the pen reading, and the actuating differential.

PART I THE USE OF THE DEADWEIGHT CHECK

Analysis of the flue gases by Orsat is the basis on which the meter adjustments are checked, and considerable difficulty had been experienced in obtaining reliable and consistent checks. When the radius was set according to the results of Orsat analyses at one rating, it was usually found that it was impossible to check the desired excess air at other ratings. This inconsistency was ascribed to almost every possible and impossible cause, among the principal ones being that the changing effect of chimney action in the second and third passes of the boiler made accurate setting impossible and that difference in fuel-bed condition caused decided differences in the differential draft for the same air supplied. It has been found that neither of these has any discernible effect on the accuracy of the meter when it is properly adjusted.

As a possible solution it was suggested that the meter be calibrated against draft directly, on the chance that the meter might not be indicating properly the rating corresponding to a given draft differential. This was to be accomplished by connecting a liquid-level draft gage in parallel with the air bells of the meter. At this time it was erroneously believed that, in general, the problem of the proper application of the meters had already been completely solved; the calibration of the meters against draft was to detect slight inconsistencies caused by the possibility of dirt or excessive friction in the meter.

Simultaneous gas analyses and draft differentials with coincident pen readings were found to be rather confusing. While individual Orsat checks were shown to be correct or incorrect, as the case might be, an inconsistent function of draft against rating was indicated, showing a minimum draft point at about 125 per cent of rating. As the rating decreased below 125 per cent, the differential draft increased. If this were a true indication of the calibration of the meters, they would be practically valueless below 150 per cent of rating. Such was probably the case for the condition in which the meters then were. However, there was so much uncertainty concerning the accuracy of the draft readings that this method was discarded in favor of the "deadweight check."

The "deadweight check" applies a method of computation, which later developments proved invaluable, by means of which the shape of the total air characteristic can be calculated for a series of rating points if the differential drafts at these ratings are known. This method assumes that, first, the resistance of the boiler as a conduit is constant for a given condition of cleanliness and that therefore the friction loss through it (in this case the differential draft between first and third passes) is exactly proportional to the square of the flow, and, second, that the actual evaporation per pound of coal is constant within the limits imposed by the accuracy of the measurements.

If, then, the differential drafts at any two ratings are known, the respective rates of air flow are proportional to the square

roots of the differentials. Also, if the coal burned is proportional to the rating, the quantity of air theoretically required for combustion is proportional to the rating, and the air flow is then proportional to the product of rating and per cent total air. (The term, per cent total air, as used in this report, is the ratio of the quantity of air actually supplied to the boiler to that theoretically required for combustion, expressed in per cent; or, in other words, it is 100 per cent plus the per cent excess air.) There are therefore two expressions—square root of differential, and product of rating and per cent total air, both proportional to the same quantity, air flow. If the change in one is known, the change in the other may be determined. Thus, for example, suppose that an Orsat analysis has shown that at 200 per cent of rating the total air is 116 per cent and the differential draft at this rating is 0.690 in. of water. If the differential at 90 per cent of rating is 0.154 in. of water, what is the per cent total air at 90 per cent of rating?

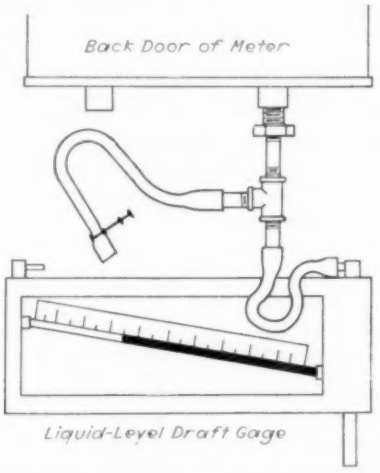


FIG. 3 MECHANISM FOR CALIBRATING AIR FLOW

Let F_1 = air flow at 200 per cent of rating
F_2 = air flow at 90 per cent of rating
$F_1 = 200 \times 116 = 232$

Now

$$\frac{F_2}{F_1} = \frac{\sqrt{0.154}}{\sqrt{0.690}}$$

So

$$\frac{F_2}{232} = \frac{\sqrt{0.154}}{\sqrt{0.690}}$$

$$F_2 = \frac{\sqrt{0.154}}{\sqrt{0.690}} \times 232 = 110$$

F_2 = 90 × per cent total air = 110

Therefore, per cent total air = 110/90 = 122 per cent at 90 per cent of rating.

Calibration of the air-flow mechanism by means of the method just outlined was then undertaken. This was accomplished by disconnecting the lines from the boiler, applying an artificial differential which is measured by a draft gage in parallel with the mechanism, and calculating the total air characteristic from the "deadweight check." The arrangement of the apparatus for this is shown in Fig. 3. The regular draft connections to the two nipples at the back of the meter are removed. The left-hand nipple, introducing draft under bell *B*, Fig. 2, is left open to atmosphere. The pipe fittings shown are connected to the right-hand nipple, which introduces draft under bell *A*, Fig. 2. In order to apply the artificial differential when checking the meter, air is sucked out through the rubber tube equipped with a pinch cock, thus creating the same suction under bell *A* and on the inclined draft gage. The air-pen reading and the corresponding draft-gage reading are recorded. (Instead of the temporary connections shown in Fig. 3, the draft gage may be permanently connected to the meter and the piping so arranged that it is not necessary to disconnect the meter, but only to remove two plugs and close two valves in order to obtain the simultaneous draft and pen readings.)

It is possible to obtain these draft and rating readings over the entire range of the meter in an hour or less. These readings may be plotted and the total air characteristic derived. If the characteristic is found to be incorrect, the trouble can be quickly diagnosed and the necessary steps taken to correct it. It may be well to remark here that the accuracy of the assumptions on which this method is based was verified by checking the characteristic by Orsat analysis. These data will be presented later.

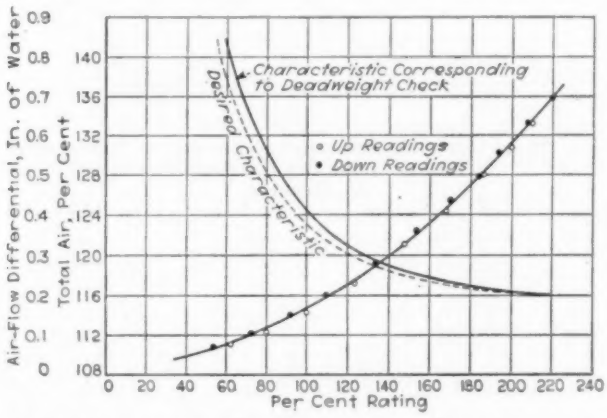


FIG. 4 DEADWEIGHT CHECK AND TOTAL AIR CHARACTERISTICS

A typical "deadweight check" is given in Table 1 as an example.

TABLE 1 TYPICAL DEADWEIGHT CHECK

BOILER METER NO.... PEN READING VS. DIFFERENTIAL DRAFT

Up		Down	
Air-pen reading	Draft-gage reading	Air-pen reading	Draft-gage reading
54	0.070	...	...
72	0.103	62	0.074
92	0.150	80	0.110
110	0.200	100	0.160
134	0.280	124	0.230
154	0.360	148	0.325
170	0.435	168	0.410
184	0.494	186	0.500
194	0.555	200	0.570
208	0.630	210	0.630
220	0.690	...	...

These readings are plotted in Fig. 4. The computation of the total air characteristic is given in Table 2 and is shown by the solid line in Fig. 4. The dotted line shows the desired characteristic.

These data show the characteristic to be a little too steep; that is, it is too high at the low ratings. (See Fig. 4.) The remedy is to move the counterweight *F* in Fig. 2 to the right—that is, to balance the meter slightly lighter at zero. If the value of total air as indicated by the deadweight were found to be too low at the low ratings—that is, less than 120 per cent at 120 per cent of rating—it would be necessary to weight the meter heavier at zero by moving the counterweight *F* to the left. It has been found possible to obtain a wide range of characteristics

TABLE 2 COMPUTATION OF TOTAL AIR CHARACTERISTIC ON THE BASIS OF DEADWEIGHT CHECK

Per cent rating[a]	Differential draft across airbells a	Differential in per cent of that at 200 per cent of rating	Per cent flow; square root of per cent differential	Flow in arbitrary units; product of per cent rating and per cent total air[b]	Per cent total air[a]
220	0.690	119.58	109.35	254.3	115.6
200	0.577	100.00	100.00	232.6	116.3
180	0.473	81.98	90.54	210.6	117.0
160	0.380	65.86	81.15	188.8	118.0
140	0.297	51.47	71.74	166.9	119.2
120	0.225	39.00	62.45	145.3	121.1
100	0.166	28.77	53.64	124.3	124.8
80	0.116	20.10	44.83	104.3	130.4
60	0.076	13.17	36.29	84.4	140.7

[a] See Fig. 4.
[b] 116.3 per cent is the desired total air at 200 per cent of rating.
$116.3 \times 200/100 = 232.6$.

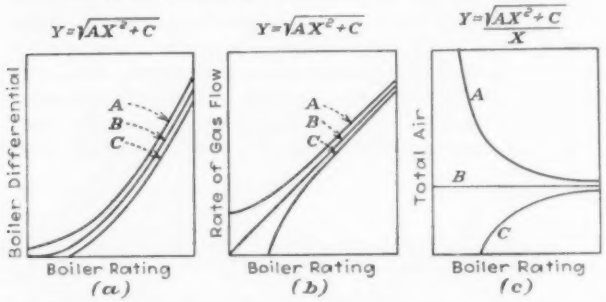


FIG. 5 THREE CURVES OF DIFFERENTIAL OBTAINED BY DEADWEIGHT CHECKS, WITH CORRESPONDING GAS FLOW AND PER CENT TOTAL AIR CURVES
(In the equation $Y = Ax^2 + C$, for $A, C = +K$; for $B, C = 0$; for $C, C = -K$.)

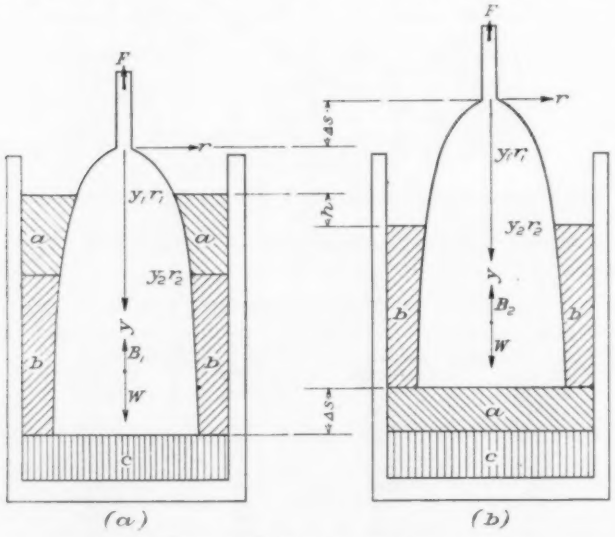


FIG. 6 DISPLACER PART OF THE MECHANISM

of total air with respect to rating, from a slightly rising one to a steeply falling one, simply by changing the balance and keeping the displacer and the mercury level in the displacer pot the same. This dependence of the characteristic on the weighting and its independence of the displacer are explained analytically in part II. The existence of this dependence had not previously been appreciated.

It is clear from this example, since the per cent total air at 200 per cent of rating is assumed, that the deadweight check determines only the shape of the characteristic curve of the meter; that is, it determines, for any radius setting, the relative values of per cent total air at different ratings. However, if the per cent total air at any one rating has been established by Orsat analysis and is used in making the deadweight calculations, the characteristic is exactly determined over the entire rating range. It is of course obvious that an assumed per cent total air value may be used, a satisfactory characteristic obtained by the deadweight method, and the Orsat analysis made last. If the Orsat analysis checks the assumed value within reasonable limits (1 to 2 per cent total air), the characteristic may be regarded to be satisfactory as originally determined. In any event, an Orsat analysis at one rating only is all that is required.

This procedure is unquestionably simpler, shorter, and more accurate than obtaining the number of Orsat checks necessary to cover the range, with the attendant probabilities of error.

PART II DETERMINATION OF THE PROPER DISPLACER CONTOUR

After making a number of deadweight checks and calculating the corresponding total air curves, it became evident that the curve of differential against boiler rating was always of the form expressed by the equation

$$y = a'x^2 + c_1$$

where y = differential
x = rating
a' and c_1 = determinable constants

In this equation the first term, $a'x^2$, represents, obviously, the differential caused by the flow of a quantity of air which is directly proportional to the rating; while the second, c_1, in reality the "weighting at zero," represents the constant differential caused by an additional quantity of air flow. Thus if the constant c_1 is zero, the flow of air is proportional to the rating, or the excess air will be constant. If c_1 is not zero, its effect will be more pronounced at the lower ratings (since the other term $a'x^2$ has a small value at low ratings) and progressively less pronounced as the rating increases. Thus when c_1 is positive, the air flow will increase from an initial constant value to values approximately proportional to the rating, giving a dropping total air characteristic; while if c_1 is negative, the characteristic will be a rising one. This is shown graphically in Fig. 5. Fig. 5(a) represents three curves of differential that might be obtained by deadweight checks. In curve B the value of c_1 is zero, in A it is a positive quantity, and in C it is a negative quantity. Fig. 5(b) represents the rates of air flow that would occur at various ratings with the corresponding deadweights, and Fig. 5(c) shows the corresponding total air characteristics. It is evident that the effect of the value of c_1 on the shape of the characteristic is of paramount importance.

Since the curve of differential is constant in form, it follows that there is only one combination of proper displacer shape and weighting to give the desired effect. To determine the displacer contour is then a problem in mechanics and the calculus.

If the arm P in Fig. 2 is considered as a free body, and any force from the pen arm through K is neglected, the equation of the force exerted by the displacer and yoke C and D must be of the same form as the equation of the differential, since that force must always exactly balance the turning force exerted by the air bells. The values of the constants a and C_2 in the new equation of force [1] may be determined from a' and c_1 for the equation of differential, using the lengths of the several arms P and L, the cross-sectional area of the draft bells, and the conversion factor for inches of water draft to pounds per square inch.

Let us now consider the displacer apart from the rest of the mechanism. In Fig. 6, F is the force acting on the meter beam. This force, as has been shown, may be expressed in the form

$$F = ax_2 + C_2 \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

Now since the displacer is a weight immersed in a fluid, the force F is

$$F = W - B \quad [2]$$

where W = the weight of the displacer, a constant
B = the buoyant force exerted by the fluid

Equating these expressions for F

$$ax^2 + C_2 = W - B$$

$$B = W - C_2 - ax^2 = K_1 - ax^2 \quad [3]$$

where $K_1 = W - C_2$

Suppose, now, that the displacer is moved upward a distance ΔS. The mercury level which before stood at the point y_1r_1 on the displacer now stands at y_2r_2, having dropped the distance h in the pot. The new buoyancy is B_2. Suppose that the distance ΔS is so small that r_2 is not appreciably different from r_1.

Then $$B_2 = B_1 - \delta\pi r^2 (y_2 - y_1) \quad [4]$$

where δ = density of the fluid displaced

Now let $y_2 - y_1 = \Delta y$

and $B_2 - B_1 = \Delta B$

Then $\Delta B = B_1 - \delta\pi r^2 \Delta y - B_1 = -\delta\pi r^2 \Delta y$ [5]

Then the average rate of change of B with respect to S is

$$\frac{\Delta B}{\Delta S} = -\delta\pi r^2 \frac{\Delta y}{\Delta S}$$

or passing to the limit as $\Delta S \longrightarrow O$, when $r_1 = r_2 = r$

$$\underset{\Delta S \longrightarrow 0}{\text{Lim.}} \frac{\Delta B}{\Delta S} = \frac{dB}{dS} = -\delta\pi r^2 \frac{dy}{dS}$$

or

$$\frac{dB}{dS} = -K_3 r^2 \frac{dy}{dS} \quad [6]$$

where $K_3 = \delta\pi$

Now in order to eliminate a variable it is advisable to valuate the quantity dy/dS. Referring to Fig. 6(a) it will be seen that the volume of fluid can be divided into three sections, a, b, and c. In Fig. 6(b) it is seen that b and c are unchanged in volume or conformation, though b has been moved upward in the pot; therefore, since the total volume of the fluid is unchanged, a in Fig. 6(b) must be equal to a in Fig. 6(a).

Hence if A is the cross-sectional area of the pot,

$$A\,\Delta S = (A - \pi r^2)(y_2 - y_1) = (A - \pi r^2)\,\Delta y$$

Then $$\frac{\Delta y}{\Delta S} = \frac{A}{A - \pi r^2}$$

And passing to the limit as $\Delta S \longrightarrow O$

$$\frac{dy}{dS} = \frac{A}{A - \pi r^2} \quad [7]$$

The change ΔS that was assumed is proportional to a change of the reading of the meter in terms of rating,

or $$\Delta x = K_2 \Delta S$$

and $$\frac{\Delta x}{\Delta S} = K_2$$

Then passing to the limit as $\Delta S \longrightarrow O$.

$$\frac{dx}{dS} = K_2 \quad [8]$$

Now differentiating Equation [3] with respect to S

$$\frac{dB}{dS} = -2ax\frac{dx}{dS} = -2aK_2 x = -K_4 x \quad [9]$$

where $K_4 = 2aK_2$

From this point, since there are two expressions for dB/dS which may be equated and an expression for dy/dS, it is simply a mathematical problem to determine r in terms of y, and S in terms of r or y. (See Appendix.)

The final equations are of the form:
for r in terms of y

$$r = \sqrt{M - \sqrt{\frac{N}{y - C}}} \quad [10]$$

for S in terms of r

$$S = \frac{PK_3^2 r^2}{a(A - \pi r^2)} \quad [11]$$

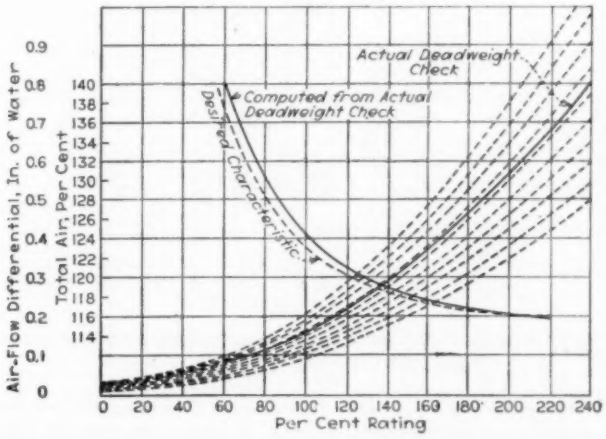


FIG. 7 IDEAL DIFFERENTIAL CURVES FOR SAME DESIRED AIR-FLOW CHARACTERISTIC

These equations are general for the air-flow mechanism in question, and the constants may be determined for any particular application.

The constants in these equations were determined for both the 400-lb.-pressure and 285-lb.-pressure boiler meters, and the graphs of the function of r with respect to y were plotted. These were compared with actual-size photographs of the displacers then in use. It was found that the displacers for the 285-lb.-pressure boiler meters checked the computed curve precisely. These displacers had been developed by cut-and-try methods by this organization some time previously. The displacers for the 400-lb.-pressure boiler meters did not check the desired contour, and it was necessary to make a templet of the computed contour and have new displacers made to that templet in the station machine shop.

From the equation of S in terms of r it was possible to determine the effect of a change of radius of the meter on the differential-rating function. From this computation it was possible to draw up a set of curves showing the differentials to be expected at various radius settings and to determine the intercept on the axis of differential—in other words, the weighting at zero—that is required for the given total air characteristic.

The family of computed differential curves for the 285-lb.-pressure boilers is given in Fig. 7. These curves show the change in differential to be expected when the radius is increased progressively in increments of 0.10 in. Any one of the computed curves of differential will produce the desired air-flow characteristic. If an actual deadweight is exactly similar to the computed curves—that is, coincident with one or dividing the interval between two in the same proportion over the entire range—it will give the desired total air characteristic.

From these computed deadweights it was then possible to determine the increment of total air that would correspond to a given change in radius when the condition of cleanliness of the boiler remains unchanged. This figure depends only on the range of radii and the normal total air. Its deviation from a constant over a range of rating is less than can be reliably measured by Orsat analysis. For use at the Hudson Avenue Station this constant has a value of 5 per cent increase in excess air per 0.1 in. increase in radius. The derivation of this factor has

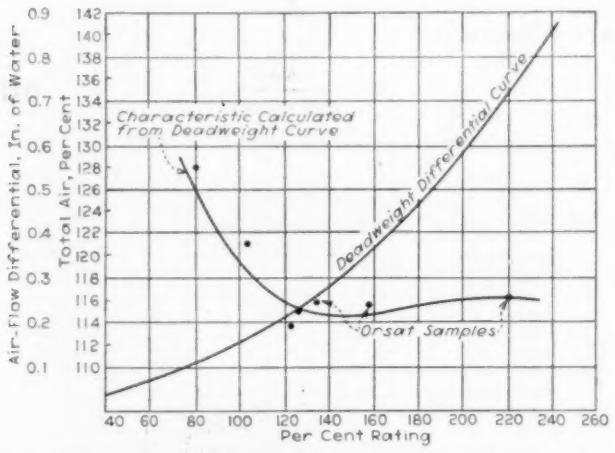


FIG. 8 COMPARISON OF TOTAL AIR CHARACTERISTIC BY DEADWEIGHT CHECK WITH FLUE-GAS ANALYSIS BY ORSAT

made unnecessary the use of the manufacturing company's comparatively complicated method involving the use of the "capacity factor."

PART III TOTAL AIR CHARACTERISTICS AND GAS-SAMPLING METHODS

It had been shown that with the correctly shaped displacer installed it was possible to control the total air characteristic entirely by manipulating the weighting at zero rating. It then became necessary to determine the characteristic desired.

This characteristic is a curve showing the excess or total air in per cent as a function of rating when the stoker is being operated with the air and steam pens together. This function determines, once the boiler and stoker are in service, the performance of the boiler; since the total air determines the stack loss directly, the refuse loss indirectly, and also the magnitude of the maintenance charges on the stoker and on furnace brickwork. Thus it is clear that assigning values of total air at which the stoker is to be operated is largely a matter of engineering judgment, rather than a problem that may be solved by precise mathematical reasoning. The principal factors to be considered are:

1 The excess air should be kept as low as possible without creating a loss due to incomplete combustion, represented by carbon monoxide in the flue gases or excessive combustible in the refuse.

2 The furnace temperature should be controlled by regulation of excess air so that the walls will not be unduly eroded by the high temperature gases or slag.

It has been found that it is possible to operate the 285-lb.-pressure boilers at as low as 110 per cent total air without any trace of CO in the flue gases at high ratings, but the furnace temperatures then run so high that the walls slag very badly. If the total air is increased to 116 per cent at 200 per cent of rating, excessive slagging of the walls stops. As the rating is decreased, the per cent total air may be kept approximately constant over an appreciable rating range, but by the time the rating has decreased to 120 per cent, the air necessary for combustion without a trace of CO has increased to 120 per cent, and at 100 per cent of rating about 123 per cent total air is necessary. Thus the characteristic that is desired for the 285-lb.-pressure boilers is that shown in Fig. 4. The meters of the 400-lb.-pressure boilers have been adjusted for approximately constant excess air throughout the operating rating range.

It then became necessary to prove that the work that had been done was justified, by demonstrating that it would pass the test "Does it work?" Everything had been based on the assumption that the differential was a true indication of the rate of air flow. The only possible check was by flue-gas analysis. The results of a series of analyses, run on a boiler that has been properly set by the deadweight check, must agree over the entire range with the characteristic predicted by the deadweight.

A series of Orsat checks was run on each boiler in the 285-lb.-pressure station to determine if these check points did coincide with the predicted curves. A typical one made before the desired total air curve had been fully agreed on is shown as Fig. 8.

Not all of the checks worked out as accurately as did this one. However, it was soon noticed that all the checks which showed inconsistent results were on boilers that were distant from the stack and that the single gas-sampling lines that were installed were on the side of the boiler away from the stack. The apparent variation of the Orsat analyses from the computed characteristics usually became greater on these boilers as the rating increased. These facts indicated the possibility that the error in the characteristic was only an apparent one, due to imperfection of the flue-gas-sampling technique rather than any true departure of these meters or boilers from the hypotheses already predicated. It was thought that, perhaps, the major gas flow was concentrated on the side of the boiler nearer the stack, due to draft conditions resulting from the offset location of the stack. If that were the case, samples drawn from either side alone would not be indicative of the average quality of the gases flowing through the boiler. It consequently became desirable to determine whether the single gas-sampling lines as then installed actually did draw representative samples of the flue gases. To do this, new sampling lines were installed in a boiler in which the difficulty was particularly serious. Two lines were installed from the north side, one running in for 8 ft. and one running in for 16 ft. This was opposite to the old single line which extended 8 ft. in from the south side. The furnace is approximately 24 ft. wide, and all three sampling lines are in the top of the third pass. The old 8-ft. line and the new 16-ft. line draw samples from about the same location. When Orsat checks were run, using all three lines, it was found that the long new line agreed fairly well with the old line, and the short new line gave widely different readings, which, when averaged with the readings from the opposite side of the boiler, checked the expected total air characteristic. The results of one of these tests are shown in Fig. 9.

When this condition was established, the necessity of a general installation of double sampling lines in all boilers was recognized. It should be kept in mind that not only must the third-pass lines provide a representative sample of the gases in that pass, but

this sample must be representative of the quality of gases flowing through the entire setting. Consequently, additional lines in the first pass to enable a check on the tightness of the setting have been found particularly helpful. With the new lines installed on all boilers, it has been possible to check the expected values of total air at all ratings and to set the meters with confidence. A series of checks, using double sampling lines in the third pass, made on one of the 285-lb.-pressure boilers after the desired characteristic had been established and checked by deadweighting, is shown in Fig. 10.

Additional data using two sampling lines, from one of the 400-lb.-pressure boilers, obtained for the primary purpose of determining the superheat curve, are included as Fig. 11. The total air characteristic was determined by a deadweight check made after the test runs had been completed and was plotted on the same graph as the results of the test.

The agreement between the total air characteristics as determined by the deadweights and checked by Orsat analysis, as illustrated in Figs. 8, 10, and 11, is perfect within the attainable

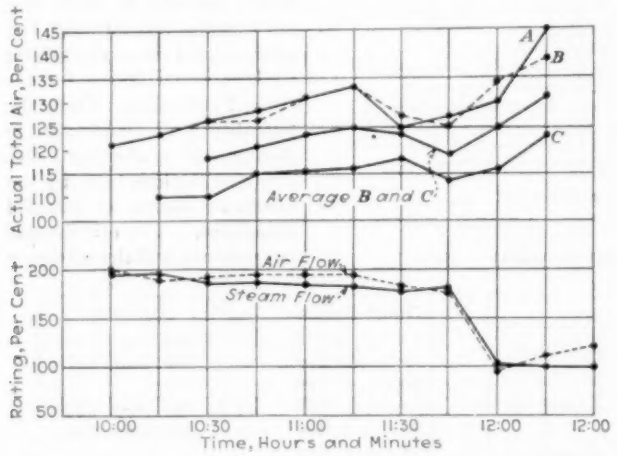


FIG. 9 COMPARISON OF SINGLE AND DOUBLE SAMPLING LINES
(Curve *A*, old line, 8 ft. in south side; *B*, new line, 16 ft. in north side, approximately same as *A*; *C*, new line, 8 ft. in north side.)

precision of measurement. The assumptions made in using the deadweight method and in deriving the displacer contour are thus proved valid.

PART IV ROUTINE FOR CHECKING METERS

As the boilers remain in service, accumulation of soot and slag on the tubes increases the draft differential between the meter connections for a given rate of gas flow. Consequently, for the same differential, the rate of gas flow will be decreased. The per cent total air with air and steam pens coincident will therefore be decreased. To counteract this action it is necessary to lengthen the radius of the meter, shown at *R*, Fig. 2.

Having established a method of meter calibration which permitted rapid and accurate manipulation of the characteristic, it became essential that a routine be developed that would insure the meters to be in proper adjustment at all times. Not only must the desired total air characteristic be determined in form, but its position must be permanently fixed (within predetermined limits) by means of successive changes of the radius length.

Moreover, a complete record of Orsat checks and radius changes with accompanying deadweight checks must at all times be kept available for easy reference.

The routine which has been developed will now be described. At intervals of ten days to two weeks, a flue-gas analysis by Orsat is run on each boiler in service. These checks are run by two men, one man analyzing samples from the 8-ft. third-pass sampling line while at the same time the other man analyzes samples from the 16-ft. third-pass sampling line and records the boiler-meter and draft-gage readings. The checks are made when the boiler is being operated so that the air and steam pens are approximately together. Five sets of samples are taken while conditions are constant, at any rating. A sample data sheet for this check is included as Fig. 12. From the Orsat analysis, the per cent total air is computed and corrected for any slight difference in the readings of the air and steam pens, to the condi-

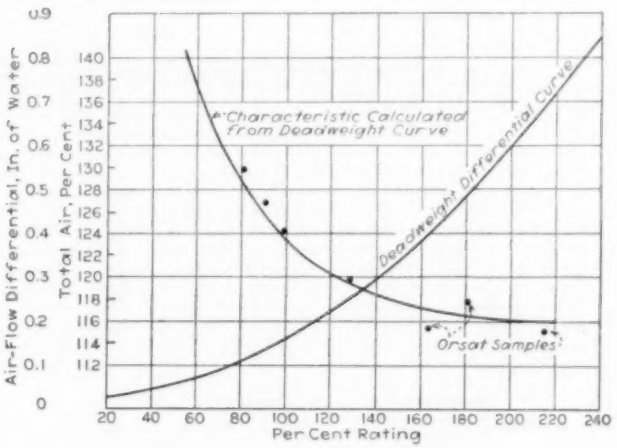


FIG. 10 COMPARISON OF TOTAL AIR CHARACTERISTIC BY DEADWEIGHT CHECK WITH FLUE-GAS ANALYSIS BY ORSAT

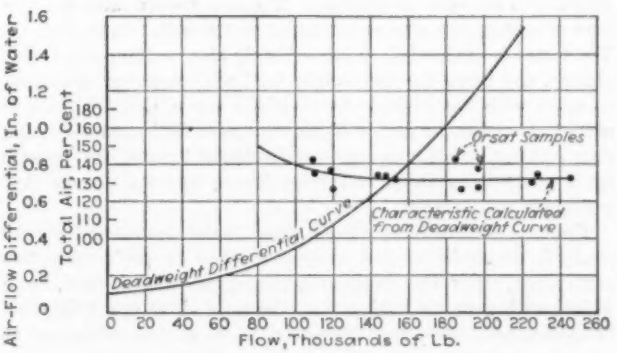


FIG. 11 COMPARISON OF TOTAL AIR CHARACTERISTIC BY DEADWEIGHT CHECK WITH FLUE-GAS ANALYSIS BY ORSAT

tion of pens exactly together. This is done by application of the formula

$$A_{pt} = A_a \times \frac{S_a}{P_a}$$

where A_{pt} = total air in per cent with pens together
A_a = actual total air, per cent
S_a = actual steam-pen reading
P_a = actual air-pen reading.

This gives the per cent total air for which the meter is set when the pens are together at the rating indicated by the air pen. The average of these data is then plotted on a sheet having on it the curve of desired total air versus rating. One of these sheets, similar to Fig. 13, which shows the record of boiler No. 14 for a three-month period, is kept for each boiler. It shows each Orsat check that is made on that boiler, its date, and the

radius at which the meter was set when the check was run. How closely the check approaches the desired condition is clearly indicated.

If an Orsat check shows that a meter is correct to within about 2 per cent total air, its setting is not changed. If the departure from the standard is greater than that limit, the radius is changed by a quantity computed from the factor of 5 per cent excess air per 0.1-in. change in radius. Radii are set only in multiples

HUDSON AVENUE GENERATING STATION
BOILER METER DATA
BOILER NO.....

RADIUS AT TIME METER WAS CHECKED..... DATE..... 192.....
DAY.....

TIME	STEAM FLOW L.	STEAM FLOW R.	AIR FLOW	STOKER TACH. PEN	STEAM TEMP.	FLUE GAS TEMP.	WIND BOX PRES.	DIFF'L DRAFT	FURNACE DRAFT	METER DIFF'L	GAS ANALYSIS NORTH SIDE CO_2	CO_2 O_2	CO_2 O_2 CO	ACTUAL TOTAL AIR	GAS ANALYSIS SOUTH SIDE CO_2	CO_2 O_2	CO_2 O_2 CO	ACTUAL TOTAL AIR	AVERAGE N&S. ACTUAL TOTAL AIR	TOTAL AIR WITH PENS TOGETHER

RADIUS RESET TO.....INCHES DATA TAKEN BY.....

FIG. 12 SAMPLE DATA SHEET FOR FLUE-GAS ANALYSIS BY ORSAT

of 0.05 in., which permits adjustment of the characteristic to within $1^1/_4$ per cent of total air. The new radius value and the date of making the change are noted on the boiler sheet, Fig. 13. The meter is deadweighted immediately after making the radius change, and when the deadweight has been computed and found to agree with the desired characteristic curve, a check mark is placed alongside the notice of the change of radius on the boiler sheet. After a radius change and deadweight check the meter is again checked by Orsat to further insure that the change that was made was correct.

The necessity of computing a great many deadweight checks made it desirable to find a short method of performing this calculation. For this purpose a nomogram was devised, from which, with a simple collineation, the total air at any rating can be determined, when the differential drafts at that rating and at a base rating are known. The nomogram for the 285-lb.-pressure boiler meters is shown with the solution of the deadweight check in Part I, as Fig. 14. On this nomogram, 200 per cent of rating was chosen arbitrarily as a convenient base, but any other value could have been used. The nomogram greatly facilitates the use of the deadweight check as a routine method.

This routine makes it possible to take care of the worst condition of any boiler in four or five hours, which allows one hour each for the Orsat checks and an hour or two, which is more than ample, for the radius change, deadweight, and its calculation. This is a distinct advance, since with the old methods of checking two or three days might be spent on one boiler, and even then there was no assurance that the meter was correctly set over the entire range.

With these methods of checking in use, only about one Orsat check in 35 gives inconsistent results, and these inconsistencies invariably disappear on additional checking. This proportion is well within the probability of error of men and instruments.

In addition, the use of this system has had some valuable by-products. It has shown that the radius of the meter need not be changed so often, nor by such large increments, as was formerly thought to be necessary. The maximum increase, over the normal period of three months between boiler cleanings, of the radius of any 285-lb.-pressure boiler meter is about 0.20 in. This corresponds to an increase in draft differential of about 20 per cent, or from about 0.60 in. of water to 0.72 in. of water at 200 per cent of rating. This is a reasonable increase, since the meter draft taps are at the top of the first and third passes, between which points there can be only a relatively light accumulation of soot as the boiler gets dirty. Most of the increase in boiler-draft loss must occur in the first pass of the boiler, and this does not affect the meter.

Knowing the normal draft drop at 200 per cent of rating for a clean boiler, it has been found possible to reset the meter radius and establish the desired total air characteristic while a boiler is out for external cleaning. Only the formality of an Orsat check is then necessary, after the boiler comes on the line, to verify the estimated degree of external cleanliness. The exit-flue-gas temperature and the efficiency

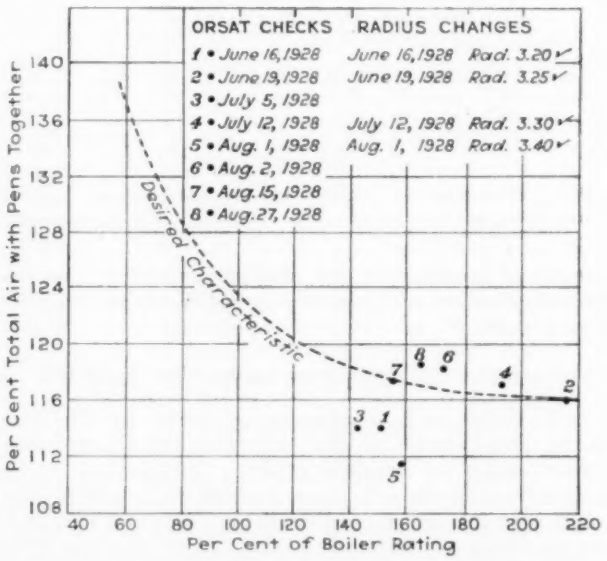


FIG. 13 RECORD OF ORSAT CHECKS AND RADIUS CHANGES

of the boiler are used as independent measures of the cleanliness of the boiler.

Furthermore, the indicated radius changes furnish a very good check on the condition of the baffling. It is apparent that if the radius change indicated by an Orsat check is not so great as it should be when considering the time that the boiler has been in service, it may very probably be that the baffling in that boiler is in bad condition.

CONCLUSIONS

The conclusions to be drawn from the data presented are

various. They cover (1) the accuracy of the use of the drop in pressure through a boiler as an indicator of rate of flow of gases and the practicability and accuracy of a meter designed to measure this flow, (2) the amount of work necessary to maintain such a meter at its maximum accuracy, and (3) the value of such a meter in conjunction with a steam-flow meter as a guide to combustion. These subjects will now be considered:

1 The difference in draft between the first and third passes of the boiler is an accurate measure of the gas flow, and a mechanism actuated by this differential draft can be constructed to indicate accurately the corresponding rate of air flow. This is evident, well this end has been accomplished can best be appreciated by the changed attitude of the operating personnel toward the efficiency of the meters as accurate combustion guides.

Results

The most important results of this work on the air-flow mechanism have been:

1 The maintenance of the stack losses at a desirably low level. This has occurred because the meters are now accepted as giving a true indication of the per cent total air. Before this work was done, a usual excuse for operating with the meter pens apart

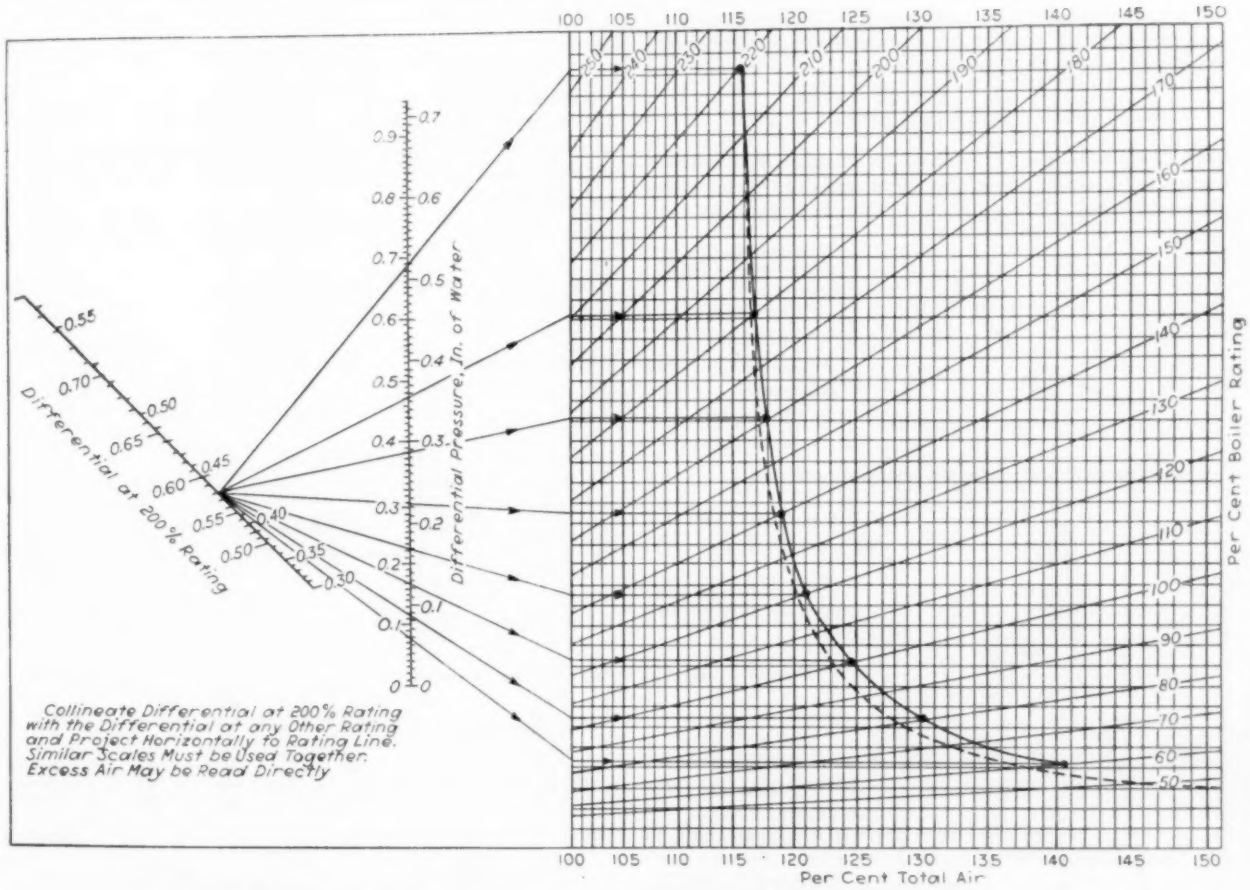


Fig. 14 Nomogram for Total Air Characteristic, 285-Lb. Boiler Meter

(Collineate differential at 200 per cent rating with the differential at any other rating and project horizontally to rating line. Similar scales must be used together. Excess air may be read directly.)

since the rate of flow predicted by the deadweight check is in such exact agreement with the flow as determined by Orsat analysis. Although, in this paper, only B. & W. three-pass cross-drum stoker-fired boilers have been considered, the principles will apply to other installations, and the displacer contour may be deduced rationally from the conditions under which the meter is to operate.

2 The maintenance of the air-flow mechanism has, by the application of the method outlined in this paper, been reduced to a completely checked routine that requires only the attention of two men for from three to five hours a month per meter.

3 The air-flow meter, maintained as described, is, in conjunction with a properly kept orifice steam meter, an absolutely reliable guide to combustion conditions. The attainment of this end was the motive for undertaking the work described. How was that the meter was not set correctly. While the meters were not always incorrect, they were so in a sufficient number of cases to cause them to be regarded as unreliable. Now, however, one can safely say that the boiler-meter indication is correct within the limits of 1 to 2 per cent of total air. The boiler-room engineers, finding that they can rely absolutely on the indication of the boiler meter, make it their business to see that the stoker attendants keep the pens together on the chart. If the pens are not kept together consistently, there is no longer any excuse; it means simply that the fires are not in proper condition. Naturally the boilers are now operated at total air values closely approximating desired conditions at all times.

2 From the point of view of calibration and adjustment of the meters, an equally great advance has been made. The maintenance of the meters in proper calibration has been rendered less

laborious as well as more effective. In a large power plant this is no small item.

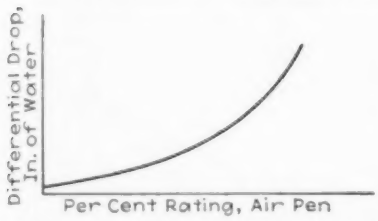
3 The ability to use the record of radii changes as a check on the condition of the exterior of the boiler tubes and baffling also represents an advance.

4 Moreover, the proper installation of meters in any extension to the boiler house is prepared for. Knowing the probable draft loss through the boiler at any rating, it is possible to determine in advance the characteristics of the air-flow mechanism for that application.

No attempt has been made to determine the monetary equivalents of the items outlined in the foregoing. It is felt, however, that it will be conceded that their beneficial influence on plant performance and cost of operation represents ample justification for the efforts involved in their attainment.

Appendix

From a series of points obtained by running a deadweight check on No. 34 boiler meter, the differential drop versus rating curve was obtained. The equation of this curve was found to be

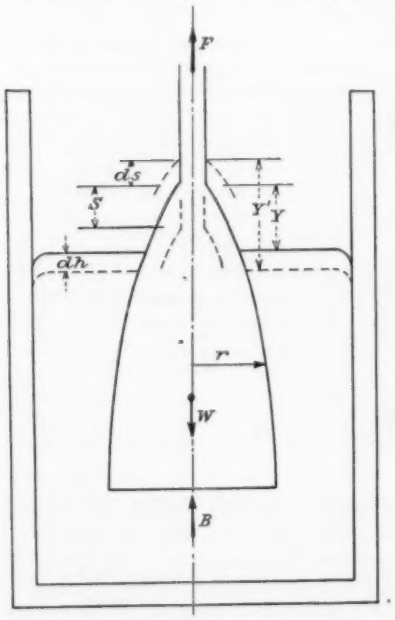


$$D = dx^2 + C_1 \ldots [1']$$

where

D = differential drop (top of first to top of last pass)
x = air-pen rating
C_1 = y-axis intercept

F is the force which actuates the air-flow mechanism.



$$F \propto D, \text{ or } F = ax^2 + C_2 \ldots [1]$$

Now from the displacer we have at any point

$$F = W - B \ldots [2]$$

where

F = actuating force
W = weight of displacer
B = buoyancy of fluid

Equating [1] and [2], we have

$$W - B = ax^2 + C_2$$

$$B = W - ax^2 - C_2$$

$$B = k_1 - ax^2 \ldots [3]$$

where $k_1 = W - C_2$

Let us now assume S = the distance vertically above zero position of the displacer at any linear displacement. The buoyancy is B as previously assumed. The vertical length of the displacer above the mercury level at the displacement S is Y. Let us also assume the displacer is moved upward a small distance ds. The mercury level will drop a distance dh. Then $Y' - Y = ds + dh$.

Also, let us assume the distance dy is so small that the radius r does not change appreciably. Then, at the new displacement, the buoyancy will be B', where $B' = B - \delta\pi r^2 dy$.

or

$$B' - B = -\delta\pi r^2 dy \ldots [4]$$

where δ = density of fluid (mercury)
$r = f(y)$

Also let $B' - B = dB$

then $$dB = -\delta\pi r^2 dy \ldots [5]$$

or $$dB = -k_2 r^2 dy$$

where $k_2 = \delta\pi$.

Dividing by dS,

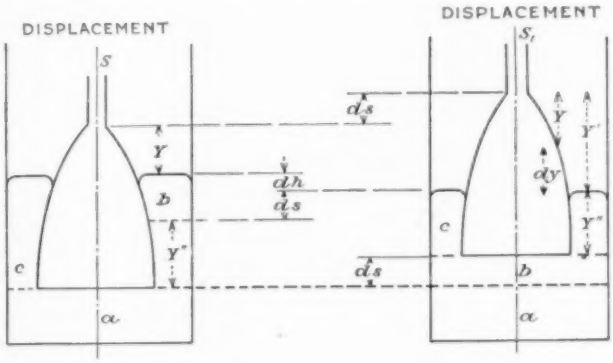
$$\frac{dB}{dS} = -k_2 r^2 \frac{dy}{dS} \ldots [6]$$

Let us assume A = area of mercury pot. From the diagram the mercury may be divided into three sections, a, b, and c; $a = a$, $b = b$, $c = c$ in both cases, since the total volume is unchanged, presupposing the mercury pot is a perfect cylinder.

Then since $b = b$

$$A\,dS = (A - \pi r^2)\,dy$$

$$\frac{dy}{dS} = \frac{A}{A - \pi r^2} \ldots [7]$$



We then have

(1) $B = k_1 - ax^2$ (2) $\frac{dB}{dS} = -k_2 r^2 \frac{dy}{dS}$ (3) $\frac{dy}{dS} = \frac{A}{A - \pi r^2}$

and

$$\frac{dB}{dS} = -k_2 r^2 \frac{A}{A - \pi r^2} = -\frac{k_2 A r^2}{A - \pi r^2}$$

also

$$\frac{dB}{dS} = -2ax \frac{dx}{dS}$$

then

$$-\frac{k_2 A r^2}{A - \pi r^2} = -2ax \frac{dx}{dS}$$

and

$$\frac{k_2 A r^2}{A - \pi r^2} = 2ax \frac{dx}{dS}$$

Also $S = k_3 \times x$ (vertical displacer motion varies as pen reading)

$$dS = k_3 dx$$

$$\frac{dx}{dS} = \frac{1}{k_3} = k_4$$

Then

$$\frac{k_2Ar^2}{A-\pi r^2} = 2a\,\frac{S}{k_3}\,k_4 \quad \text{or, } S = \frac{k_2Ar^2}{A-\pi r^2} \times \frac{k_3}{k_4} \times \frac{1}{2a}$$

$$= \frac{k_3^2k_2Ar^2}{2a(A-\pi r^2)} = \frac{k_3^2k_2Ar^2}{2aA-\pi r^2 2a}$$

Let

$$k_3^2k_2A = k_5 = k_3^2\delta\pi A$$

$$2aA = k_6$$

$$2a\pi = k_7$$

Then

$$S = \frac{k_5r^2}{k_6 - k_7r^2}$$

and

$$dS = \frac{(k_6 - k_7r^2) \times 2k_5rdr - k_5r^2(-2k_7rdr)}{(k_6k_7r^2)^2}$$

$$\frac{dS}{dr} = \frac{(k_6 - k_7r^2) \times 2k_5r + 2k_5r \times k_7r^2}{(k_6 - k_7r^2)^2}$$

$$= \frac{2k_6k_5r - 2k_5k_7r^3 + 2k_5k_7r^3}{(k_6 - k_7r^2)^2} = \frac{2k_6k_5r}{(k_6 - k_7r^2)^2}$$

and

$$\frac{dy}{dr} = \frac{2k_6k_5r}{(k_6 - k_7r^2)^2} \times \frac{A}{A-\pi r^2}$$

$$\frac{dy}{dr} = \frac{2Ak_5k_6r}{Ak_6^2 - 2Ak_6k_7r^2 + Ak_7^2r^4 - \pi k_6^2r^2 + 2\pi k_6k_7r^4 - \pi k_7^2r^6}$$

$$= \frac{4A^3ak_3^2k_2r}{4A^3a^2 - 8A^2a^2\pi r^2 + 4Aa^2\pi^2r^4 - 4A^2a^2\pi r^2 + 6Aa^2\pi^2r^4 - 4a^2\pi^3r^6}$$

$$= \frac{A^3k_3^2k_2r}{a(A-\pi r^2)^3}$$

Let

$$k_8 = \frac{k_3^2k_2}{a} = \frac{k_3^2\delta\pi}{a}$$

Let

$$k_9 = k_8A^3$$

Then

$$\frac{dy}{dr} = k_8\,\frac{A^3r}{(A-\pi r^2)^3} = k_9\,\frac{r}{(A-\pi r^2)^3} \quad \text{and} \quad dy = \frac{k_9rdr}{(A-\pi r^2)^3}$$

Let

$$U = A - \pi r^2$$

Then

$$dU = -2\pi rdr \quad \text{and} \quad dy = \frac{k_9 2\pi rdr}{2\pi U^3}$$

Then

$$dy = \frac{k_9dU}{-2\pi U^3} = \frac{k_9dUU^{-3}}{-2\pi}$$

and

$$y = \int \frac{-k_9U^{-3}dU}{2\pi} = \frac{-k_9}{2\pi} \times \frac{U^{-2}}{-2} + C$$

$$d\,\frac{U^{-2}}{-2} = \frac{1}{-2} \times -2U^{-3}dU = U^{-3}dU$$

$$y = \frac{k_9U^{-2}}{4\pi} = \frac{k_9}{4\pi U^2} = \frac{k_8A^3}{4\pi U^2}$$

$$y = \frac{k_8^2 \times \delta\pi A^3}{4\pi aU^2} + C = \frac{k_3^2\delta A^3}{4a(A-\pi r^2)^2} + C$$

We then have:

$$(a) \quad S = \frac{k_3^2\delta\pi Ar^2}{2a(A-\pi r^2)} + C$$

$$(b) \quad y = \frac{k_3^2\delta A^3}{2 \times 2a(A-\pi r^2)^2} + C$$

Dimensional check:

$$\text{in.} = \frac{\text{in.}^3 \times \frac{\text{lb.}}{\text{in.}^3} \times C \times \text{in.}^2 \times \text{in.}^2}{\text{lb. (in.}^2 - C \times \text{in.}^2)} = \text{in.}$$

$$\text{in.} = \frac{\text{in.}^3 \times \frac{\text{lb.}}{\text{in.}^3} \times \text{in.}^6}{\text{lb.} \times (\text{in.}^2 - C \times \text{in.}^2)^2} = \text{in.}$$

Final Equations

$$F = ax^2 + C_2 \qquad [1]$$

$$F = W - B \qquad [2]$$

$$B = W - ax^2 - C_2 \qquad [3]$$

$$S = k_9x \qquad [4]$$

$$S = \frac{k_3^2\delta\pi Ar^2}{2a(A-\pi r^2)} + C \qquad [5]$$

$$y = \frac{k_3^2\delta A^3}{4a(A-\pi r^2)^2} + C \qquad [6]$$

$$r = \sqrt{\frac{A}{\pi} \pm \frac{Ak_9}{2\pi}\sqrt{\frac{\delta A}{a(y-c)}}} \qquad [6']$$

Discussion

A. E. Livingston.[2] It is evident that the paper is the result of much experience with boiler meters and of real constructive thinking. There are a few questions that the writer would like to ask regarding points in the paper. In one place it reads: "Simultaneous gas analyses and draft differentials with coincident pen readings were found to be rather confusing. While individual Orsat checks were shown to be correct or incorrect, as the case might be, an inconsistent function of draft against rating was indicated, showing a minimum draft point at about 125 per cent of rating. As the rating decreased below 125 per cent, the differential draft increased. If this were a true indication of the calibration of the meters, they would be practically valueless below 150 per cent of rating. Such was probably the case for the condition in which the meters then were. However, there was so much uncertainty concerning the accuracy of the draft readings that this method was discarded in favor of the 'deadweight check.'"

Did the calibration of the meter give such a rapid increase in total air as the rating dropped below 125 per cent that the actual volume of air passing through the boiler increased? If not, then does not this contradict the assumption made in the next paragraph, that the differential draft between the first and third passes is exactly proportional to the square of the flow?

The assumption is made that the differential draft is exactly

[2] Efficiency Engineer, Staten Island Edison Corporation.

proportional to the square of the flow, the boiler acting as a conduit. Is this based on volume or weight of gases? What will be the effect of change in specific gravity of the gases due to change in temperature and pressure? The change in temperature and pressure may be caused by either change in total air, change in rating, or both.

One of the reasons given for using the "deadweight check" method is that it is more accurate than obtaining a number of Orsat checks to cover the range. By using only one Orsat check,

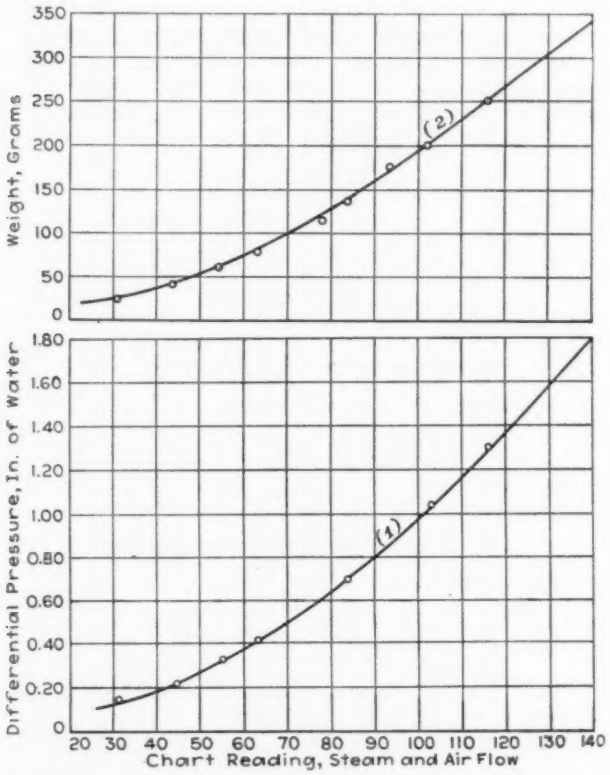


FIG. 15 AIR-FLOW SETTING, BAILEY METER, BOILER NO. 1
(Curves for 120 per cent, all ratings. Curve 1, differential pressure on air-flow bells; Curve 2, weights to be used in calibrating air-flow mechanism.)

is not the possibility of error increased? If a number of Orsat checks are made and plotted, error in any set will show up.

The author's whole procedure is built up on the assumption that the square root of the differential varies directly as the product of rating times total air. If this is true, as he has satisfied himself it is, why cannot the following short cut method be used? Run two tests at different ratings, plot square root of differential against product of rating times total air, and draw a straight line through the two points. From this line any differential could be computed for any rating and any total air. Data could readily be obtained for a differential versus rating curve, and corresponding grams-weight versus rating curve and the meter set accordingly. The grams-weight curve is used because it is easier to change weights on the bell than it is the air pressure under it when calibrating the meter.

The calibrating of the meter to give desired total air is the objective of the operating man. If he cannot succeed in doing this with the displacer supplied, would not the manufacturer's engineers be better fitted by experience to redesign the displacer? Would not the grams-weight versus chart reading curve be sufficient data for the design of the displacer by the manufacturer's engineers—assuming, of course, that they have complete details of the meter?

It is certainly an accomplishment for a test engineer, with the many demands on his time and attention, to work up the involved computations necessary for the design of an air-flow displacer. The author is to be commended very highly.

The experiences of the Staten Island Edison Corporation with boiler meters and the method of obtaining the desired calibration are outlined in the following covering the practice of the Test Department of the company.

AIR FLOW

These are a few of the questions that the test engineer responsible for boiler meters has to consider: How many trials must be made before one can get a final satisfactory adjustment on the boiler meter? Can one get a setting which gives the desired total air (either constant or variable) at all ratings? How near representative of average are the flue-gas samples used? What effect on the setting does variation in feedwater temperature have?

The Livingston Station has two bleeder-type turbines normally

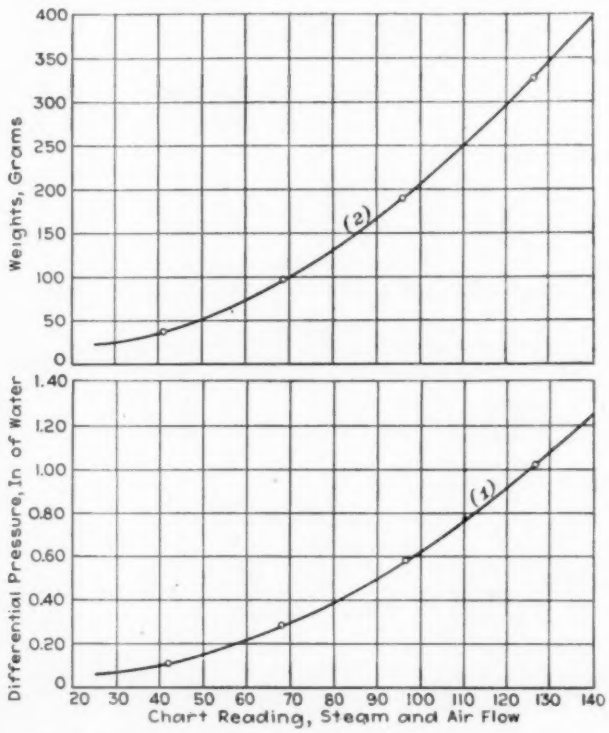


FIG. 16 AIR-FLOW SETTING, BAILEY METER, BOILER NO. 3
(Curves for varying percentage total air; straight line, 130 per cent at 100 per cent rating, 120 per cent at 350 per cent rating. Curve 1, differential pressure on air-flow bells; Curve 2, weights to be used in calibrating air-flow mechanism.)

operated with feedwater leaving heaters at approximately 230 deg. fahr. (varying with load). When one of these turbines is out, a non-bleed turbine is operated with condensate temperature about 70 deg. fahr. The variation in feed temperature and resultant variation in steam temperature cause the factor of evaporation to vary from 1.12 to 1.20, or 7 per cent. Thus the steam flow for a given heat absorption and a corresponding fixed air flow may vary 7 per cent.

Setting the meters, according to instructions of the manufacturer, for desired total air at 300 per cent rating has given much too high total air at lower ratings on two meters and much too low on one meter. Setting according to the manufacturer's instructions will be referred to as the "capacity factor" method.

Numerous Orsat readings were taken in the attempt to obtain satisfactory setting by the capacity-factor method. The results were unsatisfactory, for two reasons—one is explained in the foregoing paragraph and the other was due to taking the flue-gas sample from one point only (one-third of width of boiler setting from one side). With a perfect fire, the sample from one point only might be representative, but fires, like human beings, may be good but are never perfect.

For obtaining total air two Orsats are now used—one on each side of the boiler, with the sample line one-third of the way across the setting. Total air thus obtained varies from a close check to 20 per cent difference (120 per cent on one side, 140 per cent on other side) in the average of ten readings.

A direct method has been found to be positive in giving the desired meter calibration. Determine the desirable total air and take readings at four ratings, approximately 100, 175, 250, and 325 per cent.

Three men are required to obtain the test data. Two men operate Orsats, compute total air, and read steam temperature. One man operates the stoker and reads steam flow, steam pressure, differential pressure across bells of meters, and feedwater temperature. (Other readings are taken; only those noted are used for setting the air-flow mechanism.)

Total air values obtained from Orsat readings are given immediately to the man operating the stoker, who makes necessary adjustments to keep the total air close to the desired value.

For given heat absorption, three variables affect the steam flow as indicated by the meter—pressure, superheat, and factor of evaporation. If either or all of these items were not normal during test, corrected steam-flow values are computed. The corrected values are used in plotting meter curves.

Two curves are plotted, differential pressure and grams weight, against the corrected chart readings (see Figs. 15 and 16). Grams weight is obtained by converting inches of water to equivalent grams and multiplying by area of air-flow bell. These computations may be checked by taking readings of the air-flow pen during tests, then applying the computed weight on the air-flow bell, with lines to the boiler disconnected, before any change is made. It is desirable to take a differential reading to eliminate possible error due to friction or rubbing in the air-flow mechanism.

Air-pressure lines from the boiler are disconnected, and the air-flow radius is changed until weights on the meter bell produce chart readings in accordance with the curve. If the meter cannot be made to check the curve properly at both high and low ratings by changing the radius, then increasing or decreasing the mercury in the displacer reservoir and shifting the rider weights will usually make possible the following of the curve very closely.

It should be noted that by this method only one series of readings is taken and that the work can be completed by one man after the readings are taken. No check readings are needed as the check lies in the grams-weight versus the steam-flow curve. By this method the meter can be set for normal conditions regardless of the feed and steam temperatures prevailing at time of test.

Stoker Speed

The meter is equipped with a tachometer for recording the stoker speed. The mechanism in the meter is similar to the air-flow mechanism and may be calibrated by using the grams-weight versus chart reading curve similar to that used for the air flow.

The ideal condition would be to have the stoker-speed pen calibrated to the steam-flow pen as is the air flow. Having three pens recording together would make the record confusing. In order to have the meter serve the fireman as a guide for the stoker speed and give a record for checking purposes, without having a confusing and unreadable chart, the stoker pen is calibrated to stay a definite distance away from the steam-flow pen. At the Livingston Station the full range of the chart is seldom required for the steam flow, so the stoker pen is set five spaces above the steam flow. Five spaces are used because every fifth line is a major chart division. Zero for this pen is then one major space above chart zero.

To obtain the grams-weight versus chart reading curve, there are first taken the readings of stoker speed and differential pressure acting on the stoker-speed bells. The differential pressure is converted to equivalent force, in grams, acting on the bells. This data is plotted as shown in curve 1 on Fig. 17.

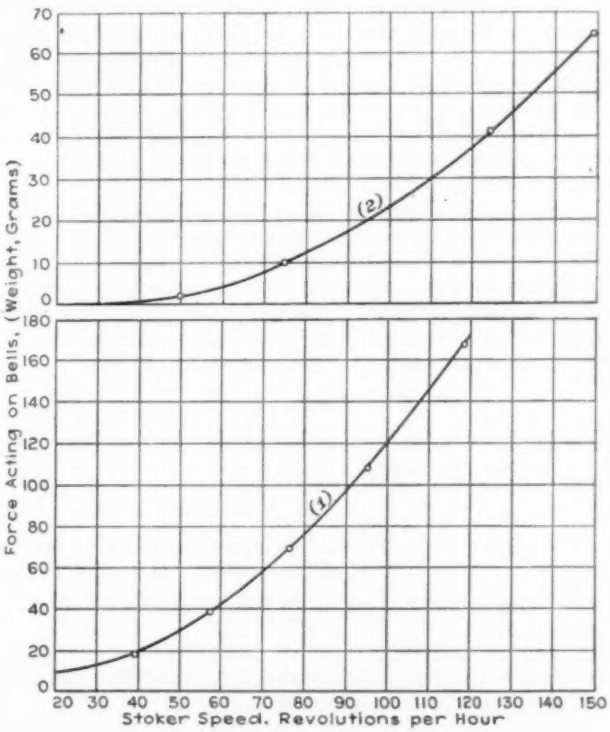


Fig. 17 Boiler Stoker Tachometer

There are known the average amount of coal fed to the boiler per revolution of the stoker and the pounds of steam produced per pound of coal. The steam flow as recorded by the meter has been found to be very inaccurate, so it was necessary to check each meter separately against a feedwater meter. A correction factor thus obtained is used in computing stoker speed required for various steam-flow pen readings.

Suppose there has just been found the stoker speed necessary to keep the steam-pen reading at 50 (one thousand pounds) on the chart. From curve 1, Fig. 17, there is found the force in grams acting on the meter bell at this speed. When the steam flow reads 50 (one thousand pounds), it is desired that the stoker pen should read 75; thus there is shown one point on curve 2. Other points are obtained in the same way.

The meter mechanism is then adjusted, using curve 2, following a procedure similar to that used for adjusting the air-flow mechanism.

A. A. Markson.[3] The writer hopes that it will please the author to know that this excellent paper has stimulated some lively thinking on the subject in the writer's organization.

[3] Mechanical Engineer, New York Steam Corporation, New York, N. Y. Jun. A.S.M.E.

Unless the writer has misunderstood the paper, he caught the fundamental point of the work as follows: The weight of gases passing through the setting may be proportioned to the square root of the draft loss across a properly selected portion of the setting by means of an intercept equation of the form:

$$Y = A'X^2 + C_1$$

The problem is, therefore, reducible to the determination of the general equation of the displacer so that the curve satisfies A' which is implicit in Equation [10].

It is obvious that C_1 is of the nature of an intercept comparable to stack effect. Unless by design, it can be made sensibly equal to zero, and it would appear, for the case given by the author, C_1 reduces to zero, for constant excess air, and need not be considered because when X is zero, C_1 is zero in his boiler (Fig. 1).

The validity of his calculations, therefore, depends on the assumption that A' is fundamentally a constant. The writer believes the fact that it worked out as a constant should be considered in the light of being a special case.

The constancy of A' depends on the assumption that the differential draft loss is proportional to the square of the gas flow. This seems only true if the heat transmission in the boiler, or rather the mean temperature of the gases under consideration, did not change with the flow, or at least was of a very small order of change. This is generally by no means so. Let us write for the weight of gas passing the draft apparatus per second

$$M_1 = FA(2g)^{1/2}D_1H_1^{1/2}$$

in which M_1 = weight of gases per second
F = a coefficient of resistance
A = an area
D_1 = mean density of gas
H_1 = the differential in homogeneous feet of gas in question and is equal to $D_1H_1/62.4$ feet of water

But $$D = D_0\frac{460}{T_1}$$

where D_0 is the standard density of the gas and T_1 is the absolute temperature, °R corresponding to D_1.

The expression for M_1 reduced to draft loss in feet of water is therefore

$$M_1 = FA(2g)^{1/2}D_0\frac{460}{T_1}\left(\frac{T_1}{460\,D_0}\times 62.4\times HW\right)^{1/2}$$

in which HW_1 is the draft loss in feet of water, or $M_1 = KT^{-1/2}HW_1^{1/2}$ constants being combined. For any other value of M, if T also changes

$$M_2 = KT_2^{-1/2}HW_2^{1/2}$$

In other words,

$$\frac{M_1}{M_2} = \frac{(T_2)^{1/2}}{(T_1)}\frac{(HW_1)^{1/2}}{(HW_2)} \quad \ldots\ldots\ldots\ldots\ldots [1]$$

rather than

$$\frac{M_1}{M_2} = \frac{(HW_1)^{1/2}}{(HW_2)} \text{ as assumed} \quad \ldots\ldots\ldots\ldots [2]$$

Using [1] instead of [2] is easy of application to any boiler. Undoubtedly it may explain the inconsistency of the curve which gave a higher draft at a lower rating. It is rational, for one thing. The effect of viscosity of gases can be neglected, in the writer's opinion, for it is not the absolute viscosity which counts, but the kinematic viscosity which does not change as much. The assumption that the actual resistance is sensibly constant over a limited range seems reasonable to the writer.

C_1, if pure stack effect, had best be made zero in the installation if possible, as it will also vary with changes of temperature. It is usually easy to do this.

The writer would like to comment on the general scheme of using the boiler meter at high ratings on stokers. The author's statement is that excess air should be kept as low as is compatible with furnace temperatures and ashpit combustible. This might be modified as applying particularly to high rates of combustion as follows: The excess air should be kept as low as furnace temperatures will allow, until a point is reached at which the gain due to decreased cinder carry-off with lowered excess air is offset by increased ashpit and CO losses. There is no doubt that the effect of cinder carry-over increases much faster at high excess air than the compensating effect of lower ashpit combustible. Of course, smoke is another factor which may have a somewhat more indefinite bearing on efficiency than the previous factors spoken of, but which nevertheless must modify combustion conditions in accordance with the plant location.

H. M. Hammond.[4] The author has given an excellent paper and is to be commended for the valuable and interesting data which he has presented. As a representative of the manufacturer of the boiler meter, the writer wants to say that we are heartily in favor of having customers make careful studies of the adjustments to be made in this meter. By going into this problem as thoroughly as he has, the author has insured that his company will realize a maximum return on their investment in boiler meters. The boiler meter, after all, is only a means to an end. When properly installed and adjusted, it will show what the operating conditions are—whether the proper amount of air is being used for best economy; and it will show the changes which should be made to improve efficiency. The results obtained are, in a measure, dependent upon the interest and attention given the meters by the operators.

The meter company renders from one to two days' service work per meter in connection with every boiler-meter installation, but it cannot possibly render sufficient time to do the thorough and complete job which the author has done, without making an additional service charge for such work. A staff of service engineers is maintained in all of the company's branch offices located in the principal cities throughout the country which is available for this service when desired.

Several years ago the most common and the largest controllable loss in boiler operation was due to the use of extremely high excess air. In attempting to reduce this loss it was only considered necessary to reduce the quantity of air used so as to maintain a lower but substantially constant excess air at all boiler ratings. With the comparatively small ranges over which the boilers were operated this was not difficult to do.

With the advent of modern stokers, pulverized-coal firing, turbulent burners, highly preheated air, water-cooled furnace walls, and other important developments, permitting the operation of boilers over much wider ranges in rating, other important limiting factors in determining the proper excess air to use for best overall results have presented themselves. Some of these factors are smoke, unburned-gas loss, refractory deterioration, the burning of stoker parts and burners, and carbon loss to the stack or into the ashpit. In at least one instance of which the writer has knowledge, the most economical excess air proved to be that which would give the desired reheat-steam temperature. In other words, in this particular case it was found that by using a higher excess air than was conducive to best combustion efficiency, in order to increase the reheat-steam temperature, the

[4] Sales Manager, Bailey Meter Company, Cleveland, Ohio. Assoc-Mem. A.S.M.E.

turbine efficiency was increased by more than enough to offset the decreased combustion efficiency.

It has been found that the above-mentioned factors or losses appear at different boiler ratings, and the magnitude of each varies with rating, so that in recent years it has become necessary, in many cases, to adjust the air-flow mechanisms in boiler meters for a variable excess air with rating.

In one case, for instance, where 3000-hp. boilers equipped with air-cooled furnace walls are fired with pulverized coal using the bin system with vertical down-shot burners, it is possible, as shown in Fig. 18, to operate with 25 per cent excess air up to 100 per cent of boiler rating. From this point the excess air is gradually increased with rating until it reaches 69 per cent at 350 per cent of rating. The limiting factor at the lower part of the curve is smoke, whereas refractory deterioriation is the criterion above 100 per cent of rating.

In another installation comprising pulverized-coal-fired boilers of approximately 2700 hp. each and using water-cooled furnaces consisting of bare water tubes set in recesses in the refractory

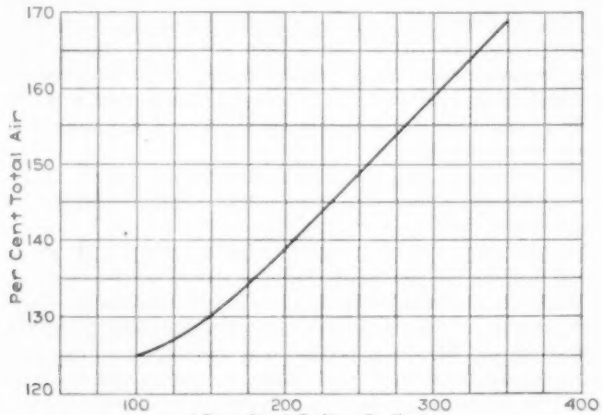


FIG. 18

wall, the boiler meters are adjusted to give 30 per cent excess air at 100 per cent of rating, 14 per cent excess air at 200 per cent of rating, and 33 per cent excess air at 350 per cent of rating, as shown in Fig. 19. In this case 30 per cent excess air is required at 100 per cent of boiler rating to prevent smoke. As the rating increases and a higher furnace temperature is obtained, the excess air may be gradually decreased to a minimum of 14 per cent at 200 per cent of rating. As the boiler rating is increased above 200 per cent the increased furnace temperature causes the refractories to start to run unless the excess air is again gradually increased with increased boiler rating.

In the last two installations mentioned, as well as in the case of a dozen or fifteen other installations in the larger central stations where a comparatively large variation in excess air with rating has been found advisable, we have designed special air-flow displacers, as was done by the author for the boiler meters at the Hudson Avenue Station.

Where only a small variation in excess air is required and where the excess-air curve is approximately a straight line, such as a gradual variation of, say, from 20 per cent excess air at 140 per cent of boiler rating to 30 per cent excess air at 240 per cent of rating, the adjustment may be made with the standard parabolic displacer by simply changing the radius R, referred to in the paper, the amount of mercury in the air-flow displacer reservoir and the positions of the balance weights.

In addition to the desirability of adjusting the boiler meter for a variable excess air with rating in order to insure best operating efficiency at all times, as outlined, it may be necessary to adjust the air-flow mechanism for something other than the standard-gram-weight calibration curve in order to accurately compensate for a number of variables in boiler operation.

There are certain factors, for instance, such as an increased flue-gas temperature with rating, an increased superheated steam temperature with rating, an increased steam pressure at the meter orifice with rating, and a decreased boiler efficiency with

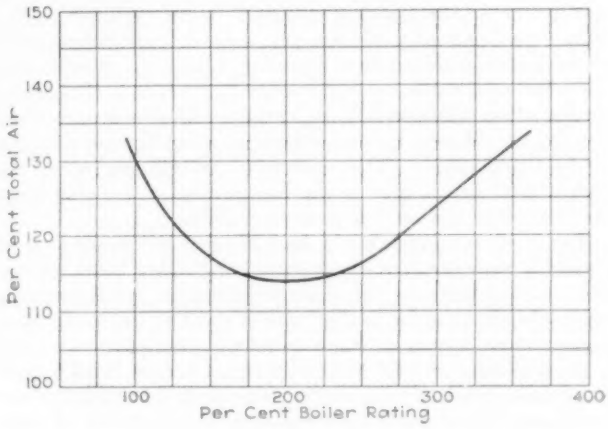


FIG. 19

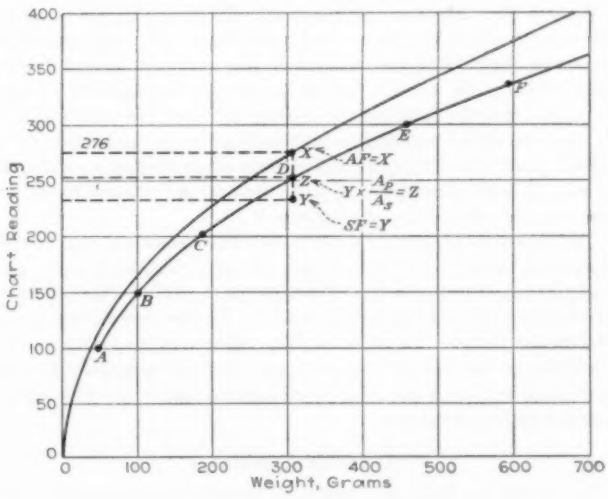


FIG. 20

rating which must be taken into consideration in adjusting the steam-flow—air-flow relationship if the excess air is to be held within very close limits at all boiler ratings. This is accomplished by making the air-flow adjustment on the basis of meter readings and complete gas analyses taken at several boiler ratings. The boiler meters in the larger power plants where it is desired to hold the excess air within 2 or 3 per cent of that corresponding to most efficient conditions should be adjusted in this manner.

The usual procedure in making an air-flow adjustment of this nature is, first, to run a dead-weight or gram-weight calibration of the air-flow recorder. A curve, such as the upper curve in Fig. 20, should then be plotted with gram weights as the abscissas and the corresponding chart readings of the air flow as the ordinates. After the dead-weight calibration has been obtained, a number of complete-combustion tests should be run on the

boiler at different rates of steaming. After having properly analyzed the results and having determined the proper excess air at each of the rates of steaming, horizontal lines are drawn on the dead-weight calibration curve at the average readings of the air-flow during the various tests. At all points where these lines cross the original dead-weight calibration curve, vertical lines such as XY in Fig. 20, are drawn, and on these lines new points such as Z, for the final air-flow calibration curve, are placed. Each of these points is determined by multiplying the average corrected steam-flow reading Y during each test by the ratio of the observed total air during the test to the desired total air for that particular rate of steaming A_p/A_s. In other words, the point on the new curve Z equals $Y \times A_p/A_s$. The curve through these points determined by the results of several combustion tests at different ratings is the new calibration curve for the air-flow mechanism. When calibrated to this new curve the desired total air will exist at the various rates of steaming when the steam-flow and air-flow pens are together.

Too much emphasis cannot be laid upon the importance of obtaining representative gas samples in adjusting boiler meters. The gas-sampling tube should be shifted about in all directions across the gas passages in the boiler in order to locate the most representative spot from which to take gas samples.

A certain amount of stratification seems to exist in all boilers. It appears practically impossible to properly mix burnable elements and the oxygen in the air in equal proportions in all parts of the furnace, and for this reason gas samples taken from different parts of the exit-gas passage will show different percentages of excess air. More excess air is usually shown in gas samples taken near the sidewalls of the boilers than from samples taken near the center, and the amount of this variation should be determined. It is equally important to take a number of gas samples from the front to the rear of the last pass of the boiler in order to detect stratification from front to back, if any. It is also important to determine the amount of air infiltration through the boiler setting before making an air-flow adjustment based on gas samples taken from the boiler exit.

A large pulverized-coal-fired boiler was recently found to be operating on 10 per cent excess air as determined from gas samples taken from the top of the first pass of the boiler, whereas the boiler meter had been adjusted for 25 per cent excess air at the boiler outlet. In other words, there was considerable infiltration of air through the setting so that gas samples taken at the boiler exit were far from being representative of combustion conditions in the furnace.

Author's Closure

It should be kept in mind that different installations have different requirements of excess air versus rating, and this paper does not pretend that the conditions desired at the Hudson Avenue Station should necessarily prevail elsewhere. However, the theories presented are general, and the fundamental principles outlined are applicable to other installations.

In connection with Mr. A. E. Livingston's difficulties regarding the use of the "capacity factor method," this method was discontinued at the Hudson Avenue Station, and the factor of 5 per cent excess air per 0.1 in. change in radius is used, for simplification only. The capacity-factor method presupposes that the meter has a particular characteristic. It may be suggested that herein lies Mr. Livingston's apparent difficulty.

In answer to Mr. Livingston's first question, it is admitted that the results were found to be rather confusing at the outset of the work. This confusion was the condition which necessitated the development of a new or different procedure and resulted in the "deadweight check" calculations. It is felt, however, that rust and sediment within the air bells on certain meters, which would result in a varying cross-sectional area of the bell at the surface of the oil as it is withdrawn, were large contributing factors in causing some of these confusing indications.

The square law is applied to the weight of gas and not to the volume. The effect of the change in specific gravity of the gases will be amplified under the reply to Mr. Hammond.

The use of an Orsat check at only one rating does not increase the possibility of error over a procedure requiring a series of Orsat checks. This is due to the fact that the tabulated records present a continuous picture of conditions and make each individual check practically a complete story when compared with the per cent total air obtained on previous checks. With increase in length of service and no change in the radius of the boiler meter, the per cent total air should decrease provided the baffling and setting have not deteriorated. As explained in the paper, it is this complete record of all Orsat checks which permits deductions concerning baffle conditions. It is true that possibly one check in every 35 will give an unreasonable Orsat analysis when compared with past indications. An additional check will invariably agree with the past record or substantiate an abnormal condition. Changes to the radius length are even forecast, and the Orsat checks then are merely formalities. This, however, would not be the case were it not for the complete record of all data pertaining to each air-flow mechanism, in which event procedure on the basis of an Orsat check at only one rating would be impossible.

Mr. Livingston suggests a short-cut method, but it is not evident wherein making two Orsat checks is simpler and shorter than making one.

In connection with the question raised by Mr. V. M. Frost, actual evaporation and boiler efficiency should not be confused. It is granted that the boiler efficiency is not a constant. However, neither is the feedwater temperature which both Mr. Livingston and Mr. Frost pointed out. As the rating increases and the boiler efficiency drops, more coal per pound of steam would be required, assuming the feedwater temperature to be constant. However, as our rating increases, our feedwater temperature increases, which of itself would require less coal per pound of steam. Thus, the feedwater temperature change obviously has a counteracting effect on the efficiency. Mr. Frost also mentioned weighed-water test conditions at which time the feedwater temperature is extremely cold in comparison with normal conditions, and the steam-flow—air-flow relation is affected in consequence. Abnormal conditions existing but occasionally on individual boilers should not be considered. Under normal operating conditions, feedwater temperature changes do not cause any difficulty. As a matter of fact, they are desirable to counteract the change in efficiency. Additional variables affecting the accuracy of the assumptions will be discussed under the reply to Mr. Hammond.

Mr. Markson questions the equation $y = a^1x^2 + c_1$, apparently overlooking the fact that the Orsat analyses check the predicted curves based on this equation. In connection with c_1 being the stack effect, this would be true provided stack effect were not compensated for by proper location of the draft taps, as is the case of the installations at the Hudson Avenue Station.

Mr. Worth expressed a fear that the boiler meter seems to present too complicated a problem for the small industrial plant. The situation is not quite as bad as it might at first appear. It must be remembered that the data presented in the paper were founded on the operation of large boilers operating at high ratings. The resulting high furnace temperatures cause comparatively rapid slagging of the boiler tubes. The action of the meter is such that, with increase in boiler service, the per cent total air will decrease without adjustments to the meter. In the industrial plant these conditions are not as severe, the ratings and tempera-

tures being relatively low. Moreover, it is not usually as essential that the absolute maximum of efficiency be obtained as is the case in the central station, since the coal consumption is not nearly so great. Consequently, if the industrial plant starts out with the type of characteristic desired, but the radius slightly longer than it should be, the natural slagging action of the boiler will gradually lower the curve in position without changing its shape. Before very long the characteristic will be in exactly the desired location. The boiler will consequently operate continuously at a fair average, and for a portion of the time the meter will be absolutely correct.

Mr. Hammond mentioned increase in flue-gas temperature, increasing steam temperature and decreasing boiler efficiency as variables affecting the steam-flow—air-flow relation. It may be well at this time to add a few more for completeness; namely, change in stack effect, change in feedwater temperature, the fact that although the meter is to measure air flow the actuating medium is flue gas and that increase in flue-gas temperature affects not only the volume but also the density and viscosity. The last two influence the friction factor coefficient. Moreover, the change in steam temperature affects not only the f.o.e., but also the steam-pen reading. It is next to impossible to evaluate and sum up these variable influences. Moreover, some are admittedly positive and some negative, and consequently tend to compensate one another. An attempt was made to evaluate these variables, resulting in a net computed error of from about +3 per cent to —4 per cent, depending on the variation in the friction coefficient. Under certain conditions the net error was calculated to be zero. However, the Orsat is the cornerstone on which the entire structure must either stand or crumble, and when the Orsat was used to check the curves predicted by the "deadweight check," verification of the correctness of our procedure was definitely established. This is the conclusive proof that the variables actually do compensate and that their net effect must be zero within the limitations of the measurements.

In closing, it may be of interest to state that the methods outlined in this paper are being successfully applied to the meters in the Gold Street Station of the company, where the boilers, stokers, and meters are different from the Hudson Avenue Station installations.

In addition, any one desirous of inspecting the complete record of the meters and observing the operation of the boilers at Hudson Avenue is cordially invited to visit the station.

FSP-52-38

Power and Heat in the Industrial Plant

General Aspect of the Problems Involved—Power and Steam-Heat Relations—Size Effects—Investment and Operating Costs—Examples of Industrial Installations

By R. J. S. PIGOTT,[1] NEW YORK, N. Y.

THE study of this subject has in the past two or three years received an amount of attention much more commensurate with its importance than it did prior to that time. One reason is that the increase in size of industrial plants has increased the amount of money involved; another is that it is now realized that a dollar saved in producing power or heat is just as valuable as one saved by improving the process employed in its generation.

For most industrial plants the cost of power, and often of heat, has been a very small percentage of the total production cost, and consequently consideration of power cost has been subordinated to the larger savings obtainable in the production process. For example, the total cost of power, building heating, process heat, water, etc., in the manufacture of military rifles was $0.28 out of a total cost of $18—a matter of 1.5 per cent; but the total bill for these services was $60,000 per month.

Another very powerful factor opposing improvement in power and heat production is quite natural and human: the operators of the plant, experienced by years in their own production, feel that few can tell them much about it—and this idea is automatically transferred to all the factors connected with production, which includes power and heat. The result is that the operators of an industrial plant have in general resisted, often strenuously, any attempts on the part of power and heat specialists to offer advice or help for the solution of their power and heat-supply problem. This resistance is gradually—very gradually—breaking down under the lead of the very large industrials (steel mills, paper mills, chemical plants, and the like) who have found their power problems big enough to force the employment of specialists.

Still another factor affecting proper consideration of the subject is the accounting situation. It seems that many industrial accounting departments have lost sight of the fact that cost accounting is a tool for the use of production management, and consequently must be planned so that it will serve an engineering purpose, not degenerating into a string of figures a month late, precise to the penny, but inaccurate.

One of the outstanding difficulties in this connection is the almost general practice of lumping the investment in power and heat-supply equipment with the main plant investment, so that power and heat costs are only reported as operating and maintenance charges. As a result, when any new power project is up for consideration it must carry both fixed and operating charges in comparison with production charges only on the old equipment, and naturally the new project is often defeated if it is for savings alone. Only when more capacity must be supplied does the new plant get any of the consideration.

Power and Steam-Heat Relations

It is obvious that heat for process produced by the direct firing of any fuel has no effect upon the problem we are considering; but heat supplied as steam at any pressure has an enormous effect upon both power and heat cost. It is hoped, however, that by this time the older conception has died a natural death: namely, the assumption that because steam was necessary, power was obtained for nothing, or conversely, because power was needed, steam cost nothing. Actually, the cost of power is greatly reduced and the steam cost materially reduced, and both costs are easily and rigorously calculable.

It should be clear that the fuel charge for a turbine whose whole exhaust and bleeder steam goes to process should be based upon the relation:

(Heat equivalent of gross kw-hr. output) + (Heat equivalent of generator losses) + (Heat equivalent of mechanical losses) + Radiation

Based on 94.3 per cent full-load generator efficiency and 97.1 per cent full-load mechanical efficiency. The electrical losses for a 1500-kw. bleeder turbine will be 86 kw. and the mechanical losses 43 kw., both of them being substantially constant and unaffected by load. Therefore the heat equivalent of a kilowatt-hour is given by the expression $3413 \times [(\text{kw.} + 129)/\text{kw.}]$, which gives for a 500-kw. load 4290 B.t.u. and for a 1500-kw. load, 3700 B.t.u.

Similarly, for a 20,000-kw. machine having 96.8 per cent generator efficiency and 98.8 per cent mechanical efficiency the values are: generator loss, 640 kw.; mechanical loss, 240 kw.; heat equivalent of 1 kw-hr. at 1000 kw., 6410 B.t.u.; at 20,000 kw., 3560 B.t.u.

Total heat abstracted from steam for power, $H = 430{,}000 + 3413 \times (\text{kw.})$ for the 1500-kw. turbine, and $H = 2{,}940{,}000 + 3413 \times (\text{kw.})$ for the 20,000-kw. machine.

Since all heat not removed as work appears in the exhaust or bleeder steam, we are not primarily concerned with the engine efficiency if the bleeder requirements are larger than needed for furnishing the required power from the pressure drop available; but if the power requirements are large relatively to steam for process heat, both the engine efficiency and the initial steam conditions become primary factors.

The curves given in Dr. Kleinschmidt's paper of December, 1928, on "Balancing Heat and Power in Industrial Plants,"[2] are precise enough for all preliminary estimates. Figs. 1 and 2 reproduced from that paper show the available power developed between various initial and final pressures. There is a difficulty always present, however—namely, that of variation in balance, both of the actual from the estimated power and steam loads, and from the variations in output from the state of business affecting the whole load. It must be recognized that very few industrials are in a position definitely to assign steam quantities to each department, since this usually requires metering to an extent quite beyond the usual equipment; and further the steam meters as actually installed and used in many industrial installations are seldom closer than 10 in accuracy. Power distribution is easier to get, but there are still many plants in which the metering is very scanty.

Size Effects

Plainly the size factor affects the cost of power, and while the comparisons about to be considered are in general not for straight condensing plants, we are under the necessity of getting most of

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Presented at the Akron Meeting, Akron, Ohio, October 21 to 23, 1929, of The American Society of Mechanical Engineers.

[2] Trans. A.S.M.E., May-August, 1929, paper No. FSP-51-27.

the data from such sources since they are the only ones on a reasonably comparable basis. Even in these, however, the costs vary over a wide range. Considering only plants built from 1920 to 1928, and the majority from 1923 to 1926, an analysis of cost data from forty plants varying from 1000 kw. to 300,000 kw. in capacity shows some very interesting facts. Values for B.t.u. per net kw-hr. are the most consistent, and as shown in Fig. 3 vary from +18 per cent to —9 per cent from the average. Reducing the points from the actual capacity factors to 38 per cent capacity factor usually brings the points much nearer the average. Fuel cost can be easily plotted from this curve, the plot for coal costing $0.18 per million B.t.u. being given, and curves for coal at any other prices simply lines parallel to the B.t.u. line.

Fig. 4 gives the average operating and maintenance costs. For the sake of clearness, the points are omitted, but in the original plot, operating labor varied +100 per cent to —30 per cent from the average, fuel cost from +100 per cent to —50 per cent, maintenance +85 per cent to —100 per cent, and supplies from +250 to —55 per cent, while the total cost above fuel varied from +120 per cent to —55 per cent. These very large variations are not surprising when it is realized that the capacity factor varied from 17 per cent to 80 per cent, the average price of labor from $0.50 to $0.90 per hour, and the design of plant to such an extent that the number of operators in certain plants was twice that of others of the same size. All these variables except design of plant could be corrected by reducing costs to a standard capacity factor, and adopting a standard price for labor, but this is a task of some magnitude, and as the plots on logarithmic paper show the trend and averages much more clearly than cross-section paper, it was not thought worth while to make the corrections.

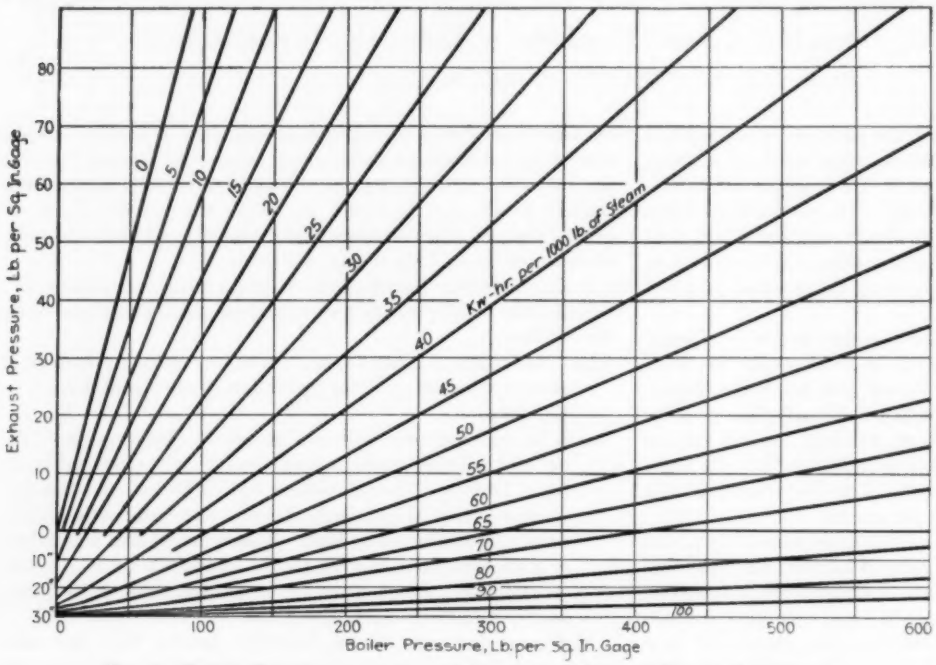


FIG. 1 POWER DEVELOPED BY IDEAL ENGINE FOR VARIOUS STEAM CONDITIONS

It should be remembered that straight-line plots on logarithmic paper represent curves on cross-section paper. One of the advantages of the logarithmic plot is that variations of a definite percentage either side of a line are lines parallel to the original lines whereas the same percentage variation on cross-section paper shows an increasing divergence as the numbers increase.

These curves may be used with safety for any good new design, as the costs are practically all on the low side for the newest plants, and tend to decrease year by year in new designs. It is noticeable that after capacity large

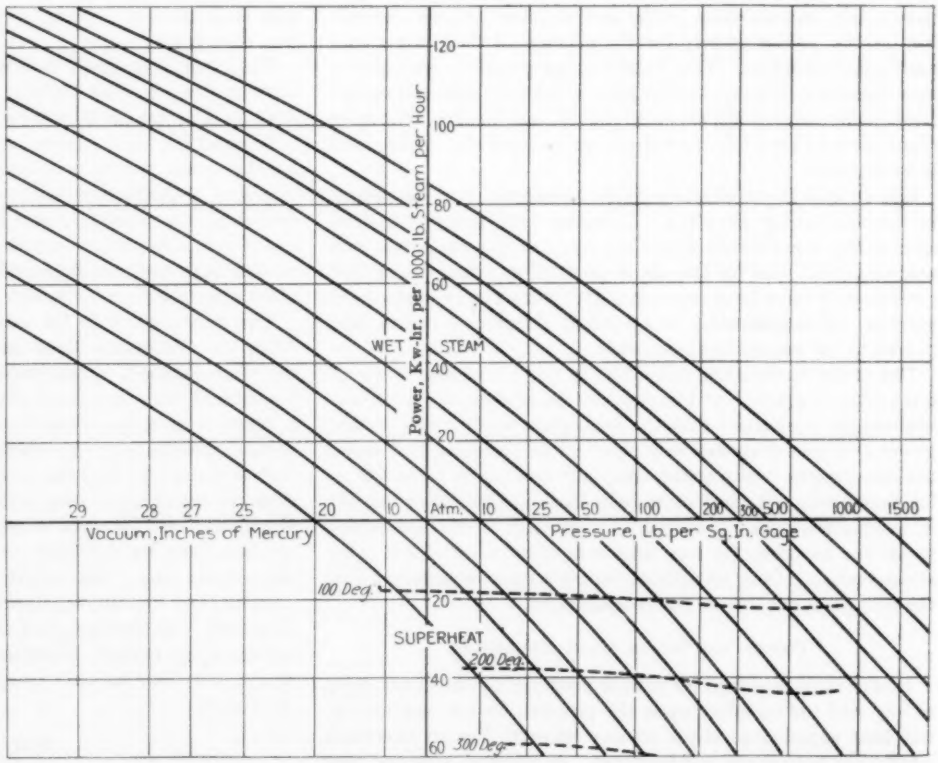


FIG. 2 POWER DEVELOPED FROM SUPERHEATED STEAM BY IDEAL ENGINE

enough to employ 30,000-kw. units is reached, there is little further decrease in B.t.u. rate, and not much more decrease in operating cost other than fuel; maintenance, in particular, is practically unaffected beyond an output of 150,000,000 kw-hr. per year.

While industrial plants (with the exception of steel mills) are not ordinarily interested per se in so large a plant as this, nevertheless these figures are needed to help determine whether to build a new plant or buy power, since the question is usually one of whether to replace an inadequate, uneconomical industrial plant with a new and, for its size, economical one, or to buy power from a public utility. Naturally, as the public utilities grow at a rate which puts new units or new plants on their lines every year or two, the comparisons must be made with economical public-utility plants. It is worth noting that utility electric power is the only commodity which has steadily decreased in price during a period when everything else has risen.

Consideration of Fig. 5 shows that the efficiency of the turbine, exclusive of external losses such as those due to bearings, governor, oil-pump drives, generator losses, etc., decrease very slowly down to the 6000-kw. size, and only a little more rapidly below that size. It will be seen that the engine efficiency

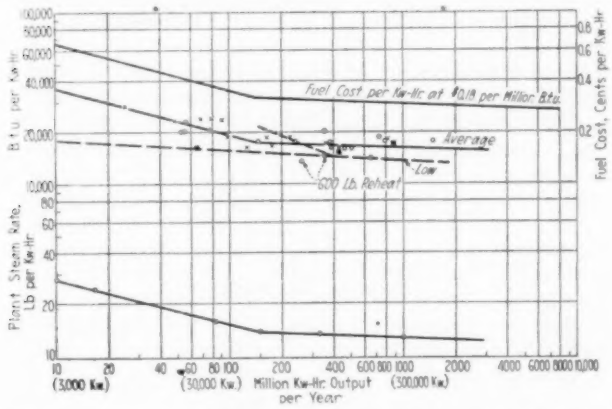


FIG. 3 B.T.U. PER NET KILOWATT-HOUR (FUEL COST AT $0.18 PER 1,000,000 B.T.U.)

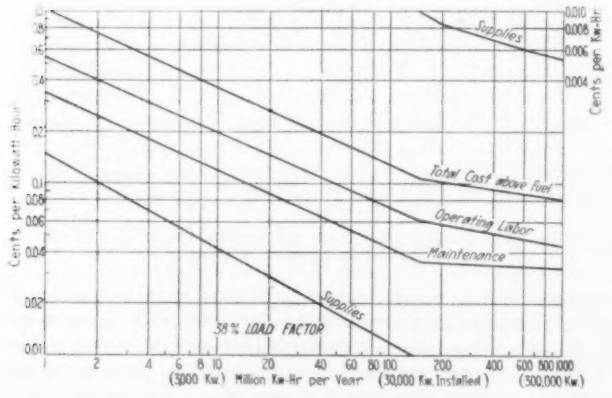


FIG. 4 COST OF POWER (PLANTS BUILT 1921–1928; COSTS, 1924–1928)

only falls from 80 per cent at 30,000-kw. capacity to 78 per cent at 6000 kw. and 48 per cent at 100 kw. The practice of thinking of turbine efficiencies in terms of brake output or, still worse, electrical output is undesirable as well as incorrect thermodynamically. Figuring on these bases gives erroneous steam flows and conditions. We are compelled to treat the external losses as part of the turbine output, so far as the steam is concerned. A single example will illustrate the force of this point. A 500-kw. back-pressure turbine has full-load mechanical and electrical losses amounting to 100 kw. For 400 lb. abs. (700 deg. fahr.) initial pressure and 160 lb. abs. back-pressure, the full-load steam would amount to 36,300 lb. per hour, the exhaust heat content would be 1296 B.t.u. per lb., and the efficiency 62 per cent. If figured on electrical output direct the exhaust quality would apparently be 1305 B.t.u., and the efficiency 51.7 per cent, the ex-

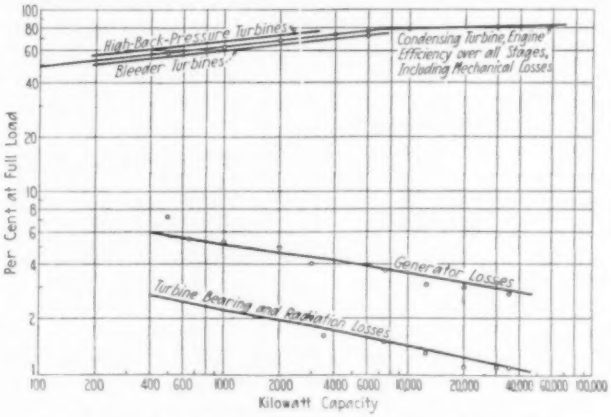


FIG. 5 TURBINE EFFICIENCIES AT VARIOUS OUTPUTS

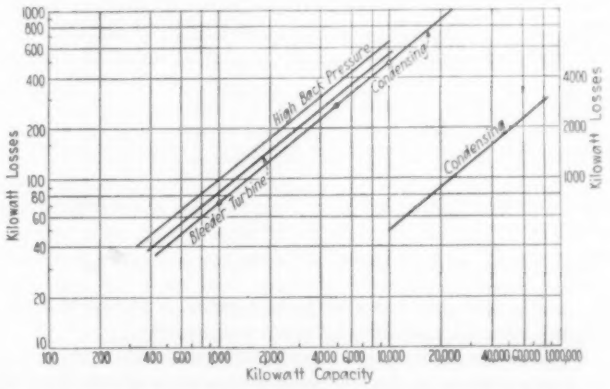


FIG. 6 TOTAL MECHANICAL AND ELECTRICAL LOSSES FOR VARIOUS TYPES OF TURBINE GENERATORS

haust heat in this case not being so far out. But at 100 kw. load the total steam required is 14,170 lb., efficiency 48 per cent, and exhaust heat content 1308 B.t.u. per lb. If figured on electrical output the exhaust quality would apparently be 1329 B.t.u. and the efficiency 25.5 per cent.

A further reason for using the method indicated is that it furnishes a reliable means of figuring the steam rates or total steam from no load to full load. The method is brief and convenient and is worked as follows: From the steam conditions selected, the adiabatic drop is taken from the Mollier diagram, the full-load wheel efficiency taken from Fig. 5, and the "indicated" steam rate then figured. Multiplying by the kilowatt load plus the kilowatt total losses from Fig. 6 gives the full-load steam. For no-load steam we merely take advantage of the fact that the no-load wheel efficiency varies only from 65 per cent of the full-load efficiency (Fig. 5) in a 500-kw. turbine to 70 per cent for a 30,000-kw. machine.

The no-load "indicated" steam rate is therefore simply the full-load indicated steam rate divided by 0.65 to 0.70, depending on the size. Multiplying this rate by the losses (which are not materially less at no load) gives the no-load total steam. The re-

mainder of the line is straight between these two points for all bleeder turbines, and up to the opening of the secondary valve for straight condensing machines. Bleeder or back-pressure machines are generally built without secondary valves, as bypassing stages is usually not feasible on account of interfering with the bleeder points.

The engine efficiency of back-pressure turbines is higher than that of straight condensing turbines not only because the wheels are much larger but because the work is all done in the high-pressure and superheat region and the stages are more efficient. But the turbine mechanical losses are higher, as the machine has

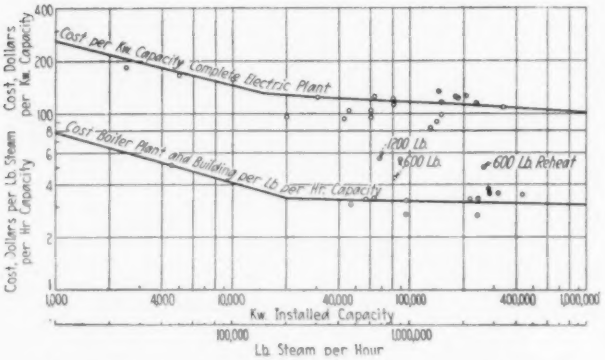


FIG. 7 INVESTMENT COST, STEAM AND ELECTRIC PLANTS

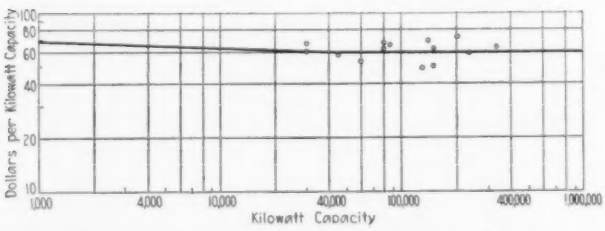


FIG. 8 INVESTMENT COST, ELECTRIC PLANT INCLUDING BUILDING

the weight and dimensions of the high-pressure portion of a condensing machine averaging 2.5 times the capacity.

The bleeder turbines are lower in efficiency than normal condensing ones because the low-pressure stages are always designed for full flow to develop the output with no bleeding, but are nearly always much underloaded in actual operation due to bleeding. The efficiency of these stages is of course lowered considerably, and as the high-pressure end is increased in size the no-load curves are also increased an average of 1.5 times. Fig. 6 is convenient for taking off the losses.

INVESTMENT AND OPERATING COSTS

Plant investment costs vary not only with size but with design. It may be definitely assumed, however, that with present experience with different designs the costs may be kept on the low side. The largest variables are foundations and building structures; the former is to a great extent determined by the character of the ground, the latter partly by the character of the design and partly by the architectural features. For industrial purposes it may be taken for granted that all architectural cost effects will be very low—the building will simply be a casing for equipment; the same condition is also largely true for public-utility plants not located in large cities, such as New York, Chicago, and Philadelphia. Such location always forces the building cost up, and in general all labor costs, both for construction and operation.

The variation of cost per kilowatt for complete electric plants is given in the upper curve of Fig. 7. But in an industrial plant the boiler equipment is generally very much larger per kilowatt of electric capacity on account of the increased steam rate due to bleeding or back-pressure service, and in many cases because of additional direct steam load for manufacturing service. The investment and operating costs for straight electric plants, therefore, cannot be used directly for estimating results in an industrial plant, but they do at least provide the data from which may be obtained the necessary separate information on steam-generating and power-generating sections of the industrial plant.

The data from half the plants investigated was in shape to separate boiler-room and turbine-room costs. The first information needed, that of boiler-plant cost, including the necessary building, is plotted in the lower curve of Fig. 7 against steaming capacity installed. Similarly the cost of the electric equipment, including its proportion of building, is plotted in Fig. 8. The values in Fig. 8 must be used with caution, as the back-pressure

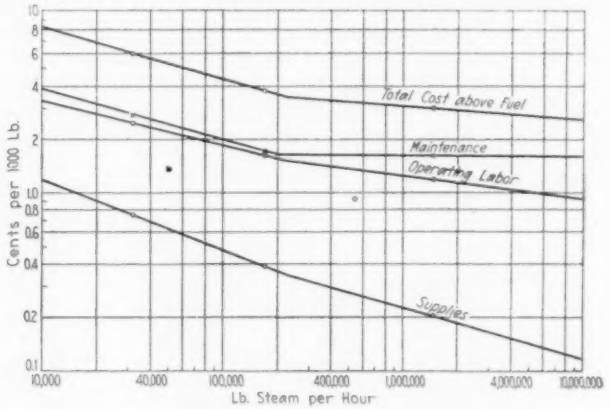


FIG. 9 COST OF STEAM PRODUCTION, INCLUDING BUILDING MAINTENANCE

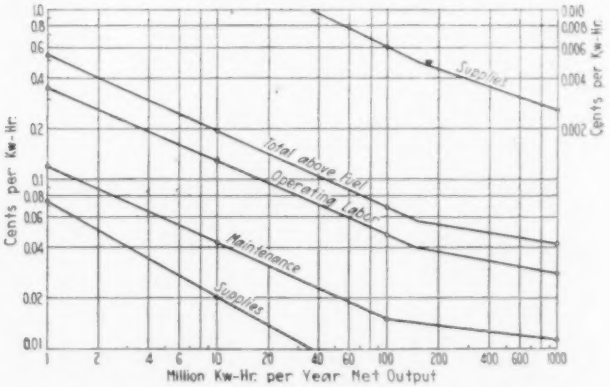


FIG. 10 PRODUCTION COST OF POWER ONLY (BUILDING MAINTENANCE INCLUDED)

turbines, and especially bleeder turbines of more than one point bleed, are more expensive and the costs will rise to some extent.

The operating costs have similarly been split into steam generation and power generation, each including a proportionate building maintenance (small in any case), and are plotted in Fig. 10. With the foregoing data we have the means of estimating with good accuracy almost any kind of industrial set-up. One other variable must be covered, namely, load factor. In all the curves given in Figs. 3, 4, 9, and 10 the costs given are for plants averaging 38 per cent capacity factor, the latter being defined as

$$\frac{\text{Net kw-hr. per year}}{\text{Installed capacity in kw.} \times 8760}$$

The variations of cost with capacity factor are rather hard to pin down, as the only published data are the studies of E. H. Scofield accompanying the A.E.R.A. Prime Movers' Committee Report of 1915, some of his more recently published figures, and some data on the Boston Edison plant. Calculations on a number of plant performances for design purposes corroborate these data and show that within reasonable accuracy the total operating costs in a given plant vary as shown by a line inclined at 20 to 22 deg. to the horizontal on log paper (see dotted line in Fig. 3), or in other words, total operating costs vary as the 0.76 power of the kilowatt-hour output. With this rule of variation it is easy to interpret costs for other load factors.

Industrial Installations

A few actual set-ups may be of interest. The first is a large foodstuff and beverage plant. When the problem of more capacity came up, the plant consisted of 26 boilers in sizes from 350 to 438 hp. in two boiler houses, two vertical-compound-engine generating plants, and one Diesel plant aggregating 2500 kw. capacity. The loads are rather steady all week, the only power shutdowns being on a few holidays, and partial load on Sundays. The steam demand reaches 200,000 lb. per hour at 140 lb. pressure, with average of from 90,000 to 120,000 lb., 20,000 to 40,000 lb. process steam per hour at 30 lb., and 40,000 to 60,000 lb. per hour at 5 lb. The electrical peak is 2500 kw., 1800 kw. being the average. The difficulty in choice of new equipment lay in the fact that large changes in the various kinds of product will likely take place, and considerable electrification of present steam drives is desirable (ice machines, pumps, and compressors). Consequently the tendency will be for electric load to go up at a much greater rate than steam demand, and there may even be a reduction in the latter.

The boilers vary from 20 to 40 years in age, so that complete replacement in any case is necessary. The firing equipment is all chain-grate natural-draft, burning a particularly difficult coal, running about 10,000 B.t.u. per lb. containing up to 16 per cent moisture, 20 per cent ash, and 3.5 per cent sulphur, and having an ash-fusion point of 2100 deg. fahr. The boiler efficiency has been about 60 to 62 per cent and could not be much improved in the present equipment.

The new plant will consist of three 10,760-sq. ft. 450-lb. (625 deg. fahr.) boilers with water walls and pulverized-coal firing, and two 2500-kw. bleeder turbines. Space is available for an additional boiler and turbine. The bleed point is 140 lb. gage and the back pressure 5 lb. gage. The demand for 140-lb. steam for old engine drives is so large that there is more than ample steam to supply the power still leaving from 20,000 to 50,000 lb. to be dumped into the 140-lb. mains through a reducing valve. The 30-lb. steam is supplied, as before, from the 140-lb. mains through a steam accumulator. The cost of steam with $2.90 coal will be approximately 26.46 cents per 1000 lb. and of power, 2.430 mills per kw-hr. The fixed charges for steam will be 7.54 cents per 1000 lb. and for power (including a proportion of the steam-plant fixed charges) 3.514 mills per kw-hr. The total costs are then 34.00 cents per 1000 lb. of steam, and 5.94 mills per kw-hr. for power. Obviously the situation is so favorable to industrial power generation that public-utility power could not compete except under a condition discussed below. A limited breakdown connection with the public utility is provided, but no regular power use. The reduction in cost over present operation in this industrial will pay a return of about 25 per cent on the investment, particularly as one of the old power-house buildings was remodeled for the new equipment at a lower cost than that of a new building.

Another plant manufacturing artificial silk was faced with a somewhat similar problem, but in this case the expected increase of production required steam and power in about four times the former quantities. The old plant consists of one small turbine generator and a turbine-driven refrigerating machine. The steam load is so large relative to power that even with as low a pressure as 250 lb. (550 deg. fahr.) there will be live steam to the extent of 30,000 lb. per hour reduced in winter, but an even balance in summer. The new plant would be made up of three 1500-kw. turbines bleeding a small amount at 140 lb. for heaters and evaporators (30 per cent make-up exhausting to the 15-lb. system), 15 lb. back pressure, and three 6500-sq. ft. boilers. The steam demand 24 hours a day 7 days a week for 7 months is 131,000 lb. per hour and 124,500 for 5 months, all the 15-lb. steam from the evaporators and turbines being used to heat 2500 g.p.m. of hot water and other mill process. The heat balance is arranged so that by shifting load between the heater condenser on the turbine exhaust and the mill heaters, no steam is wasted, but in winter some must be put through a reducing valve to the 15-lb. system. The costs are 28.95 cents per 1000 lb. low-pressure steam operation ($4.55, 13,000-B.t.u. coal) and 7.15 cents fixed charges, or a total of 36.10 cents; and power, 2.26 mills operation and 1.37 mills fixed charges, or a total of 3.63 mills.

Another scheme of operation has been proposed several times in the last few years and offers much promise, although so far it has only been put into practice in one plant; that is, for the public utility to build a 1200-lb. station at the industrial, serving steam at anything up to, say, 250 lb. gage and such power as the industrial requires, and taking off whatever of the remaining power it requires for its own service. This power has the advantage of being available at peak, as distinguished from "dump" power, and is therefore much more valuable to the power company. The principal restriction is one of size, as the scheme cannot be applied to small plants directly.

The idea may work out for a group of plants, provided the distances are not too great, since the fixed charges on transmission of power and steam may determine whether the scheme is profitable or not.

As a matter of interest, the case of the beverage plant has been refigured for comparison. A 1200-lb. plant with a 10,000-kw. bleeder turbine is predicated; of course, no such plant has been built yet, but there is no reason to suppose that a small 1200-lb. turbine cannot be built in the future.

The estimated results are as follows: Same steam and power supply to the industrial, investment cost $1,800,000. The total net power developed is 9700 kw., of which 1800 go to the industrial. Production cost, 1.340 mills per kw-hr.; fixed charges, 1.105 mills per kw-hr.; total, 2.445 mills per kw-hr. Steam costs, 25.62 cents per 1000 lb. production; fixed charges, 9.36 cents; total, 34.98 cents. It will be seen that the power cost is lower and the steam cost slightly higher. But in the case of the 450-lb. plant, the total cost per year for power and steam is $569,900, whereas the cost from the 1200-lb. plant is $503,880 per year. In addition the power company has 69,300,000 kw-hr. per year at 2.445 mills per kw-hr.

There are four ways in which the industrial can plan its power and steam supply:

- *a* Complete plant, with suitable reserve, no utility connection
- *b* Normal-load plant, no reserve, public utility for excess power and reserve
- *c* Public-utility high-pressure plant, supplying steam and power to the industrial and furnishing excess power to its own lines
- *d* Purchase all power, and operate a low-pressure industrial steam plant.

The first two schemes may be desirable for plants having relatively low power and high steam requirements, and the choice between them can only be made for each individual case. The

third scheme will likely be desirable for large plants, or a group of plants close together, aggregating a large demand. The fourth scheme will generally apply only to the small plant with low steam demand relative to power.

It must be clear that there is no general solution for the problem of heat and power supply for industrials, as the permutations are endless. Every case must be solved for itself. The variables are: price of coal and labor, relative power to steam, load factor, investment, and operating cost as affected by size, and growth rate. As has been attempted in this paper, the best that can be done is to collect data which will give the engineer a reasonably assured basis for his calculations.

Discussion

BURT A. WALTZ.[3] In the work of the last year on the Mahoning Valley Sanitary District's problem of new water supply for Youngstown and Niles a steam pumping station and boiler-house development called for studies similar to those outlined in this paper.

In studying the designs of various modern boiler installations it was noted that various considerations result in designs involving wide variations in ratings of boilers.

Briefly, there now are boilers capable of operating as high as 500 per cent of rating, in round numbers. The method of developing such capacities is of course no mystery, but the use of unit costs for plants operating at widely varying capacities seems to introduce a factor which the paper does not recognize. The writer believes a discussion of the effect of boiler ratings upon unit costs would be of value.

EDGAR J. KATES.[4] The paper, while up to date as regards the characteristics of steam-power equipment, fails to take account of the Diesel engine's place in the power scheme.

The closing pages of the paper state the four ways in which the industrial can plan its power and steam supply. The last case, that of the small plant with low steam demand relative to power, is sometimes much better served by Diesel power than by purchased current. Purchased power is generally not cheap in the small plant, whereas Diesel power is, because the excellent fuel economy of the Diesel engine is obtained even in the smallest sizes and because Diesel attendance costs are low. Consequently, Diesel power often costs less than purchased power in small plants, and Diesel-engine reliability is no longer questioned.

Furthermore, it is quite practicable to utilize at little expense much of the waste heat discarded by the Diesel engine in the jacket water and exhaust gases. Where this is done, the fuel consumed in the coordinating low-pressure steam plant is correspondingly reduced, and if the heat demand is small relative to power, the fuel-burning plant may be entirely dispensed with.

Careful studies of cases such as these will frequently show that the Diesel power plant will supply power and heat at a considerably lower cost than either steam-power equipment or public-utility power together with low-pressure boilers.

R. R. JONES.[5] The author makes a statement to the effect that few plants, with the exception of steel mills, would be interested in a plant as large as 30,000 kw. or producing 150,000,000 kw-hr. per year. From this the writer takes it that the power requirements of factories producing rubber goods are not as fully understood as they should be. There are in this locality at least three such factory power plants whose sustained peaks are in excess of 30,000 kva. and whose annual production approaches 200,000,000 kw-hr. a year. Indeed, there are but few industries in which the subject of power and heat are so important as in the rubber factory.

The industry is also a large user of steam. In fact, the steam load for process work is, assuming the power is generated in a modern condensing plant, rather greater than the steam required to generate the power. Using a concrete example, if a Ford or similar small tire requires for processing 95 lb. of steam, it also requires for mechanical fabrication about $6\frac{1}{2}$ kw-hr., or a ratio of 1 kw-hr. to 14.6 lb. of steam. While the steam load required for the process work has been slowly declining from year to year, the electric power required, either directly applied or indirectly in the form of compressed air, water, etc., has been increasing and probably will continue to do so for some time. That this increase represents a healthy condition may be shown by the examination of production sheets of ten years ago and of the present time. If the power consumed per man per month ten years ago and the production in pounds during the same time are compared, they bear a ratio of about 1 to 1, while at the present time the ratio is about 2 to 3. That is, if a man used 1 kw-hr. to produce 1 lb. of product ten years ago, he now uses 2 kw-hr. and produces three units of goods. This trend is observed, of course, not only in the rubber business, but in all lines and is one of the reasons why the subject of industrial power is justly receiving so much attention.

Using the figures previously mentioned, of the ratio of steam for process to steam for power, brings the interesting conclusion that, if the process-steam load and electric load were synchronized and some cycle of high pressure working could be devised that would give a steam rate per kilowatt-hour of 14.6 lb. and still leave the steam in proper condition for process work, the rubber industry could generate all its power as a by-product. This conclusion would seem to point to another, and that is that most industrial plants should be considering some high-pressure cycle as a means of reducing their power costs.

There is some confusion among non-technical men resulting from the published statements of the results obtained at various boiler pressures by the utility companies and the great lack of uniformity in the selection of boiler pressures by them, but the writer thinks this can easily be explained by calling to their attention the small increase in thermal efficiency with increase in pressure, operating condensing as against the large increase in kilowatt-hours per 1000 lb. of steam when working non-condensing, as the initial pressures and temperatures are increased. If there is anything that just fits the case, it is the high-pressure non-condensing combination in an industrial plant using large quantities of process steam. And it may be shown equally well that this applies to the smaller as well as the larger, having due regard to the fact that each case is a problem in itself.

The fact that the steam requirements and electric demands are not or may not be synchronous indicates that the full advantage of high back-pressure working may not be realized unless such plants operate in parallel with condensing units located in the same plant as the non-condensing units or that the industrial plants operate in parallel with a utility. This combination of a high-pressure industrial plant and a utility is theoretically the best combination for an industrial plant not already equipped with an efficient condensing plant, but practically it does not always work out that way, because the usual rate sheet of a utility is not developed with that end in view.

The fact that the industrial with its high-pressure plant can make a portion of its power cheaper than can the utility and that the utility can probably make the remainder cheaper than can the industrial indicates there is room for both, and with proper cooperation in the making of rates both can profit.

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[5] Chief Engineer, Firestone Tire and Rubber Company, Akron, Ohio. Mem. A.S.M.E.

In general, it will be found that all or any fraction of the load of a factory making rubber goods is a desirable load. Usually the load factor is extremely high, and the power factor is held as near unity as possible. The load is much more constant than a railway load, and this constancy can and possibly will be improved by the use of steam accumulators or other means.

WILLIAM F. RYAN.[6] The paper gives much useful information in a most usable form. The charges assumed for heat energy can be readily corrected to any basis of accounting, and it is no criticism of the paper as a whole to point out that method of cost accounting is not applicable in many industries.

The author charges the consumer of steam the same price for a heat unit, whether it is delivered at 700 deg. fahr. or at 100 deg. fahr. For many processes this system is fair enough for all practical purposes, but in others it is wholly unfair and would discourage the use of exhaust steam.

Even in such a simple process as heating a building a heat unit is worth more at high temperature than a heat unit at low temperature. In more complicated processes, such as multiple-effect evaporation, the value of steam is as much a function of saturation temperature as it is of heat content.

For a concrete example, consider a chemical plant generating steam at 150 lb. gage pressure and operating non-condensing turbines which exhaust at 10 lb. gage. The exhaust steam is used for the concentration of a salt. On the author's basis of cost accounting the consumer of exhaust steam would be charged about 85 per cent of live-steam value. Yet live steam can be used triple effect where exhaust can be used only double effect; in other words, it takes 50 per cent more exhaust steam than live to produce the same result. Obviously the supervisors of the concentrating operation would resist the use of exhaust steam at 85 per cent of live-steam cost when it has only 67 per cent of live-steam effectiveness.

It would be uneconomical for such a plant to generate power in condensing turbines and to supply live steam to the evaporators, but that result might eventuate if the consumer is not offered some incentive to use exhaust steam.

It is an almost general principle of cost accounting to base charges on cost of production. Why should exhaust steam be an exception? In the type of plant under discussion exhaust steam is as much a "product" of the power station as is electric power. What does it cost to produce exhaust steam?

In the chemical plant used as an illustration, the turbines take steam at 150 lb. gage pressure with moderate superheat and exhaust at 10 lb. gage. The water rate is about 35 lb. per kw-hr. If no exhaust steam were sold, but the heat in the steam were utilized to the fullest possible extent in power product (that is, by the use of condensing turbines), then the water rate would be about 15 lb. per kw-hr. It follows that the power plant purchases 20 lb. additional live steam in order that it may deliver 35 lb. of steam to the process. Exhaust steam can be sold without loss if the producer receives 20/35, or 57 per cent, of live-steam value.

There are other factors in the "cost of production" of exhaust steam, but the additional live steam required is the principal expense. In the writer's opinion, no attempt should be made to reduce the cost of exhaust steam by taking into account the lower capital cost, lower maintenance expense, and lower auxiliary-power consumption of a non-condensing plant as compared with a condensing one. The cost-accounting system should offer some incentive to the producer as well as to the consumer.

In a plant producing a single product, division of costs between electric power and exhaust steam may be largely academic, and any system, rational or arbitrary, may suffice. In a plant where one product consumes most of the electric power and another product consumes most of the exhaust steam, it is extremely important that the cost of both forms of energy should be accurately determined.

[6] Mechanical Engineer, Stone & Webster Engineering Corporation, Boston, Mass. Mem. A.S.M.E.

AUTHOR'S CLOSURE

Referring to the remarks by Mr. B. A. Waltz in which he states that the use of unit costs for plants operating at widely varying capacities introduces a factor which this paper does not recognize, perhaps Mr. Waltz has not recognized that the operation of boilers at high ratings does not necessarily change the unit investment costs to any material amount. Whenever a boiler is designed to be operated at very high ratings, which implies low unit costs per kilowatt or per pound of steam for boiler surface, the situation also implies that much more heat-recovery apparatus is included to keep the overall efficiency high. It is the old question of whether one prefers to buy heating service in boiler tubes, economizer tubes, or preheater plates. As a matter of fact, experience has shown that in plants designed for high boiler-room efficiency in which the boilers are operated at average ratings varying from 190 to 600 per cent, the overall cost of the boiler-room equipment does not vary more than 5 to 10 per cent. This justifies the use of unit costs in the manner employed in the paper.

Referring to Mr. E. J. Kates' remarks, it is quite true that the paper did not discuss these Diesels, but in the space permissible for this contribution, not even the steam-power situation could really be adequately covered. There is unquestionably a large field in which the Diesel will ultimately be used, especially if it is developed for high speeds, so that the bulk and investment costs are not so great as at present. One point is worth noting, however, and that is that in the recovery of heat from Diesel-engine exhaust, it must be done by heating equipment of the economizer or preheater type in which the heat-transfer coefficient is very low. This of course raises the cost of the equipment for recovering a given quantity of heat. Furthermore, the amount of heat available in the exhaust of the Diesel engine due to its high efficiency is much less than that available from a steam turbine of the same output. Consequently, in many industrial plants it would be necessary to add steam boilers for process steam in addition to the Diesels. This in general will defeat the Diesel installation. The author is quite in agreement with Mr. Kates, however, that much more serious attention should be given to the possibility of Diesel-engine power for those plants where the heat balance is suitable for it.

Mr. R. R. Jones takes some exception to the omission of rubber plants from the big power and steam users' group. Attention should be called to the fact that few plants with the exception of steel mills can be considered 30,000-kw. power plants. There are not two dozen rubber-production plants in the country that use 30,000 kw. in a single plant, and compared to the total power plants in the country, this is relatively few. A few other large users can also be included if it is worth while specifying them—namely, pulp and paper plants, large smelting plants for copper, nickel, etc., and large chemical industries.

With regard to Mr. Ryan's remarks on the basis of charging for low-pressure steam, the author believes that he covered himself by showing that accounting within the industrial plant is still a completely unstandardized matter. For many purposes the author would modify the suggested method, which, however, would fit the majority of cases rather well. In those cases where the temperature of the operation puts the value on high-pressure steam greater than on low-pressure steam, of course some such weighting of the costs as Mr. Ryan proposes must be developed.

FSP-52-39

Recent Instances of Embrittlement in Steam Boilers

By FREDERICK G. STRAUB,[1] URBANA, ILL.

The author discusses briefly the cause of the recent boiler explosion at Crossett, Arkansas, attributing the failure to embrittlement produced as a result of using soda-ash treatment on a water too low in sulphate content. Other instances of cracking of seams in steam boilers are also described. Emphasis is placed upon the regular inspection of leaky seams in steam boilers, particularly when the boiler water does not meet the A.S.M.E. recommendation.

AS A RESULT of the explosion of one of the boilers in a power plant at Crossett, Arkansas, during May, 1929, the question has been raised as to how much attention must be given to the examination of boilers in order to prevent explosions of this nature. Data have been gathered in regard to steam boilers which have been badly cracked, as well as from those which have actually exploded. These data have been gathered through the courtesy of the boiler manufacturers, the steam-boiler insurance and inspection companies, and various operating companies whose boilers have cracked.

All of the instances of cracking which will be discussed are typical instances of embrittlement cracking. The criterion of whether embrittlement cracking has occurred is the manner in which the cracks proceed through the steel. If the small incipient cracks are truly intercrystalline and the metal is good in all other respects, it is felt that the cracking may be classed as embrittlement. One must realize that an examination of a large crack where the plates have started to pull apart will reveal transcrystalline failure. Also, where cracks have not started the plate will have regular slip lines and will experience transcrystalline failure when subjected to stress. Embrittlement cracking is caused by chemical weakening of the grain boundaries, and where the weakening has not taken place the plate is normal. If the weakening progresses far enough, static failure (transcrystalline cracking) will naturally occur due to overloading of the weakened plate.

The boiler plant at Crossett, Arkansas, consisted of six 500-hp. boilers operating at 215 lb. pressure. The boilers were about six years old. During the first two years of operation the make-up water was taken from a pond and received no treatment. Because of dirty steam and some scale, a system of treatment was started using a deconcentrator in connection with soda ash, and this method of treatment had been in use about four years when the explosion occurred.

Boiler No. 6 exploded. The seam opened up the entire length of the drum and both heads blew out. Boiler No. 5 was knocked out of its setting and badly damaged. When the remaining boilers were subjected to hydrostatic tests, they were found to be badly cracked. The cracking took place from rivet hole to rivet hole in the riveted areas. Examination of the fine incipient cracks showed them to be typical embrittlement cracks.

The boiler water was analyzed daily for alkalinity, but no attention had been paid to the sulphate content, which was decidedly low.

In addition to the Crossett explosion, data are available in regard to embrittled boilers in the following locations: Virginia, Ohio, Michigan, Louisiana, New York, Colorado, and Minnesota. The main data in regard to these plants are as follows:

Virginia. Three 500-hp. water-tube boilers operating at 425 lb. pressure cracked after two years of operation. A crack opened up in one boiler for a length of over 7 ft. from rivet hole to rivet hole, as shown in Fig. 2. Rivet heads cracked off and



FIG. 1 POWER PLANT AFTER BOILER EXPLOSION

cracks were found in all three boilers. The small cracks were found to be typical embrittlement cracks.

The make-up water came from a well and was not treated. It contained a large amount of sodium carbonate and very little sulphate. The boiler water was analyzed daily for alkalinity, but no record was kept of the sulphate content.

Ohio. Four 722-hp. boilers operating in Dayton at 145 lb. pressure were found to be badly cracked after one year's operation. The cracks occurred in practically all the seams of the drums. The boilers were in a dangerous condition when the cracking was detected. Tube ends were also found to be badly cracked. The cracks were typical embrittlement cracks.

A well water was used for make-up water, and was zeolite-treated before being fed to the boilers. The sodium carbonate

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[1] University of Illinois.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 BOILER-FEEDWATER ANALYSES

(Results in parts per million)

Location of plant...	Crossett, Ark.	Va.	Ohio	Mich.	La.	N. Y.	Colorado	Minn.
Source of water...	Pond	Well	Well	Well	Well	Surface	Surface	Surface
Treatment given..	Soda ash	None	Zeolite	Zeolite	None	Hot process	Soda ash	Zeolite
Sodium chloride...	..	7	12	15	39	2	10	16
Sodium sulphate..	6	10	45	20	1	8	5	8
Sodium carbonate.	16	213	206	119	257	35	5	290
Calcium carbonate	45	8	4	4	2	5	8	4
Magnesium carbonate.....	20	7	0	2	2	0	7	3
Iron and aluminum oxide.....	..	3	3	..	4	1	..	2
Silica.....	..	25	..	..	12	26	..	2

TABLE 2 BOILER-WATER ANALYSES

(Results in parts per million)

Location of plant.....	Crossett, Ark.	Va.	Ohio	Michigan	La.	N. Y.	Colorado	Minn.
Steam pressure, lb. per sq. in.....	215	425	145	150	200	175	380	200
Sodium hydroxide....	368	565	2180	850	1608	510	350	900
Sodium carbonate....	170	155	660	510	869	120	110	257
Total alkalinity.....	680	1020	3200	1700	3750	834	600	1507
Sodium sulphate.....	225	50	710	320	5	357	600	340
Ratio of sodium sulphate to total alkalinity.....	0.3	0.05	0.2	0.2	0.0	0.4	1.0	0.2
A.S.M.E. recommended ratio.....	2.0	3.0	1.0	1.0	2.0	1.0	3.0	2.0

analyzed weekly. The results of these analyses gave the following information:

Ratio $\frac{Na_2SO_4}{\text{Total alkalinity}}$	Portion of the three years
Below 1	14 per cent
Between 1 and 2	32 per cent
Between 2 and 3	32 per cent
Above 3	22 per cent

The phosphate content was zero at times, and high at other times. The cracks were found to be intercrystalline and could be readily classed as embrittlement cracks.

Minnesota. One 500-hp. boiler operating at 200-lb. pressure had been in operation on zeolite-treated river water for about seven years. The insurance inspector notified the operators about the danger of embrittlement because of the high causticity in the boiler water, at least two years before the cracking occurred. The operators asked the water-treating company about the possibility of danger and were told that no trouble would result from the use of their treatment. Consequently no attention was paid to the sulphate content. The boiler developed serious cracking, and only timely examination by the cautious inspector saved an explosion. The cracking was found to be typical of embrittlement cracking.

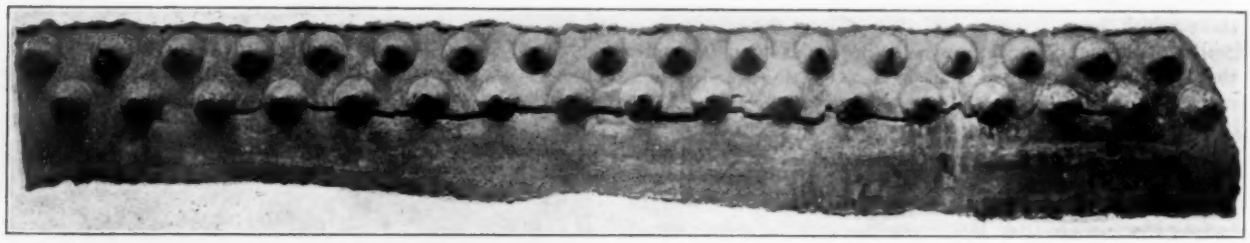


FIG. 2 EMBRITTLEMENT CRACK IN MAIN PLATE OF 425-LB. BOILER

content was high in respect to the sulphate content. A high caustic alkalinity was carried in the boiler water.

Michigan. Three 500-hp. boilers being operated in Pontiac at 150 lb. pressure were found to be badly embrittled. The boilers were five years old. For three years the water treatment used gave a high sulphate and low caustic content in the boiler water. The water treatment used during the last two years reversed this condition and gave a water which did not meet the A.S.M.E. Boiler Code recommendation. The cracks were found to be typical embrittlement cracks.

Louisiana. Rivets were found cracked off from the boilers after four years' operation at 200 lb. pressure. No straps were removed and no cracking could be detected in the plates by means of an external examination of the boilers. The rivets were found to have cracked in an intercrystalline manner.

The water in use was a well water practically free from sulphate and contained sodium carbonate.

New York. Two 500-hp. boilers were in operation at this plant. The steam pressure was 175 lb. One boiler had been in use one year on hard water before the second boiler was installed. At the time the newer boiler was installed, a hot-process lime softener was put in operation. The resulting boiler water was low in sulphate and high in causticity. After seven years' operation, sixteen rivet heads were found cracked off from the inside of one of the seams of the newer boiler. When the butt straps were removed they were found to be badly cracked. The cracking was typical of embrittlement cracking.

Colorado. Serious cracking was detected in a 1300-hp. boiler operating at 380 lb. pressure. The boiler was three years old. A fairly pure make-up water was used and soda ash treatment was given. At times sodium sulphate, phosphate, and tannin were used. A composite sample of four boilers had been

Other cases are on record, but these serve to show clearly that various types of boilers operating at different pressures are encountering embrittlement. The only factor common to all boilers is the occurrence of high causticity in the absence of sufficient sulphate content. This boiler water did not meet the A.S.M.E. recommended ratios for the prevention of embrittlement.

The data from the embrittled boilers cited show that the cracking took place in boilers made by practically every large boiler manufacturer. It also occurred in seams subjected to high temperature differentials as well as those entirely away from the heated areas. Boilers receiving excellent fabrication cracked just as rapidly as those not so well made.

Most of the cases were detected before it was too late as the results of timely inspection. A few had reached the dangerous stage, due to the fact that no attention was paid to the type of water in use. It was not realized that waters not embrittling in nature may be readily made so by certain methods of water treatment.

Emphasis should be given to the section of the A.S.M.E. Boiler Code which reads: "Pending further operating data from boilers in service, it is recommended that the requirements of Pars. 1–44 of Section VI of the Code be extended to all riveted seams, and that careful examination of all seams be made if leaks occur and do not remain tight after calking."

Discussion

H. A. ROBBINS.[1] The author is entitled to a vote of thanks

[1] Superintendent of Power, Brooklyn-Manhattan Transit Corp., Brooklyn, N. Y.

for keeping this matter before the public. In the earlier years there appeared to be a tendency to keep under cover cases of hydrogen embrittlement, as it was then spoken of, caustic embrittlement, embrittlement, or cracking. This possibly was due to the fact that those who had experience did not feel that they knew enough about the subject to start a general discussion.

In 1917 in one of our plants, equipped with 72 650-hp. boilers operated at 200 lb., trouble was experienced with leaking girth seams and rivets, which could not be made tight by calking. The boiler company was called in and a thorough inspection was made of every boiler in the plant. It was necessary to replace drums or sections of drums on approximately a dozen boilers.

The first idea of course was that the quality of the steel or the manufacturing methods were at fault. However, it was found later that the trouble could at least be avoided by the proper treatment of the feedwater. Water conditions were such that caustic soda and soda ash had been used for a number of years. Records showed that the NaOH had exceeded the Na_2SO_4. Following the discovery of the troubles we were advised to carry the Na_2SO_4, equal to or higher than the NaOH, which we have continued to do.

Although no repetition of the trouble has been experienced we are now endeavoring to bring the boiler water closer to the ratios recommended by the A.S.M.E. for prevention of embrittlement.

ALPHONSE A. ADLER.[3] The author assumes that intercrystalline failure of boiler plate is the criterion for caustic embrittlement. This may be so, but it is not conclusive. It is possible that the weakening at the grain boundaries may be due to other causes, and before general acceptance more proof should be obtained that nothing but caustic embrittlement could bring about this condition.

Any complete investigation should study not only the effect of boiler water used but also the previous history of the manufacture of the boiler plate, its fabrication into finished boilers, and the general design of riveted joints. Assuming that the steel is of exactly the same grade as was used before caustic embrittlement received attention, there are other factors that should be considered.

Soon after the meeting at which this paper was read the writer had the opportunity to inspect a piece of the seam which was reported to have come from one of the exploded boilers mentioned in the paper. The workmanship of the specimen was not up to standards required in good riveted-joint design. Some of the more important items to which attention is directed are:

1 A design in which the stress between rivet holes is the same in both boiler and cover plate to insure equal deformation under stress and insure uniform loading of rivets.

2 Adjustment of rivet-driving pressure to insure absence of initial stress on completion of riveting. (It is conceivable that rivet pressures may be so high that a seam may crack without bursting pressure in the boiler.)

3 A strictly uniform pitch of rivets to insure uniform deformation under stress between rivet holes.

4 Accuracy in driving rivet shanks normal to the plate surfaces to prevent tension components in the rivet.

In addition to the foregoing it is suggested that an investigation be made as to the effect of the injury due to the cold rolling on boiler plate especially in the heavier sizes.

JOHN HUNTER.[4] As a supplement to the investigation and conclusions on caustic embrittlement published by Professor Parr and F. G. Straub in Bulletin No. 177 of the University of Illinois, the paper on "Recent Instances of Embrittlement in Steam Boilers" is of great value in calling attention to the possibility of this condition arising, by citing definite instances where it has recently occurred. Those interested in the design, manufacture, and operation of steam boilers, especially with the wider use of higher steam pressures and temperatures, are fortunate in having at hand the results of such an important investigation and study of this subject, so vital in its relation to the safety of human life and property, relating to the cracking of boiler plates at the riveted joints, commonly known as embrittlement. By accepting and using the conclusions resulting from this investigation, published in Bulletin No. 177, serious accidents will be averted.

It has been the writer's privilege to investigate actual cases of the occurrence of this condition of embrittlement in boilers. A serious case was encountered in 1922 at one of the boiler plants in Dallas, Texas, where five boilers of the bent-tube type were condemned on account of cracks in the lower drum plates between the doubling plates, which had been removed for inspection. These cracks extended nearly the length of the drums and ran longitudinally between the rivet holes, with star cracks radiating therefrom similar to those shown in Mr. Straub's paper. In the same plant new drums were being installed in a longitudinal-drum boiler, replacing drums removed following cracking of the same nature. These boilers had been in service from four to eight years.

An investigation of feedwater in this plant, which was obtained from wells, indicated a high concentration of caustic. Although but little was known of the subject at that time, the conditions in this plant seemed to follow very closely those where similar failure had occurred in boilers of a scout cruiser in the U. S. Navy, and it was concluded that the cracking in this instance was due to caustic embrittlement resulting from the nature of the boiler water. Inquiry in other plants in Dallas, which were using city water in place of well water, showed that similar trouble had not been experienced.

In further investigation of this subject, the writer was called to Houston, Texas, in August, 1922, to examine a boiler that had developed steam leaks. The boiler was of the longitudinal-drum type of 1000-hp. capacity, and had been in service for ten months, burning oil. External examination showed the fusible plug out of place. The fusible metal of the plug was later found perfectly intact at the bottom and inside of the drum. The brass body of the plug had been completely eaten away by the action of the boiler water.

Further examination of the steam drums disclosed several small leaks between the rivets of the circumferential joints, which when thoroughly cleaned on the outside uncovered cracks between the rivet holes in these circumferential seams in all three drums, extending from the water line on one side to the water line on the other side. In other words, on the surface of the joints covered by water, these cracks were found only on the outside landing of the joints. The boiler water at this plant had the same pronounced soapy feel as the water at the Dallas plant, previously mentioned.

TABLE 3 SUBSTANCES AS DETERMINED

Substance	Symbol	Grains per U. S. standard gallon No. 1 Boiler water	No. 2 Well water
Silica	SiO_2	0.35	0.70
Iron oxide	Fe_2O_3		
Aluminum oxide	Al_2O_3	0.13	0.08
Calcium oxide	CaO	1.70	2.56
Magnesium oxide	MgO	0.32	1.04
Sodium oxide	Na_2O	38.04	8.74
Sulphuric anhydride	SO_3	13.44	6.96
Chlorine	Cl	25.10	2.18
Nitric anhydride	N_2O_5	trace	trace
Carbon dioxide fixed	CO_2	1.42	5.23
Carbon dioxide free	CO_2	...	0.38
Organic matter	...	0.70	0.50

[3] Consulting Engineer, New York, N. Y.
[4] Consulting Engineer, St. Louis, Mo.

The analyses in Table 3 represent two samples of the Houston water: (1) Sample taken from the boiler; (2) raw water from the old well which was used as make-up water in the evaporator or as direct feed for the boilers when the evaporator is by-passed.

Table 4 gives the combinations under which the substances in Table 3 react in the water.

TABLE 4 COMBINATIONS

	Grains per U. S. standard gallon	
	No. 1 Boiler water	No. 2 Well water
Silica	0.35	0.70
Iron and aluminum oxides	0.13	0.08
Calcium sulphate	0.82	...
Calcium carbonate	2.43	4.57
Magnesium sulphate	...	...
Magnesium carbonate	0.67	2.18
Magnesium chloride	...	...
Sodium sulphate	23.00	12.34
Sodium chloride	41.40	3.59
Sodium hydroxide	7.80	...
Sodium bicarbonate	...	3.97
Sodium nitrate	trace	trace
Organic matter	0.70	0.50
Total solution	77.30	27.93
Suspended matter	33.00	2.54

The well water contained sodium carbonate as a bicarbonate; also sodium hydroxide as a result of hydrolysis of sodium carbonate. Several sections of the boiler plate at the cracked joints were cut out and sent to the laboratory for analysis. These met the A.S.M.E. Boiler Code.

The workmanship on the joints was carefully examined and the rivet holes found true and well filled by the rivets. The plates were reasonably close together, and from general observation had not been abused in laying up the joints. The drums had been manufactured with the ordinary care. In ordering new drums for this boiler, directions were given the factory that all joints be calked on the inside to prevent any possibility of water getting between the plates, which under the existing water conditions seemed to be the cause of the plates cracking.

The study of feedwater treatment by those interested in this subject has developed means of taking care of the scale-forming matter, and at the same time avoiding the possibility of embrittlement, even with the higher steam pressures and temperatures. Their work should receive careful consideration and their recommendations should be followed closely. The requirement of inside calking on all drums or the use of seamless hammer-welded construction will furnish an additional safeguard. In the meantime the suggestion in the last paragraph of the paper, recommending careful examination of all seams indicating leaks for the possible existence of embrittlement cracks, should be followed.

J. C. McCabe.[5] It is assumed that engineers interested in steam boilers know that the external plate surface elongates and the inner surface shortens when rolled, about $1/2\ \pi \times$ plate thickness. It is evident that with plates of increasing thickness the structure of the steel plate is seriously disturbed if worked cold. In riveted joints of best design and workmanship rivets along the calked edge carry a load at the yield point in tension. These rivets have been strained beyond the yield point. The calking on plates of moderate thickness leaves the inside (dry side) of the plate in tension at the yield point and the outer surface in compression.

Observation of the rivet holes after the removal of rivets from a joint will give some conception of the enormous stresses set up in the best type of joint and the best workmanship. An examination of a riveted joint will disclose that the major stresses in the plate are not parallel to the calked edge. The major stresses or deformations are frequently about 45 deg. from the line of rivet centers. This condition can be best observed by constructing a flat joint.

Professor Straub has done a public service in calling attention to the hazard from a high concentration of sodium hydroxide. In Bulletin 155 on page 28, he says the following conditions must exist to induce embrittlement and cracking: First, "the actual stress must be above the region of the yield point of the metal; second, the concentration of the sodium hydroxide must be in excess of 350 grams per liter (350,000 parts per million, about 20,000 grains per gallon)." This would mean from 1600 to 1900 lb. of sodium hydroxide in a boiler of about 100 hp., if the concentration was uniform for the whole water content of the boiler.

A simple observation of the initial boiling point of this concentration in an open vessel and the rapid rise as the concentration increases will do much to impress men responsible for boiler operation. The recent practice of ascribing all cracking of boiler plate to embrittlement is far from the facts. In a calked riveted joint there is a condition with one side of the plate in tension and loaded to the yield point and the opposite side of the plate in compression at the yield point and usually the ligament between the rivet holes loaded to the yield point from the "barreling" of the rivet in driving. Why not attack the disease rather than the symptom?

Professor Straub lays much stress to the cracking around grain boundaries. He speaks of three boilers at Pontiac, Michigan, where cracked plates were found. The writer made an investigation of these cases and found the cracked butt straps exposed to the products of combustion, a temperature differential ranging from 60 to 80 deg. fahr., between the outer strap and the shell plates, and the rivet holes elongated from 0.010 in. to 0.016 in. girthwise. A micro photo showed that three out of four cracks were through the ferrite grain and the fourth partly around the grains and across the grains.

Observation indicates that in most cases of cracked plates the plate was loaded transversely as rolled. A plate so loaded is usually from 2000 to 10,000 lb. lower in tensile strength. This seems to be due to the elongation of the ferrite grains and throwing the grain boundaries more into parallel. The writer would expect to find frequent cracks around the grain boundaries when loaded transversely as rolled in any case.

About one year ago the lower drum of a Badenhausen boiler at the Ford Motor Company's Highland Park plant developed a crack of several inches on each side of a blowoff outlet which was also used for feeding water. A temperature difference of 150 deg. fahr. was found between the top and bottom of the drum. About $^2/_3$ the drum was in compression. The bottom $^1/_3$ was in tension and finally failed through the weakest section.

About the middle of November, 1929, a few heads were found broken off in the lower drum of a large boiler in the Rouge Plant of the Ford Motor Company in Dearborn, Michigan. This drum was 60 in. dia., the drum plates were $1^5/_{16}$ in. The heads were made of 1-in. plate. The heads were double riveted. A ligament of 70 per cent was left between rivet holes. A few cracks were found between rivet holes. This cracking was due to the stresses set up by temperature differentials in the shell plate. An area measuring $4^1/_2$ ft. girthwise and 23 ft. long was in direct contact with water at a temperature about 240 deg. fahr. while the balance of the drum was in contact with water at a temperature about 400 deg. fahr. The operating steam pressure was 250 lb. Two plates were used to form the drum. The butt-strapped joints in this case served as columns between the heads. The straps butted on both flanges. The load on the metal of the ligament at 250 lb. was about 5200 lb. This fact would not fit in the "caustic theory." The temperature difference was enough to strain up to the yield point with result-

[5] Chief Inspector, Department of Labor and Industry, State of Michigan, Detroit, Mich.

ant cracking at ligaments. The temperature differences will average about 150 deg.

It seems that our present knowledge of materials should enable us to design, construct, and use steam boilers without the misapprehensions of steam users of 75 years ago, who believed that explosions of boilers were due to an "unknown explosive gas" which formed in steam boilers and exploded without warning. It is now known that the old-time explosions were due to the poor slag-laden iron used.

It is about time to stop forming relatively thick plates cold and the construction of riveted joints strained, in part, to the yield point. These conditions are the cause of every so-called embrittlement crack. Why not recognize the value of fusion and forge welding under proper control as to fabrication and annealing? It is certain that the courts will give such recognition if the question is submitted regardless of codes or petty politics.

E. R. Fish.[6] There are few if any boiler manufacturers who have not been perplexed with serious cases of the cracking of plates in some portions of boilers of their manufacture. A good many will recall cases that happened many years ago for which no adequate explanation could be offered. The case is somewhat analogous to the physical troubles of the human body. Prior to certain developments of the medical science, people were afflicted with and died from ailments the nature of which was not fully understood, but which in the light of modern developments are readily diagnosed and treated. This has given rise to the statement by uninformed people that we are now subject to diseases that nobody had a few years ago. The diseases of course existed, but were not properly recognized.

While this theory of plate enbrittlement due to the presence of sodium hydroxide is to some extent controversial, the bulk of the evidence certainly justifies the theory that has been worked out. There is so much confirmatory evidence that most of us are convinced that it has merit, and until at least as strong evidence along other lines is presented, the acceptance of the embrittlement theory is fully justified. It must be recognized, however, that the factors at work are not fully understood and that there may be variations in the manifestations that are hard to reconcile.

The end that all are desirous of attaining is the adoption of means that will prevent this cracking which is a serious menace to the safety of high-pressure steam boilers. This means of prevention must, of course, be something that is reasonable and practicable, and this is the case with the means suggested in line with the embrittlement theory. Whether this theory is or is not ultimately disproved or still further strengthened, it behooves those responsible for the operation of boilers to follow the last paragraph in the paper, the epitome of which is, "Eternal Vigilance is the Price of Safety."

Alex D. Bailey.[7] So far in this discussion no one has had a kind word to say for the operating man who must live with these boilers which it appears are susceptible to so many baleful influences, and which may suddenly take a notion to rearrange the design of the power plant and the surrounding equipment as shown in the photograph in Professor Straub's paper. Of course the operating man, not being an expert, is apt to become confused and alarmed by the discussion of the chemists and the metallurgical experts who argue over his case and discuss the different ailments to which his equipment is liable to succumb. He is apt to think of the age-old definition of an expert and wonder if the tendency is not in that direction in this particular case. He has been running boilers for a long time and has never been blown up yet. He must remember, however, that steam pressures and temperatures are increasing, rates of heat absorption have increased by leaps and bounds, and with these changes, new problems have arisen in connection with the preparation and maintenance of proper feedwater and boiler-water conditions. Like other features of plant operation, the study of feedwater has become highly specialized.

The operating man may think he is in the position of the patient who, desirous of maintaining the spark of life and what little health he has left, calls in medical experts and diagnosticians for consultation. Each one of these is an expert on some particular disease to which the human body is heir, and it is only natural that each doctor should examine the patient for symptoms of the disease in which he is most interested and about which he knows the most. As a consequence there may be some discussion among the consultants and the patient will be scared stiff on being told that he may have any one or all of a number of deadly diseases. In conclusion, however, he will probably be told that by following a careful diet, being temperate, and avoiding over-exertion, he will live to a ripe old age, and if he is a wise patient, he will exercise all due precautions to preserve his health.

It seems that we in the operating field are in much the same position. It behooves us to listen attentively and to follow the advice of reputable professional men who have made this subject their life's work. Though they may disagree in detail, they are all working toward the same end, and in the final analysis they are all talking about the same thing, namely, the behavior of metals under stress, subjected to varying temperatures and pressures, and exposed to water containing various ingredients.

While it may be wondered why the experts apparently disagree regarding details, it must be admitted that discussions and differences of opinion are the basis of our progress, and that without them, we would rapidly get into a rut of complacent know-it-allness which would be deadly to advancement and hazardous to the safety of equipment and personnel. Discussions such as this are a healthy sign, and speaking for the operating man, the writer believes all should profit by them.

The fact that boilers, large and small, are continuing to fail, sometimes with considerable loss of property and even life, is sufficient warrant for each of us to consider his own case carefully, and to avail himself at every opportunity of the latest information pertinent to the subject. It is up to us to obtain the best professional advice, and to heed and follow carefully the admonitions and advice of those who have the latest and most up-to-date information on the subject.

R. E. Hall.[8] Without exception, the laboratories with which the writer is connected recommend that the concentration of alkalinity in the boiler water be maintained at the lowest concentration compatible with protection of the boiler surfaces proper. In a statement in the May 28 issue of *Power* we defined the limiting desirable concentrations of alkalinity in terms of hydroxyl radical. They may be restated here as pH values ranging from 10 on the low side to 11 on the high side with 11.5 as an extreme upper limit in cases in which high silica waters demand it for the maintenance of scale-free surfaces. It should be noted that the pH value of the boiler water making contact with the boiler should never be less than 9.6 measured on a cooled sample.

Maintenance of these low boiler-water alkalinities is fraught with effort. Several years ago before conception was had of

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[7] Superintendent, Generating Stations, Commonwealth Edison Co., Chicago, Ill.

[8] Director, Hall Laboratories, Inc., Pittsburgh, Pa.

any protective action of phosphate against embrittlement, we pointed out that through use of phosphate control of boiler water alkalinity became feasible, permitting maintenance of the recommended sodium-sulphate–total-alkalinity ratio in the boiler water at all operating pressures (Prime Movers Committee, N.E.L.A. Publication No. 256–71, p. 9, 1926), a thing impossible with soda ash as final conditioning chemical.

With the frequently changing fashions that are in these days being experienced in recommended sodium-sulphate–total-alkalinity ratios, any consistency of maintenance for those who attempt to keep pace with the prevailing mode is attainable only on the basis of extremely low and carefully controlled boiler-water alkalinity. In the ratio of sodium-sulphate–boiler-water-alkalinity a desired quotient may be obtained either by making the numerator (sodium sulphate) large or by making the denominator (boiler-water alkalinity) small. In our own work, we choose the latter alternative as inestimably the better practice since it represents greater safety to the boiler, better quality of steam, and greater economy of operation.

The present sodium-sulphate–alkalinity ratios are arbitrary. Agreement is pretty general that the efficacy of the sodium sulphate–alkalinity ratios in the boiler water in preventing deterioration of the boiler metal is due to the formation of a protective coating over the surface of the metal, especially in seams and elsewhere where concentration of the boiler water may occur. On this supposition, it becomes feasible to define exact ratios of sodium sulphate to alkalinity on the basis of the definite principles of solubility relations and concentrations involved at different operating pressures. Data are lacking or are uncertain at present above normal boiling temperature, but can be readily obtained. It would seem that there is no field open to investigation now which would give results of more value than that which would give definite information regarding the conditions in the boiler water necessary for the formation of films protecting the boiler metal from attack by the concentrated boiler water.

In Professor Straub's paper a number of very interesting cases of boiler-metal deterioration are added to those already described in his previous papers or those in conjunction with Professor Parr. As in the previous papers the effect of the specific chemical, caustic soda, is emphasized as fundamentally causative of deterioration.

The widespread interest in this subject among all those interested in boilers or their operation is attested by the numerous articles pertaining thereto that have appeared in the technical press during the year. Some of these agree that the cause of deterioration of the metal is chemical only, and especially due to caustic soda. Others, however, are looking to other factors as contributory, or primarily causative. It is notable in all these publications that attention is confined to boilers, although in many other fields there is accumulating a vast store of information relative to deterioration and cracking of metal. Cases of this character include the work of D. J. McAdam, Jr., of the Naval Experiment Station, Annapolis, Md.; H. S. Rawdon, Chief of Department of Metallurgy, Bureau of Standards, Washington, D. C.; T. S. Fuller, General Electric Company, Schnectady, N. Y.; and F. N. Speller, Chief Metallurgist, National Tube Company, Pittsburgh, Pa. In a paper to be published soon we shall discuss the bearing of the work of these various investigators on the question of boiler-metal deterioration.

Interest in and better acquaintance with deterioration of the boiler metal as evidenced by cracking, whether intercrystalline or transcrystalline, are calling attention to many examples of interest. Attention has been called to a case in which cracking developed in parts of a boiler, although the boiler water was practically neutral at all times, and the sodium-sulphate–alkalinity ratio therefore practically infinite.

Deterioration by cracking may occur therefore over a range of boiler-water alkalinities extending from zero as already noted to those high concentrations noted particularly in the Ohio and Louisiana cases cited in the paper under discussion. It is interesting in this connection to note that in the Ohio boilers, with total alkalinity of 3200, sodium sulphate of 710, and operating pressure of 145 lb., tremendous deterioration has developed in one year, while in the Louisiana boilers with total alkalinity of 3750, sodium sulphate of 5, and operating pressure of 200 lb., four years of operation had developed no deterioration detectable in the plates by external examination, except in some rivets.

One might expect, if alkalinity and sulphate are the sole or primary controlling factors in the deterioration, and if their effect is not subject to wide variation by reason of other contributing factors, that the Louisiana boilers would have shown by far the greater deterioration. The reverse is true, and it is extremely unfortunate that the analysis of the two cases has been limited to chemical considerations, when observations on the fiber stresses and various other characteristics of the boilers might easily have served to explain the anomaly.

The Crossett boilers are of particular interest inasmuch as exact data on the boiler waters of all six boilers are available over a period of years, and because of other significant features pertaining to them. Discussion thereof will be included in the report of a cooperative investigation of these boilers by the Naval Experiment Station, Annapolis, Md., and Hall Laboratories, Inc.

The following paragraph regarding the Crossett case occurs in Professor Straub's paper:

"Boiler No. 6 exploded. The seam opened up the entire length of the drum, and both heads blew out. Fig. 1 shows a portion of the plant after the explosion. Boiler No. 5 was knocked out of its setting and badly damaged. When the remaining boilers were subjected to hydrostatic tests, they were found to be badly cracked. The cracking took place from rivet hole to rivet hole in the riveted areas. Examination of the fine incipient cracks showed them to be typical embrittlement cracks."

The paragraph quoted is not in accordance with facts. The writer was present at the Crossett plant following the explosion and during the examination of the remaining boilers. The general manager of the Crossett plant states that no tests other than those of which the writer is cognizant had been made.

The facts in the case are that boiler No. 6 exploded. Disruption occurred along the seam of the lower drum of the two-drum vertical boiler and the heads blew out. The statements in the above quotation relating to character of cracking are presumably based on examination of samples from this boiler only.

Boiler No. 5 was displaced from its setting. As external examination of seams and rivets gave no evidence of cracking in this boiler, it was thought worth while to examine the seam on the lower drum, correspondingly exactly to the line of rupture in boiler No. 6. Accordingly the butt straps were removed for approximately three-fourths the length of the drum. Careful examination of butt straps, boiler plate, and rivets failed to reveal any characteristic cracking whatever. These observations with low-powered microscopes in the field will be checked, however, as a part of the cooperative investigation with the Naval Experiment Station mentioned.

Boiler No. 4, also displaced from its setting, was not examined or tested in any way.

Boiler No. 3, on hydrostatic test, blew out one head. The crack proceeded along the rivet line for some distance, then proceeded at both extremities by rupture of the head plate itself. Whether the boiler plate itself was cracked at all or not cannot

be stated at this time, but the condition of the metal will be reported on in the cooperative investigation.

Boilers No. 2 and 1 stood their hydrostatic test with no evidence of deterioration. Furthermore, three and four rivets in each boiler were removed along the seam (corresponding to that ruptured in No. 6) at points which by their appearance might attach suspicion to them. Examination of the cleaned exposed surfaces and of the rivets, as carefully as might be with a low-powered lens, gave no indication of cracking in these two boilers.

There is evidence, therefore, that two boilers of the six had cracked, and no more. We are justified in concluding that these boilers, all of the same type, have not all deteriorated equally, although subjected to like alkalinity conditions over a period of six years, and that other causative factors along with alkalinity have been determinative of failure.

Another consideration of interest applies to the Crossett case. Deconcentrators were used with these boilers starting in 1926. The chemical service obtaining with deconcentrators was for six months only. However, during the winter and spring 1928, the writer wrote twice to the Crossett plant urging the use of phosphate in conditioning the water, as its use would both lower the boiler-water alkalinity and also serve to give such protection from boiler cracking as lies in its power. No action was taken at the plant to carry these recommendations into effect.

The fact that success in water conditioning depends more upon exact chemical control than type of processing equipment led the Hagan Corporation about two years ago to abandon the deconcentrator, which was the focus around which our earlier investigations on water centered. The investigations originated for the deconcentrator provided the facts in the end that led to its discontinuance.

It is believed that the general phenomenon is the same whether cracking of the boiler metal occurs at zero or high alkalinity, and corrosion at stressed points, whether by caustic or other component in the concentrated boiler water, is one basic contributing factor. But as so aptly illustrated by the Crossett case, in which like operation and like boiler-water alkalinity in all boilers denoted equal opportunity for caustic corrosion in the seams of all, the rate of deterioration was not at all similar in the different boilers. The rate of deterioration of the boiler metal is dominated as well by other factors, prominent among which are the stresses, and in particular the cyclic stresses that are specific to the individual boiler and not common to all. When once erection is completed the stresses characteristic of each boiler are its irrevocable heritage.

The boiler metal, as furnished by the manufacturer, possesses an elastic and an endurance limit whose progress in the operation of the boiler is in one direction only, namely, downward. Corrosion accelerates this progress. Maintenance of protective films, integral with the metal, whether the oxide film of the boiler surfaces proper or a sulphate or phosphate film in the seams, by preventing corrosion, stays the downward progress. It seems logical to conclude that, beginning with its initial operation, every precaution should be taken to exclude from the boiler the processes of corrosion whether they be those of a neutral or a highly alkaline boiler water. The means of so doing are well recognized and definite.

Author's Closure

The subject of embrittlement has been under discussion so much during the last few years that any paper on this subject cannot attempt to cover all of its phases. The author, in preparing this article, made no attempt to discuss the entire subject. He felt that the interested parties were familiar with the conclusions reached in his previous publications (Bulletins Nos. 155 and 177, Engineering Experiment Station, University of Illinois; *Mechanical Engineering*, July, 1928, pp. 523-8). At no time has any attempt been made to infer that water alone was the cause of this difficulty.

The discussion which has been brought up illustrates very clearly that the whole question of embrittlement is a combination of stress and chemical attack. This has been the main conclusion of all the work which the author has conducted on this subject.

It is rather unfortunate that apparently a mistake has been made in reporting the exact details of the Crossett explosion. Since the author was not called to Crossett, but merely handled sections of the cracked plate which were forwarded to his laboratory, he had to rely entirely upon data furnished to him by the operators of the plant. The material reported is in accordance with the data furnished by the operators.

FSP-52-40

An Apparatus for the Determination of Creep at Elevated Temperatures

By A. E. WHITE,[1] C. L. CLARK,[2] and L. THOMASSEN,[3] ANN ARBOR, MICH.

This paper describes an apparatus designed for the purpose of measuring creep of metals at elevated temperatures. Due to the slight changes in dimensions encountered, such an apparatus must be sensitive to small variations. This particular machine is capable of measuring variations in length of the test specimen to about 2.8 millionths of an inch per inch.

The three essential parts of a creep-measuring apparatus are listed as (1) a means of applying and maintaining a constant load, (2) a means of obtaining and maintaining constant a given temperature, and (3) a means of accurately determining any deformation produced in the test piece. The manner in which this machine fulfils each of these conditions is described.

PRACTICALLY every one interested in the behavior of metals at elevated temperatures is now agreed upon the fact that at elevated temperatures, at least at those temperatures above a given fixed temperature, there is no stress below which a given metal will not continue to elongate or creep. The problem of determining safe working loads for a metal at these elevated temperatures therefore resolves itself not so much into one of determining the load at which no apparent creep would occur, but rather as the determination of the rate of creep for any given fixed load.

There is a wide range of permissible deformation allowable in the various industries. Some, as in oil-pipe still work, can allow elongations of the order of one per cent or greater per year, while in others, as in turbine manufacture, the permissible deformation is of the order of a small fraction of a per cent per year.

Even the relatively large rate of creep of 1 per cent per year involves only an elongation of 1.14 millionths (0.00000114) of an inch per inch per hour. If rates of creep of this magnitude are to be accurately determined it is necessary that either the test be conducted for long periods of time or else that an extremely sensitive measuring system be employed. As one is generally desirous of securing results as quickly as possible, the latter method is to be preferred.

A creep-testing apparatus has been developed at the University of Michigan which is capable of measuring elongation to 2.8 millionths (0.0000028) of an inch per inch on a 2-in. gage length. This machine is not claimed to be original in all respects, as in its design use has been made of ideas obtained from various sources. It is capable of yielding satisfactory results, however, and it is felt that a description of it might be proper at this time.

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[2] Assistant Investigator, Department of Engineering Research, University of Michigan.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

DESCRIPTION OF APPARATUS

There are at least three essential parts to an apparatus to be used for the determination of the creep characteristics of metals at elevated temperatures. First, there must be some means for applying and maintaining a constant load; second, a means of producing and maintaining a constant temperature; and third, a means of accurately determining any change in the dimensions of the test specimen that may occur.

Loading Mechanism

A view of the creep apparatus now in use is given in Fig. 1, which shows the apparatus to consist of 5 individual units. A sketch of the side and front view of one of them is shown in Fig.

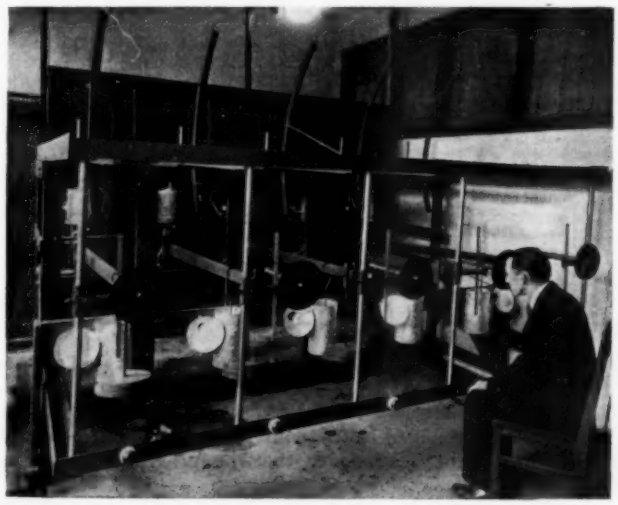


Fig. 1 Apparatus for Determining Rate of Creep of Metal

2. The loading device consists of a lever F pivoted at one end and with the test specimen fastened to the lever through a saddle arrangement close to the pivoted end. The load G is applied through another saddle arrangement at the opposite end, and the lever ratio is such that any applied load is multiplied 24 times at the test specimen. Knife edges are used at all points of bearing, and at some points, for example, at K, the knife edge is given a slight bevel so as to allow a certain amount of side adjustment.

The test specimen used has a diameter in its reduced section of 0.505 in. and an overall length of $5^1/_2$ in. Consequently the specimen is not of sufficient length to extend from the lever to the upper support but is screwed into adapters. As these adapters must themselves extend into the furnace, they were made from a highly alloyed heat-treated steel.

A rather wide range of loadings is possible with the set-up as shown, loads from 4000 to 80,000 lb. per sq. in. being easily

obtained. For loads below 4000 lb., a smaller lever is used, by means of which loads as low as 1000 lb. per sq. in. can be obtained.

Heating and Temperature-Control Apparatus

The specimen is heated with an electrical resistance furnace, *D*, Fig. 2. A sectional view of this furnace is shown in Fig. 3. The furnace is wound so that the temperature over the entire length of the test specimen is practically uniform. The temperature gradient over the 2-in. gage length should never be greater than 10 deg. fahr.

A clamp arrangement is attached to the top of the furnace which swings into slots in the rod into which the upper end of the test specimen is screwed. This allows the furnace to remain in the same relative position with respect to the test piece regardless of what amount of elongation occurs. Another slot is provided close to the bearing *K* so that the furnace may be raised and fastened above the test specimen.

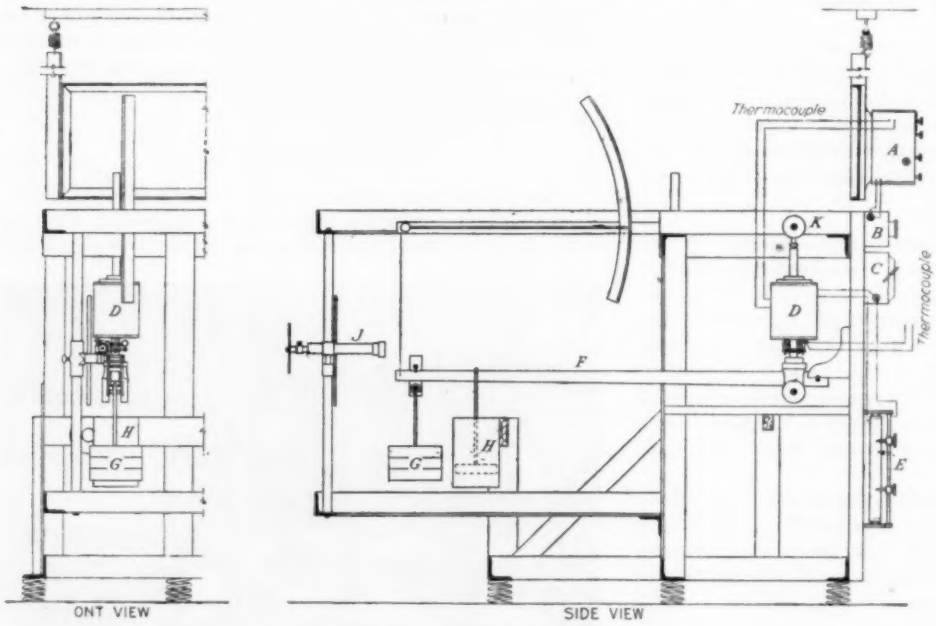


Fig. 2 Assembly of Flow-Test Apparatus

For maintaining the temperature constant, a separate Wilson Maeulen temperature controller is used for each individual furnace. The control thermocouple is placed close to the furnace winding, and a separate couple is fastened to the test piece itself. External resistances are placed in series with the furnace so that the current may be adjusted to a value just slightly above that required to maintain the furnace at the desired temperature. The automatic control throws off and on the full current.

The thermocouple for determining the actual temperatures of the test piece is wired to the outside of the specimen, care being taken to insure that the bead of the couple is pressed against the surface of the metal. These couples are connected to a ten-point Leeds and Northrup temperature recorder. Observations have shown the temperature at any given point not to vary by more than plus or minus 2 deg. cent.

Elongation-Measuring Device

An optical system is employed for determining any changes in length of the test specimen. A general view is shown in Figs. 1 and 2, while a detailed sketch is given in Fig. 3.

The apparatus consists of two parts. One part has for its function the transference of any motion of the test piece to some point outside the surrounding furnace, while the other measures any deformation that may occur.

A sketch of the mechanism employed for bringing any motion of the test piece outside the furnace is shown in Fig. 3. It consists of two collars, one of which is fastened to the upper gage marks on the test piece and the other to the lower ones. To each of these collars, rods are fastened which extend below the furnace. Rollers to which mirrors are attached are placed between the rods and held in place by friction produced by two springs drawing the rods together.

An elongation of the test piece will cause the lower collar and the inner rods to move downward with respect to the remaining rods. This movement will produce a rotation of the rollers, which in turn will rotate the mirrors fastened to these rollers.

On the first units constructed the rods were fastened to the upper and lower collars through the knife-edge arrangement shown in Fig. 3. Some difficulty was encountered in keeping the screws tight, and also in removing them at the end of the test. A slightly different arrangement is now being used which has yielded satisfactory results. This consists in replacing the knife seat with a projecting pin having a slight taper. The knife edge on the rod is replaced by a slightly tapered hole which fits over the projecting pin with their tapers in opposite direction. This amounts to replacing the knife edge with a "knife circle." The rod is held on to this pin by the same springs as hold the rollers in place.

The method used for recording the deformation is based upon the Gauss-Poggendorf principle of measuring angles by means of mirrors. There are two main methods that utilize the principle for measuring the elongation in the tensile test, namely, the Bauschinger and the Martens.

In the first, the mirrors are attached to cylindrical rollers which are turned when the specimen elongates, thereby transforming the linear motion of the rods coming from the above-described collars into rotation. In the Martens apparatus the elongation is transformed into rotation by inserting metal prisms of rhombic cross-section between the rods. As the specimen elongates the prisms will tip, rotating attached mirrors.

The angle of rotation is in both cases measured by observing through a telescope the movement of the images of a fixed scale in the mirror.

In both systems an adjustment of either the prisms or rollers is necessary after a certain amount of deformation has occurred. In the case of the Bauschinger method, this amount of measurable deformation is only limited by the range through which the scale can be observed; while with the Martens one must also take into consideration the danger of the prisms' turning over.

The measuring system employed in this creep apparatus is a modification of the Bauschinger which, in the opinion of the authors, has some advantages over the Martens arrangement. These may be listed as:

1 The edges of the prisms in the Martens apparatus will have some tendency to dig into the rods, especially during a long-time test, unless the edges of the prism are dull. In the first case the movement of the prisms will be impaired, while in the second, rotation would be produced and the exact diameter of the rotating part would be unknown.

2 A true cylindrical roller can be made more easily than a small prism of accurate dimensions. For the accuracy of measurements desired in this work a hardened steel would be suitable for the rollers, while with the prisms an extremely hard material, as stellite, is desirable.

3 The necessary readjustments that must be made in either

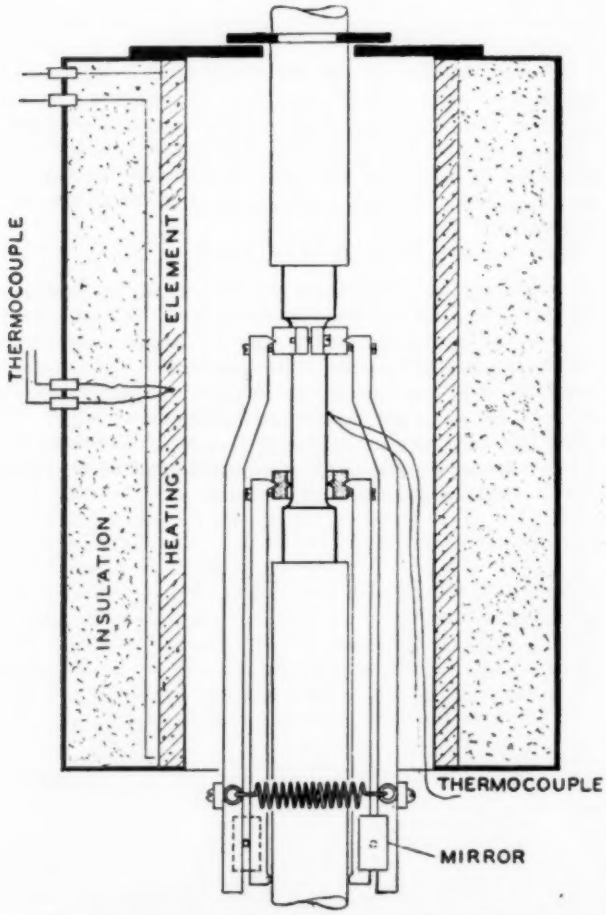


FIG. 3 FURNACE AND EXTENSOMETER USED IN FLOW TEST

system after a certain amount of deformation are more easily accomplished with the modified Bauschinger system than with the Martens. The ease of readjustment becomes of even more importance when it is considered that such changes must be made while the rods are heated, and that the apparatus must be disturbed as little as possible.

The actual amount of magnification that may be obtained with this type of system can be shown by considering the geometry of the system. In Fig. 4, R_1 and R_2 are the rods leading from the upper and lower collars, respectively. The solid and dotted circles are the positions of the roller before and after R_2 has been displaced downward. The lines AB and $A'B'$ are diagonals before and after rotation. The point B besides being rotated is translated a distance equal to the displacement of the center of the roller C. The sum of these two distances, the arc of the angle of reflection and the translation, constitutes the distance V. Considering only small angles, negligible error is made by assuming that a diagonal in the direction near to the perpendicular to the rods rotates as if the tangent of its angle is equal to V/d, where d is the diameter of the roller. Such a diagonal may be taken as a perpendicular to the surface of the mirror fastened to the roller, and therefore the mirror also rotates around an angle with tangent V/d.

If the angle at which the beam of light strikes the mirror is α, then the angle of the reflected beam would also be α. If the mirror be rotated through an angle β and it is required that the reflected beam have the same direction as before, namely, through the axis of the telescope, the incoming beam must suffer a change in direction of 2 $(\alpha + \beta)$. The angle of rotation of the mirror is thus doubled by reflection.

Now from simple geometry, it is seen that $V : d = s : l$. If

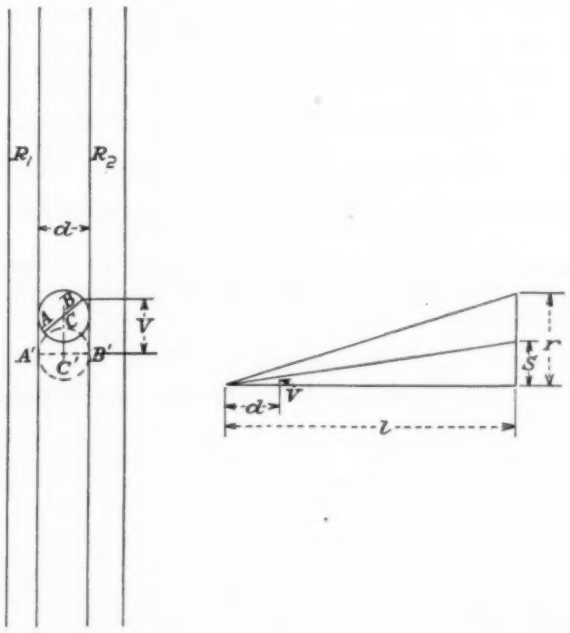


FIG. 4

only small angles are concerned, $s = {}^1/_2\, r$ without appreciable error. The equation thus becomes $V: d = r : 2l$, or $V = dr/2l$. .

In the set-up being considered the diameter of the rollers is 0.1 in. and the distance from the telescope to the face of the mirror 85.625 in. The ratio of V to r is therefore 0.1 : (2 × 85.625) or 1 : 1712.5.

The scale employed is graduated to half-millimeters. The readings are taken over a 2-in. gage length. On the assumption that readings can be accurately obtained to a quarter of a millimeter, the accuracy becomes 0.25 mm. on scale = 0.0000028 in. per inch on test piece.

PROCEDURE FOR OPERATION

After the extensometer apparatus has been assembled, the test specimen is brought to the desired temperature and maintained there for one day before the load is applied. Readings are taken immediately before and after the application of the load so the amount of elastic deformation may be obtained.

The temperature fluctuations, even though of the order of ± 2 deg. cent., are sufficient to cause a noticeable rotation of the mirrors. For this reason, when readings are being taken it is necessary to read a range on the scale rather than a definite

number. The reported reading is the mean value of the range.

The test at any given load is continued for a sufficient length of time to allow an accurate measurement of its rate of creep. It has generally been found that 400 hours is suitable. In some cases when the points fall almost exactly on a straight line, shorter periods of time may be used.

After the rate of creep for a given load is definitely known, then the load is either increased or decreased, depending on the amount of flow previously obtained. This process is repeated at least four times so as to give sufficient values to be plotted logarithmically.

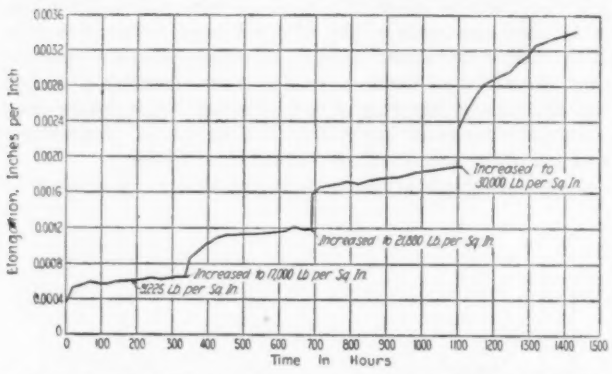


FIG. 5 CREEP TESTS ON SELECTED ALLOY STEEL AT 1000 DEG. FAHR. (538 DEG. CENT.)

(Chemical composition of "R" Steel: C, 0.02; Mn, 0.49; Si, 2.36; Cr, 18.00; W, 0.00; V, 0.01; Ni, 10.27.)

TYPICAL EXAMPLE OF RESULTS

As a typical example of the results which may be obtained with this apparatus, Fig. 5 is given. This particular alloy was tested at 1000 deg. fahr. and subjected to four different loads which ranged from 9225 lb. to 30,000 lb. per sq. in.

The smallest division on the elongation scale used represents a value of only one hundred-thousandth of an inch, yet in practically all cases the points fell on uniform curves.

It should be observed that in each case after the application of the load an appreciable time was required before the specimen began elongating at an approximately constant rate. For example, when the load was increased from 9225 lb. to 17,000 lb. per sq. in., over 100 hours were required for the rate of elongation to become constant. In some runs, even longer periods of time than this are required.

LIMITATIONS AND ADVANTAGES

The chief limitation to the use of this type of apparatus is that it must be practically free from all vibration. In our laboratories this has been accomplished by mounting the apparatus as a whole upon springs, by attaching dashpots, *H*, Fig. 2, to each lever, by suspending all control instruments free from the apparatus, and by using fine and flexible wires for all electrical connections.

The advantages may be listed as follows:

1 Readings of a high degree of sensitivity are obtainable
2 The readings are taken directly on the test piece itself, and are independent of any deformation which may occur in other parts of the apparatus
3 No side openings are necessary in the furnace which may affect the temperature distribution
4 The test piece itself is short enough so that conditions are more favorable for securing both uniformity of temperature and uniformity of metal.

FSP-52-41

Economics of Heat Transfer

By W. J. WOHLENBERG,[1] NEW HAVEN, CONN.

There is a set of proportions ideal for the operation of every manufacturing structure, the discovery of which is predicated upon ascertaining what may be called the economic units into which any structure may be divided. A statement of the cost of operation of an economic unit is here called the economic factor. Such factors assume, depending on their form, either maximum or minimum values for the ideal set of proportions. It is shown how an attack from this point of view may be employed in an attempt to evaluate ideal proportions for the heat-absorbing surface of a steam generator.

DISCUSSION of this subject will be begun by stating the following hypothesis: *There is a set of proportions which will prove most economical in operation for every engineering structure.*

The discovery of this particular set of proportions is predicated, first, on the ascertainment of what may be called the economic units into which any system or structure may be divided. The economic unit may, in this sense, be defined as any part of the system or structure within which a change of proportions will still leave the cost of operation for the other parts of the structure or system the same. A general statement of the cost of operating an economic unit may be set up in mathematical form, and such a statement is here called an economic factor. The unit should be so proportioned that such an economic factor assumes a minimum value. In another case the economic factor may express the amount of saving that is possible by a change in one or more dimensions of the structure. Then the value of the economic factor is obviously a maximum when it represents the most economical conditions.

For a steam-generating plant the economic unit is, in the last analysis, the steam plant; but it may be taken as a reasonable procedure when investigating the most economical extent of the heat-absorbing surface, to consider the extension of this surface as continuing from any point in the convection zone.

The economic factor may then be set up as follows:

$$I = \frac{R}{2000}\sum NL_s - \left[\frac{R}{2000}\sum Nq + X_s x_s + X_s x_s + X_c x_c\right] \quad \ldots\ldots [1]$$

in which

R = cost of burning fuel, say coal, dollars per ton
L_s = fuel saving effected by additional heat-absorbing surface, expressed as lb. of fuel per hr.
N = hours per year of operation at each load
q = extra fuel, lb. per hr., chargeable to auxiliary because of extension of surface
$X_s x_s$ = additional fixed charges plus maintenance on extra draft-producing equipment necessary for extra surface
$X_c x_c$ = additional fixed charges plus maintenance on extra additional building that may be required by the additional surface
$X_s x_s$ = fixed charges plus maintenance on extra surface in place.

In this equation I now represents the net saving in dollars per year because of the extra heat-absorbing surface. It is to be noted that, as the economic factor stands, no charges are made for extra labor or extra fuel costs due to stand-by losses. It is thus assumed for this case that extra labor and fuel stand-by costs are the same whether or not the extra heat-absorbing surface is present.

The most economical proportions of the steam generator under consideration are thus those which will yield a maximum value for the foregoing statement of the economic factor. Should even a small addition to the convection zone yield a negative value to this factor, it shows that the boiler convection zone is

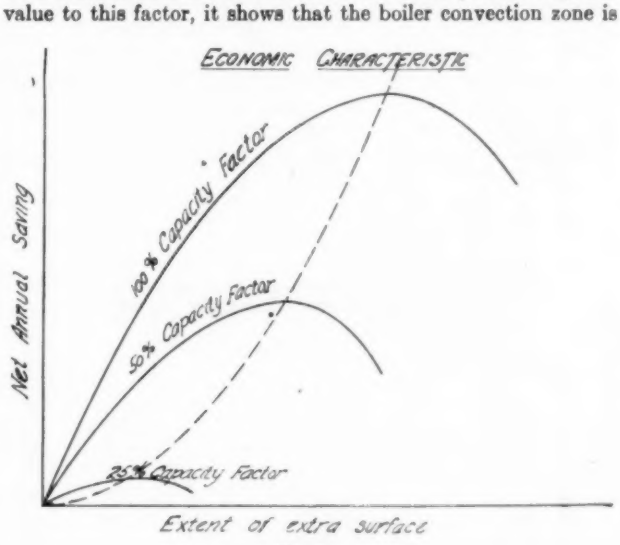


FIG. 1

already too large to be justified. A solution for values of the economic factor as the convection zone is extended in length—for instance, by economizer or air-preheater surface—will yield a set of curves of which those shown in Fig. 1 are typical, if the relation between first cost of the surface and its extent may be represented by a smooth continuous curve. This would be the case, for instance, if the cost per unit of surface were a constant regardless of its extent or if the change in cost per unit surface varied continuously and without any sharp breaks as the surface was extended.

The abscissa in Fig. 1 represents the extent of the additional surface, and the ordinate represents the dollars saved per year. If the unit is in operation all of the time, the capacity factor is 100 per cent, and the hours of operation per year N is equal to 8760. This is the condition under which the fixed charges have the least influence in determining where the surface should cease and the hourly fuel saving has the greatest influence. Therefore a larger extent of the surface is justifiable than for the lower capacity factors of 50 and 25 per cent.

The peaks of the curves shown in Fig. 1 may be used to plot a second curve shown in Fig. 2, in which the best surface is directly related to the capacity factor.

Whether or not such savings over the straight boiler operation will be realized depends upon how accurately the average capacity factor and the fuel cost over a period of time may be estimated. This is often very difficult, but in any case an analysis of this kind yields information by means of which guessing may be

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Lecture delivered at the S.P.E.E. Summer School for Teachers of Mechanical Engineering, held at Purdue University, Lafayette, Ind., June 27 to July 18, 1929.

based on a better foundation. It will at least show whether any worth-while savings are possible.

General Solution of the Problem

Inspection of the expression for the economic factor I shows

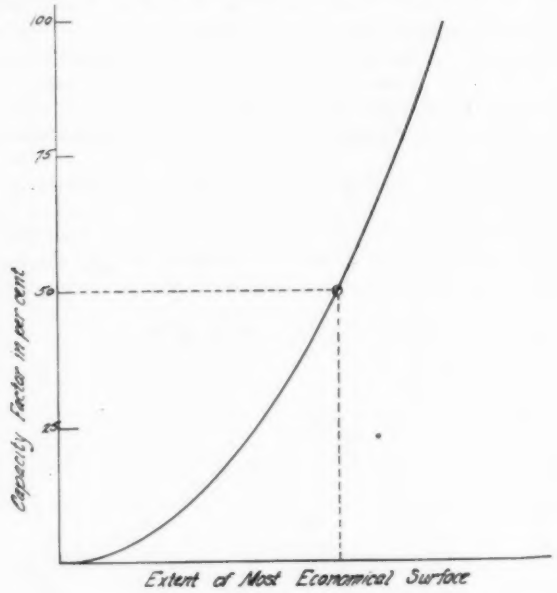


Fig. 2

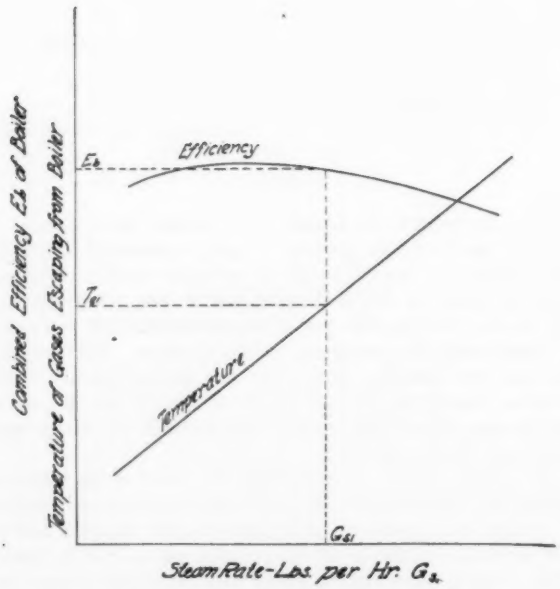


Fig. 3 Performance Characteristics of Boiler

that the only terms in it which relates directly to the performance as changed by the additional equipment are:

L_s = the fuel saved by the additional surface, and
q = the extra fuel requirement by auxiliaries.

These factors must thus be evaluated before any values can be worked out for the factor I. The problem thus divides itself into two parts. The first part is concerned with the influence on energy distribution and energy losses of the extension of the surface. The information yielded, from the energy point of view, is established once and for all time if the physical proportions of the unit are unchanged and if the data on which the computations have been made are reliable. It is in the second part of the solution that such factors as the fuel cost R, the hours of operation N, and the costs X and x are introduced. Such quantities may change from time to time and are subject to outside influences, and values adopted for them in arriving at the best proportions of the equipment must be chosen only after weighing each item very carefully.

Solution for the Energy Distribution

In beginning this phase of the solution of the problem let us assume that both the total output, pounds per hour per steam generator, and the fuel have been determined. Also that the combustion arrangements have been fixed so that it is possible to

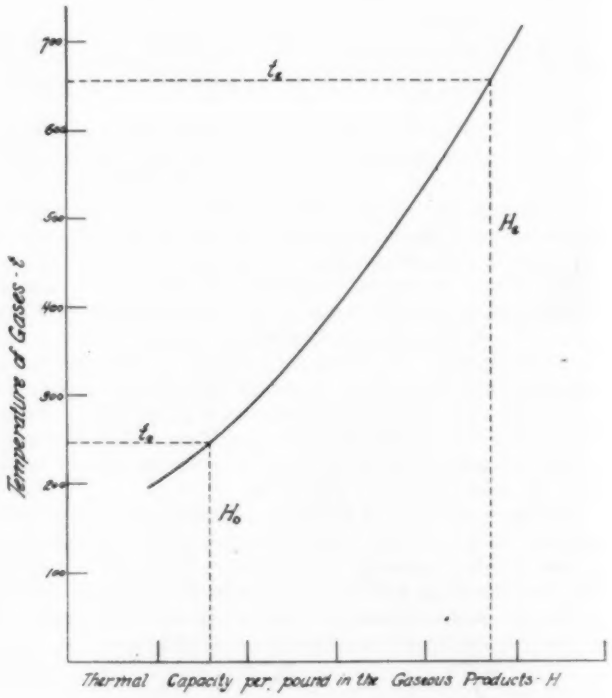


Fig. 4 Thermal-Capacity Curve

predict the amount of air that will be required for combustion for each pound of fuel that is burned.

The steam-generating unit will consist of the combustion apparatus and the boiler, plus such additional heating surface in the form, say, of economizer surface as it is found economical to install. It is the influence of the extent of this additional surface which it is proposed to investigate.

The available data on operation include a curve, Fig. 3, furnished by the manufacturer, showing the relation between the temperature of the gases escaping from the boiler and the steam output for the combustion conditions as they may exist when burning the fuel under consideration. A second curve on the same figure shows the relation of the combined boiler efficiency to the steam output, and thus determines the fuel cost if the unit is operated without any extension of the heat-absorbing surface.

The additional surface will reduce the energy loss in the escaping gases by reducing the temperature of these gases. This reduction in loss may be represented as a fuel saving, whose magnitude for a given rate of steam production depends on the hours

per year at which this unit is operated at the specified rate or rates.

The energy as heat that is saved by the extra surface is of course fixed by the reduced thermal capacity of the gases as they escape at the lower temperatures, and the amount of the extra surface that is required depends upon the mean temperature differences between the gases and the surface and on the velocity of the gases as they sweep over the given arrangements of these surfaces. Such a curve as is shown in Fig. 4 is thus necessary for accurate evaluation of both the reduction in thermal loss and the surface required. It is a segment of the curve showing the relation between the thermal capacities per pound of the gaseous product and the temperatures of the gases.

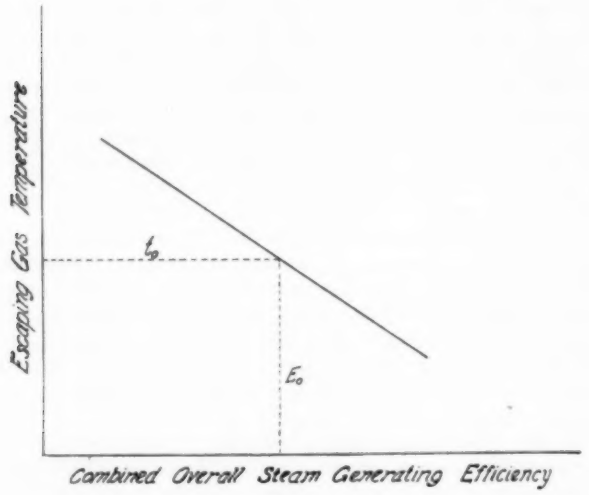


FIG. 5

It represents the temperature path along which the gases are cooled.

Another curve shown in Fig. 5 shows the relation between the overall steam-generator efficiency E_o and the final escaping-gas temperature t_o. Values for its determination are derived from heat balances, in which may be included for each of a number of final temperatures the following losses:

1 Loss due to incomplete combustion
2 Loss due to radiation and unaccounted for
3 Loss due to energy carried out in final escaping gases.

The magnitude of the third of these losses may be taken directly from the thermal-capacity curve shown in Fig. 4, when the gaseous weight P per pound of fuel is known.

With this information the curve shown in Fig. 6 may be found. Here the final escaping-gas temperature t_o is related to the gas weight g in pounds per pound of steam generated. A point on the curve may be found for any temperature t_o by determining first the pounds of fuel that will be required to generate a pound of steam. Thus

$$G = \frac{(i_1 - i_f')}{E_o \cdot V} \quad \ldots\ldots\ldots\ldots\ldots\ldots [2]$$

in which $(i_1 - i_f')$ represents the heat added per pound of steam E_o, the overall efficiency at temperature t_o, V the calorific value of the fuel, and G the pounds of fuel per pound of steam. The gas weight in pounds per pound of steam then follows from the relation

$$P \cdot G = g \quad \ldots\ldots\ldots\ldots\ldots\ldots [3]$$

in which, as before, P represents the total gas weight per pound of fuel.

The relations shown by the curves in Figs. 4, 5, and 6 are obviously determined independently of the equipment involved. They set up for the assumed combustion conditions and final escaping-gas temperatures what the energy relations must be. The equipment will now have to be proportioned to meet such of these conditions as we choose to fit into the picture.

Temperature of Gases Escaping From the Boiler When the Economizer Is in Place

For given conditions as to steam generation, initial feedwater temperature, and final steam heat content, the temperature of the gases escaping from the *boiler* proper is lower when the economizer surface is in place than it would be should no such extra surface be installed. This reduction in temperature at this point is due to that reduction in gas weight when the extra surface is in place which results from the reduced fuel consumption. This in turn reduces the gas temperature at any section of the gas stream if the combustion conditions remain constant.

Now in order to find out how much surface should be installed in the economizer it is obviously necessary to know what the temperature of the gases entering the economizer is, when it

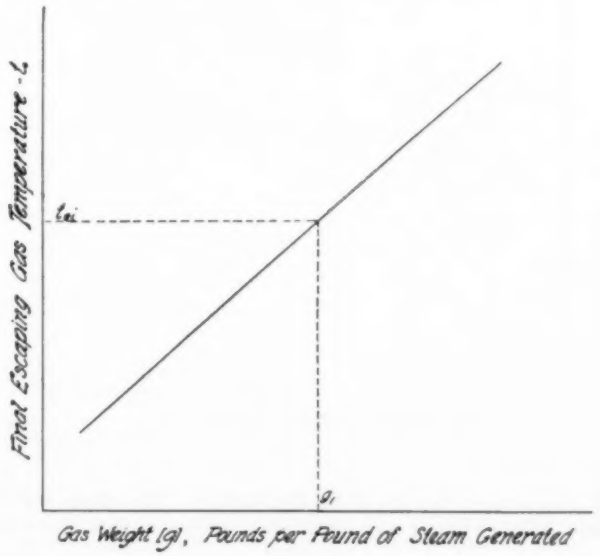


FIG. 6

(the economizer) is in place beyond the convection zone of the boiler. It is therefore necessary to relate the temperature t_e of the gases escaping from the boiler to the gas weight flowing through the boiler when the boiler-surface temperatures are substantially as they would be with the economizer in place.

Fortunately, with the data at hand this is a rather simple problem, because the boiler-surface temperatures are fixed by the saturated-steam temperatures going with the pressure which exists. This is not altered when additional heat-absorbing surface is installed beyond the so-called boiler proper. We now use the data furnished from the manufacturer's curve, Fig. 3, together with that shown in Fig. 6 to compute values for a new curve relating the gas weight flowing through the boiler to the temperature of the gases escaping from the boiler, which is the temperature at which they enter the economizer. Thus, for a given steam-generation rate, G_{s1} pounds per hour, the gas temperature from Fig. 3 is shown to be equal to t_{e1}. From Fig. 6 we find that at the temperature t_{e1} the gas weight per pound of

steam is g_1. The total gas weight, pounds per hour, for the temperature t_{e1} escaping the boiler is thus

$$g_1 \times G_{s1} = G_b$$

By computing a number of such points the curve of Fig. 7 is established.

The only error that is introduced in this computation results from the fact that the superheater, if one is in place, will absorb less heat with the economizer in place than it does if the boiler alone is absorbing the energy. The same quantity of steam is of course flowing through the superheater in either case, but less gas weight is flowing over the superheater in the first case than in the second. This may make it necessary to use a slightly larger superheating area under the second as compared to the first condition, but the error introduced in the total gas-temperature curve is from this cause only a very small one.

Energy Relations in the Economizer

With the foregoing information it is now possible to determine for any final escaping-gas temperature t_e the temperature t_s

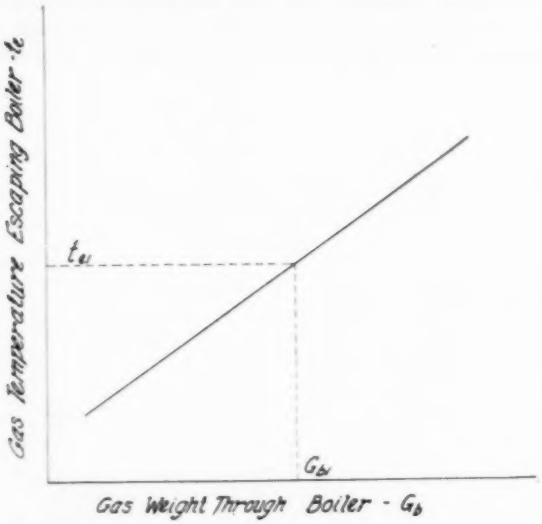


Fig. 7

of the gases entering the economizer surface. The energy which must be absorbed at the economizer surface is then equal to

$$G_s \cdot G \cdot P\,(H_s - H_o) \ldots\ldots\ldots\ldots\ldots\ldots [4]$$

in which the thermal capacities $(H_s - H_o)$ are taken from the curve of Fig. 4 corresponding to the temperatures t_s and t_o. The fuel weight G is determined from Equation [2], i.e.,

$$G = \frac{(i_1 - i_f')}{E_o \cdot V}$$

in which the overall efficiency E_o may be taken from the curve shown in Fig. 5.

But the energy shown by [4] must be absorbed in the feedwater, whence

$$G_s \cdot G \cdot P \cdot (H_s - H_o) = G_s(\Theta_s - \Theta_o) \ldots\ldots\ldots [5]$$

in which Θ_o and Θ_s represent respectively the temperature of the feedwater entering and of that leaving the economizer surface. This equation is solved for the temperature Θ_s of the water escaping from the economizer.

There are now available, with one important exception, sufficient data to solve for the surface requirements in the economizer. Thus the total energy to be absorbed is represented by the relation

$$G_s \cdot G \cdot P(H_s - H_o)$$

and the temperature differences at the two ends of the economizer surface are

$$(t_s - \Theta_s) \text{ and } (t_o - \Theta_o)$$

The important exception is the remaining data which are necessary to establish what the unit rate of heat transfer may be for this surface.

Mode of Heat Transfer in the Convection Zone

The transfer of energy from gases to a surface across which they are flowing is in reality a very involved process. The final stage of the transfer is conduction through a thin surface film of gas which adheres to the metal surface. If any element of the metallic surface is to receive considerable energy by this process, the gases containing the energy as heat must be washed up to it at a high rate. This washing or scrubbing action is important also in reducing the thickness of the conducting gaseous film, which offers the principal resistance to the flow of energy into the metallic surface.

It is not only the general state of the motion of the gases, but also the smallest details of this motion, such as turbulence, which play an important part in the rate at which the energy transfer takes place. The density of the fluid, its viscosity, and also its thermal capacity enter into the problem. Finally, for the higher temperatures the chemical composition of the gas may have to be considered because certain constituents radiate energy as heat.

At the usual temperatures which exist in the lower part of the convection zone of a steam generator the radiational transfer is not a considerable factor. The temperatures are low here, and for close tube spacings the individual gas streams have small mean diameters, so that the thickness concentration functions of the radiating constituents of the gases have but small values. For such cases at least, then, the variables may be reduced to the ordinary variety aforementioned.

Boussinesq was the first to derive a complete rational equation based on the dimensional reasoning for the so-called coefficient of heat transfer. His equation involved the tube diameter, the velocity of the fluid, its density, its conductivity, its viscosity, and its specific heat or thermal capacity. It was developed for gases flowing through tubes, and may be applied with confidence only to rather simple cases when the conditions are within the limits for which the constants were determined. It assumes uniform thickness of the final conducting film, and so may not be applied in its original form to the case of gases flowing transversely across the outside of tubes and through which water or steam is flowing. The latter is the usual arrangement for the larger steam generator.

In a paper presented before The American Society of Mechanical Engineers in 1926, Chappell and McAdams include the derivation of a formula which, as an example, it is proposed for use in such cases. In their development they begin with the work of Boussinesq and submit this, together with some of the later developments, to a critical review. The final formula for staggered tubes is stated as follows:

$$h = \frac{0.8\sqrt[3]{T_f}\,V^{(0.60\,+\,0.08\,\log\,D)}}{D^{0.53}}\left(1 + \frac{D}{W}\right) \ldots\ldots [6]$$

in which

h = coefficient of heat transfer B.t.u. per sq. ft. per degree per hr.

T_f = arithmetical mean temperature of gas stream and tubes, deg. fahr. abs.
V = mass velocity of gas, lb. per sec. per sq. ft. of free area of cross-section
D = outside diameter of pipe in inches
W = distance in inches between center lines of adjacent pipes in a row taken at right angles to air flow.

The last term of this equation is a correction factor which is applied when more than a single tube is involved and when, furthermore, the tubes are in a staggered arrangement. In using such a formula it should be kept in mind that the constants in it have been determined for certain specific arrangements, sizes of tubes, and velocities. If in another case there is a wide divergence from these conditions, they may not fit. It is thus always better to have a few check points, at least, for such arrangements and conditions as are under consideration. The formula then serves as a basis for establishing the relativity of one condition with respect to another, and the whole scene is shifted and anchored to such values of the constants as will suit the few points of reliable information which may be at hand concerning the particular type of arrangement under consideration.

Selecting the Surface Arrangement

Assuming that either from sufficient direct experimental evidence there is sufficient information for determination of the values of the coefficient of heat transfer and their variations or that suitable constants have been established for such a formula as [6], the solution of the problem may be continued. At this point it will be necessary to assume either the surface arrangement and its proportions or the gas-flow conditions. In the first case the resulting gas flow will have to be determined usually by trial and error, and in the second case the flow areas must be so proportioned that the assumed conditions of flow result. This is usually the more simple form of procedure.

By means of it a number of different cases may be investigated to show, for instance, the influence of gas velocity on the economic proportioning of the surface. For a given set of conditions otherwise, there would then result one economic characteristic for each of the assumed gas-flow rates.

Having determined the flow conditions and hence the coefficient of heat transfer, the extent of the surface is determined for any given final temperature by the usual procedure. The mean-temperature difference is computed as before outlined by reference to the gas and water temperatures at the boiler and economizer ends, respectively. The surface is then made just large enough so that it will absorb the energy which must be transferred from the gases in order to reduce their temperature from that of the boiler exit to that assumed for the economizer exit. The solution is carried through for a number of final temperatures t_e.

Fuel for Extra Draft Requirement

In some instances the increase in draft loss through the system may provide the limiting conditions. If it were not for the rapidly increasing resistance to the flow produced by increasing the gas velocities, it would be desirable, as has been shown, to greatly increase these in order to increase the rate of heat transfer. Indeed, for a given rate of gas flow in pounds per hour over a given amount of surface, the higher the mean gas velocity the lower the final escaping-gas temperature and hence losses. From this point of view by itself, it would thus be desirable to increase the gas velocities above present practice, up to such magnitudes that the resulting increased heat-transfer rate would attain such levels as to cause the resulting rise in temperature of the metal itself to become the condition limiting the gas velocities.

The resistance to flow, however, increases very rapidly as the velocities become higher and higher, and long before the above limiting surface temperatures are reached the required extra power expenditure for draft loss is prohibitive.

The resistance to flow or draft loss may be set up in an approximate mathematical form as follows:

$$h = h_o \left(\frac{G}{G_o}\right)^2 \cdot \frac{T}{T_o} \cdot \frac{N}{N_o} \quad [7]$$

In this formula G_o, T_o, and N_o represent, respectively, rate of gas flow (lb. per sq. ft. flow area per hr.), mean absolute temperature of the gases, and length of the flow path as measured in any convenient units for a draft loss of h_o inches of water. The length of the flow path may be expressed, for example, as the number of tubes deep in the direction of the gas flow, or as

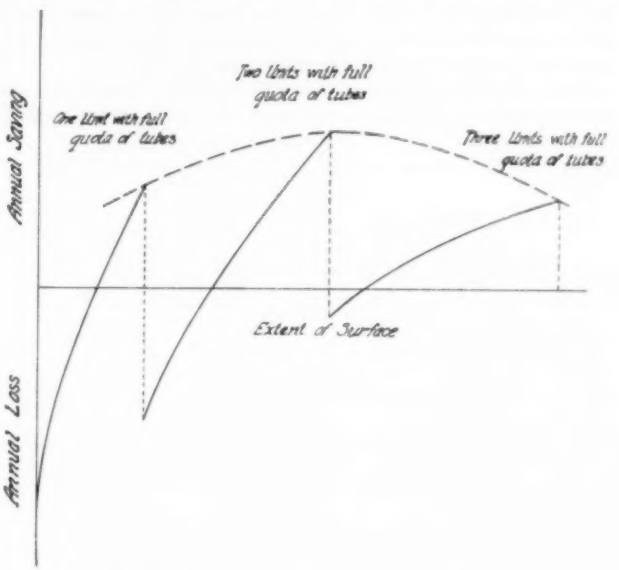


Fig. 8 Economic Characteristic

the length of the tubes in case the flow is parallel to the tubes.

For any other set of flow conditions, then, with substantially the same type of tube arrangement and tube size the draft loss is h inches of water when the gas-flow rate is G lb., the mean absolute temperature T degrees, and the flow path is, say, N tubes deep.

The extra power requirements for draft may thus be computed after having determined the extra draft loss. The extra fuel requirements for this additional power will then depend upon other conditions in the plant such as the existing heat balance, i.e., whether or not the exhaust steam from auxiliary units may all be condensed for feed heating, or whether an electric motor is used for driving the fan and which is supplied from a house generator or indirectly from the main generators.

This now concludes the discussion of the problem as to methods of evaluating the various quantities of energy that are involved. That which remains is concerned with the substitution of values in the equation representing the economic factor.

Setting Up the Economic Factor

At this stage of the solution the following information is available for the given rate of steam generation with the given conditions of combustion: The temperature t_b of the gases escaping from the boiler when no economizer surface is in place. The temperature t_e of the gases escaping from the economizer

when the unit consists of the boiler plus the economizer. The combined efficiencies of operation in the two cases may now be obtained from the temperature efficiency characteristic of Fig. 5. Let the efficiency for the temperature t_b be E_b and that for t_e be E_e. The fuel saving, L_s lb. per hour, because of the extra surface is now

$$L_s = \frac{Q}{V}\left(\frac{1}{E_b} - \frac{1}{E_e}\right)$$

in which

Q = heat absorbed by the water and steam in B.t.u. per hour, and

V = calorific value of the fuel.

It should be noted that the credited heat saving is not exactly equal to the heat absorbed by the economizer surface, because, when this is in place, the boiler proper, by virtue of the reduced gas flow, absorbs a little less energy than it does when this is not there. The economizer thus absorbs a little more energy than is represented by the actual heat saving due to it.

The only other factor in Equation [1] which merits any discussion now is perhaps the first cost X_s of the extra surface. How this varies with the extent of this surface is of course determined by the variation of the cost per unit of surface with its extent. In some special cases the cost per unit of surface may remain constant or vary in a continuous manner with the extent of this surface, but usually this is not so. Let us consider for a moment an economizer which is made up of units comprising upper and lower drums connected by the surface in the form of tubes. The principal cost of such a structure is in the drums. The first square foot of tube surface would then have to be charged with the cost of the tubes plus the drums, and for small amounts of tube surface in conjunction with each pair of drums the cost per unit of effective surface would be exceedingly high. As the number of tubes between any set of drums is increased, the unit cost is reduced, and finally with the full quota of tubes in place it has been reduced to a reasonable value. An economic characteristic developed on the same basis as that shown in Fig. 1 may, for such cases, appear as shown in Fig. 8.

It is obvious that for such conditions the real unit of heat-absorbing surface may be a set of drums with its full quota of tubes, and the problem resolves itself into the question of how many such units should be installed.

FSP-52-42

Effect of Air and Turbulence on Condenser-Tube Deterioration

Progress Report No. 3 of the A.S.M.E. Special Research Committee on Condenser Tubes,[1] Comprising the 1929 Report of Its Sub-Committee on Questionnaire[2]

DURING the past year the Sub-Committee has been engaged in an intensive study to devise means of determining the quantities of air and the degree of turbulence within the condenser water boxes and tubes, and correlating the results with tube deterioration. The work so far has been carried out in the Hudson Avenue and the Gold Street stations of the Brooklyn Edison Company, supplemented by laboratory experiments at Hudson Avenue. The methods used are illustrated in the diagrams and photographs included in this report.

The field work consisted of obtaining samples at the points illustrated in Fig. 1 at various points in the circulating-water cycle. Complete data are tabulated and show a decided increase in the volume of air from the inlet tunnel to the condenser discharge due to the release of the air with drop in pressure.

In order to visualize the free-air content, glass tubes were attached to the condenser heads as shown in Figs. 2 and 3. In Fig. 2 the samples were taken from the inlet water box of the condenser and discharged into the outlet water box. These tubes were attached at right angles to the face of the condenser head in prolongation of the actual condenser tubes. A continuous stream of small bubbles was noted with intermittent flow of large bubbles which broke up into a large spiral of smaller bubbles. The vertical glasses were installed both at Hudson Avenue and Gold Street on two different types of condensers. One was a single pass at Hudson Avenue and the other a two pass at Gold Street. The results were about the same and are illustrated in Fig. 3. The water level at the top of the pass surged up and down in 10- to 15-sec. cycles for a distance of from 12 to 18 in., thus indicating that the upper tubes were entirely empty at times and then filled with water. Figs. 4 and 5 show the tube-failure records for two condensers. It will be noted that the failures are coincident with the tubes which are subject to this intermittent flow of air and water, thus verifying the findings of a year ago that turbulence and entrained air are one of the principal causes of tube failure.

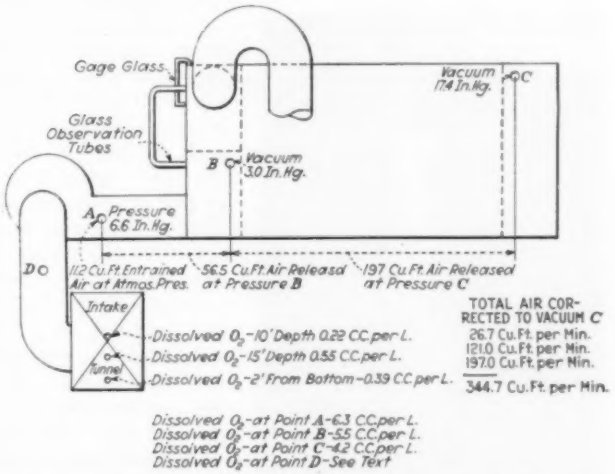


FIG. 1 DIAGRAMMATIC SKETCH OF INVESTIGATION OF AIR QUANTITIES IN CONDENSER

In carrying out the various experiments in the field some rather peculiar conditions were found which have not been fully solved as yet, the principal one being an apparent infiltration of air through the circulating-pump glands which on checking were tight and properly sealed when in operation. To attempt to solve this problem a laboratory set up was made as shown in Fig. 6. It consists of a paddle in a bottle driven by a shaft through a tightly fitting cork. The shaft was rotated with the bottle completely filled with water. Air bubbles gradually seeped through the cork along the shaft and down the impeller. The immediate conclusion appears to be that no matter how tightly sealed a shaft may be mechanically a film of air will work through under operating conditions unless an additional seal such as a water seal is provided. This is further borne out in the field, because when a circulating pump is shut down, the water in the condenser and pump will remain from 3 to 5 hr. before the vacuum is broken sufficiently to release the suction. The other source of entrained air which is under investigation is

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[1] The personnel of this Committee is as follows:
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Presented at the Annual Meeting of the A.S.M.E., New York, December 4, 1929.

TABLE 1 AIR QUANTITIES IN NO. 1 CONDENSER CIRCULATING WATER

Time	Circ. pump speed	Condenser circ. water temp., deg. fahr. Tide	Inlet	End first pass	Outlet	Vacuum, in. hg. Pump dischg.	Inlet top first pass	Inlet top second pass	Dissolved oxygen, cu. cm. per liter Pump dischg.	Inlet top first pass	Inlet top second pass	Air released, cu. ft. per min. corresponding to vac. From pump discharge to inlet first pass	From inlet first pass to inlet second pass	Entrained air entering condenser, cu. ft. per min., corrected to second pass vacuum	Total air corr. to vac. inlet second pass, cu. ft. per min.
April 18, 1929															
10:00 a.m.	Low	95.4	46.0	47.5	52.0	1.7	11.5	22.0		4.6	4.1		83.6	42.0	
10:15 a.m.	High	95.8	45.0	46.5	49.0	6.6 pressure	3.0	17.4	6.3	5.5	4.2	56.5	197.0	26.7	344.7
3:00 p.m.	Low	97.9	45.5	48.5	53.0	1.2	10.3	20.8		5.1	3.7		203.0	36.5	
3:15 p.m.	High	98.4	45.5	48.0	50.5	8.0 pressure	1.2	16.1	5.5	4.9	4.1	39.8	110.0	24.2	216.7
April 20, 1929															
10:00 a.m. to 10:15 a.m.	Low	97.1	45.5	47.5	52.0	1.1	9.7	21.3		3.6	2.2		215.0	38.6	
3:00 p.m. to 3:15 p.m.	High	97.0	45.8	47.0	50.0	7.6 pressure	1.1	17.2	4.9	4.5	3.6	26.5	134.0	26.2	220.0

A sample taken at the pump discharge at atmospheric pressure shows 16 cc. entrained air in 19 liters, equivalent to 11.2 cu. ft. of free air per minute.
The quantity of circulating water at low speed = 70,000 g.p.m., while at high speed this is increased to 100,000 g.p.m.
The pH value of the circulating water is 7.5 (slightly alkaline).

the possibility of a vortex in the suction leg. In order to determine the effect of such a vortex, samples were taken on the suction leg during operation of the pump and then periodically after the pump was shut down so that any air in the center of the suction leg would have an opportunity to diffuse. Samples obtained at point *D*, Fig. 1, with the pump operating indicated 0.10 cu. cm. of dissolved oxygen per liter. Twenty minutes after the pump was shut down the dissolved oxygen at the same point was 1.5 cu. cm. per liter and after 30 min. this had increased to 1.8 cu. cm. per liter. The increasing amount of air near the side of the pipe after the shut down of the pump would indicate that air was being sucked through the center of the column of water, and that in passing through the impeller this air is more thoroughly mixed with the water, thus accounting for the higher air content after passing through the pump, rather than because of gland leakage.

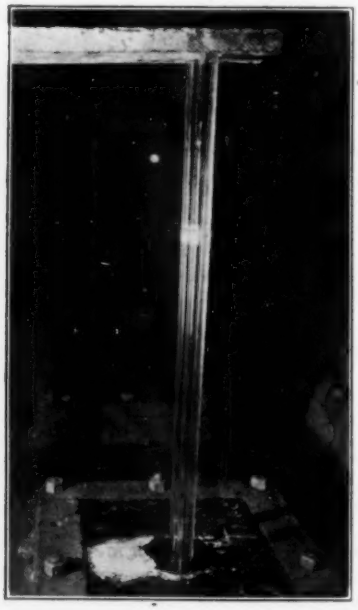


FIG. 2 GLASS TUBE ATTACHED TO NO. 3 CONDENSER AT HUDSON AVENUE STATION

In the laboratory, samples of circulating water were placed in a bottle, subjected to various degrees of vacuum, and the release of entrained air noted. This set-up is shown in Fig. 7.

Table 1 shows a maximum of 344.7 cu. ft. per min. of entrained air entering the second pass, which corresponds to 2.57 per cent of the total flow.

From the foregoing summary of the work already done it is evident that the results warrant a further study.

PLAN FOR FURTHER WORK

In a further study of condenser-tube deterioration the sources of information should be selected carefully. A list of approximately 12 operating companies and all the larger condenser and tube manufacturing companies should be obtained and their cooperation secured, preferably by personal contact, in an intensive study of the situation under the control of the sub-committee.

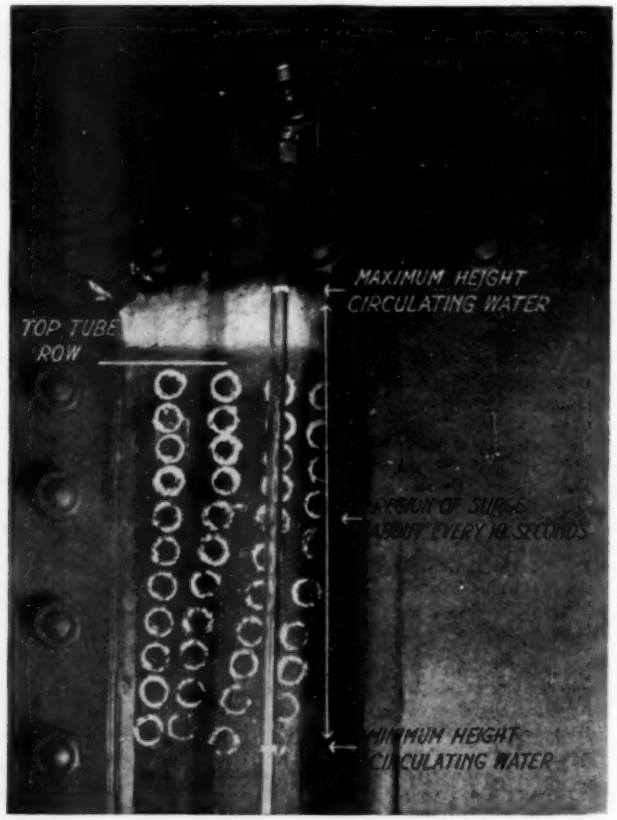


FIG. 3 GLASS TUBE ATTACHED TO NO. 5 CONDENSER AT HUDSON AVENUE STATION

Printed forms for data, sample attached, should be prepared periodically, and the operating companies requested to fill these in and send them to the sub-committee chairman. Periodic field inspection should be made by one of the sub-committee

members. An outline of the work to be covered is as follows:

1 *a* The quantity of air entering with and released from the circulating water at various points within the condenser and piping should be determined
   b How this air gets into the condenser should be established
   c Methods should be devised to reduce the quantity of free and released air
   d The effect upon tube deterioration should be noted by varying the quantity of air.
2 *a* The turbulence within the water-boxes and tubes should be studied
   b Means should be devised to reduce this turbulence
   c The effect upon tube deterioration should be noted by varying the degree of turbulence
   d The extent of the water level fluctuation in the upper rows of tubes and the length of time these contain no water should be determined
3 The pH value of the circulating water should be determined.

Suggested methods of obtaining information are as follows: The quantity of free air entering with the circulating water may be obtained by permitting a sample bottle of known volume to

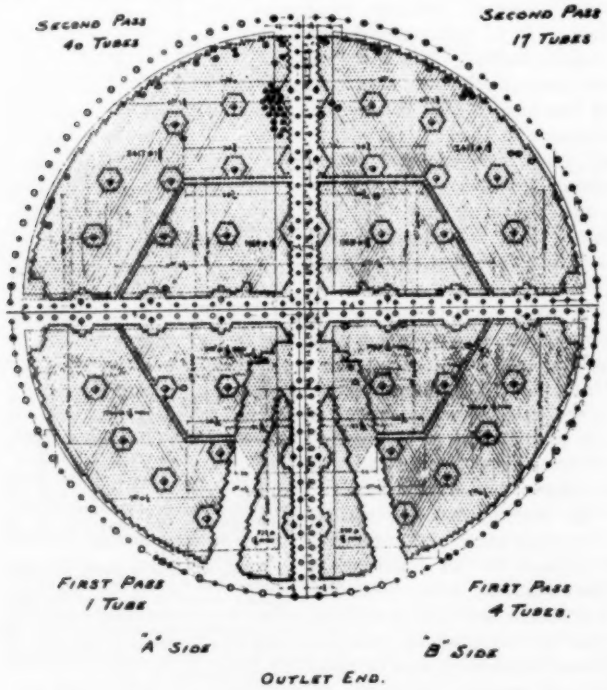


FIG. 4 TUBE-FAILURE RECORD FOR CONDENSER NO. 1

overflow for several minutes from a tap at the pump discharge and then rapidly corking the sample bottle. By permitting the sample to rest for a short period the free air will rise to the surface of the water in the bottle and measurement will then determine the proportion of air and water.

To obtain the quantities of air released from the circulating water in passing through the condenser it will be necessary to use an indirect method—i.e., determination of the oxygen content of the water in various parts of the system, and then calculating the corresponding quantities of air on the basis of 21 per cent O_2 and 79 per cent N_2 corrected for the existing pressures.

Samples for the determination of the O_2 content must be taken at the pressure existing at the point under consideration, and this will require a special sampling device. Fig. 8 shows such a device and indicates its method of operation.

The installations upon which these experiments are to be carried out should represent the best and worst conditions in a single plant in so far as tube deterioration is concerned. To expedite the work the installation selected should have complete records on tube failures in order that the present rate of tube failure may be established. If the records do not disclose the preceding rate of failure the condenser should be retubed and its rate of tube failure determined.

Having established the rate of failure and noted the quantities of released air, some method of reducing the quantity of released air flowing through the tubes should be instituted.

In order to accomplish this there have been several methods developed. One is to connect an air off-take from the top of the

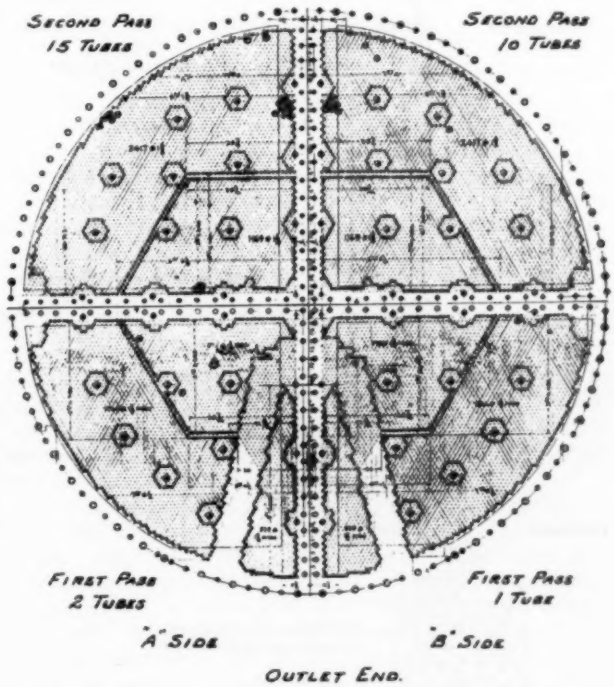


FIG. 5 TUBE-FAILURE RECORD FOR CONDENSER NO. 3

inlet waterboxes to the tail pipe. This method by-passes a large proportion of the released air. Another is to prevent the release of air in such way that it can concentrate in the upper tubes, by making the waterboxes shallow vertically; thus having a series of inlet boxes fed by separate inlets from the pump. Another proposed method is to proportion the inlet box to the flow.

On a small installation it might be feasible to put in a large enough pump to keep the circulating water under pressure throughout the condenser. The vacuum in the tail pipe may be broken by venting this to the atmosphere or a suitable baffle or damper arrangement could be placed in the entrance of the discharge piping thus restricting the flow and increasing the pressure within the condenser.

After these changes have been made the quantities of air (both free and released) should again be determined and a new rate of tube failure established and compared with the original.

STUDIES OF TURBULENCE WITHIN THE WATER BOXES AND TUBES

A study of turbulence is extremely difficult owing to the inability to visualize the action of the water as it passes through the water boxes and tubes.

No definite conclusions can be drawn unless some method is devised for observing the flow of the circulating water. A more or less elaborate set-up of instruments could be used to determine

the pressures, the velocities, and the direction of these velocities. The reliability of the results, however, would be doubtful inasmuch as the obstruction to free flow imposed by the instruments themselves may have a marked effect upon the characteristics that normally obtain.

Some suggestions for obtaining observed data are:

1 An operating condenser fitted with ample glass observation windows and illumination.

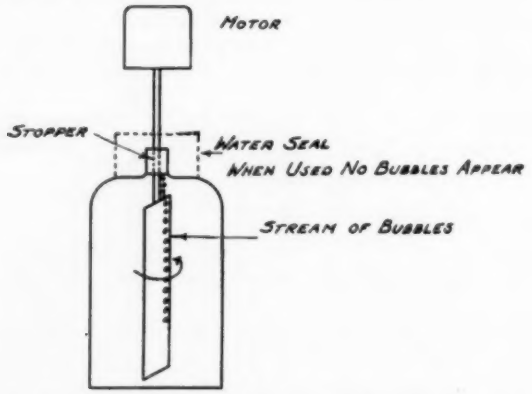


Fig. 6 Apparatus Used to Determine Infiltration of Air

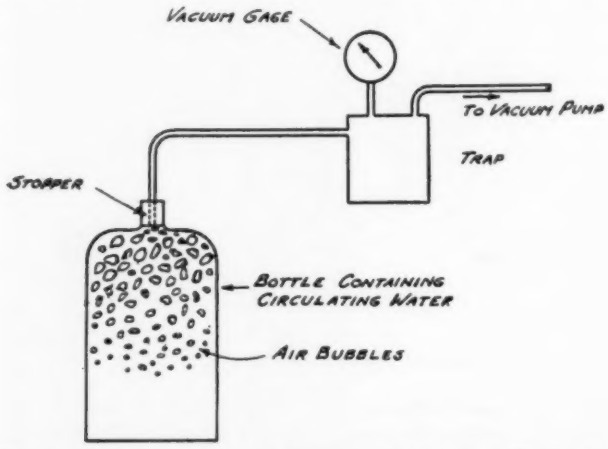


Fig. 7 Apparatus Used to Subject Circulating Water to Various Degrees of Vacuum and to Note the Release of Entrained Air

2 A variable tap and return arrangement utilizing a glass tube.

The operation in an experimental condenser fitted with liberal window space would approximate operation in an actual installation and would serve as a guide for comparison. By simulating water-box conditions as observed by several different applications of suggestion 1, the resulting tube flow could be noted, or by simulating observations as per suggestion 2, the corresponding water-box conditions should compare.

Also the effect of changes in tube entrance and water-box design could be observed readily, and any such experiments should first be tried in a condenser fitted with window installation. In designing this condenser provision should be made for varying the volume of the water boxes and their shape. It is suggested that the manufacturers of condensers and tubes be induced to build such an installation for the use of the sub-committee.

A tap and return arrangement could be used on an operating installation and an idea of the character of the flow within the tubes noted, and the movements of the air bubbles within the tubes observed. Such an arrangement should be tapped into the water-box head and returned to a point at different pressure. By tapping into the water-box head the direction of flow will be directly opposite that obtaining in the actual tubes within the condenser. This, however, cannot be avoided and will produce more accurate results than by tapping into the side of the water box, thereby causing an additional 90 deg. bend in the flow from water box to tube. This additional bend would materially affect the character of flow at the entrance to and within the tubes.

The condition of water flow within the water boxes and at the tube entrance can be noted best by observation, and the degree of this will depend upon the strength of the illumination and the clarity of the circulating water. The depth of vision when using an ordinary white light even of high candlepower is quite limited. It is therefore necessary to devise some form of illumination that will permit observation through the depth of the water box and so arranged as to cause practically no change in the flow within the water box. It is possible that a light other than white may produce the desired results. The installation of mercury or neon gas tubes may accomplish this. Experiments along these lines should be tried. The hydrogen-ion concentration of the circulating water may best be determined by the use of a hydrogen-ion comparator.

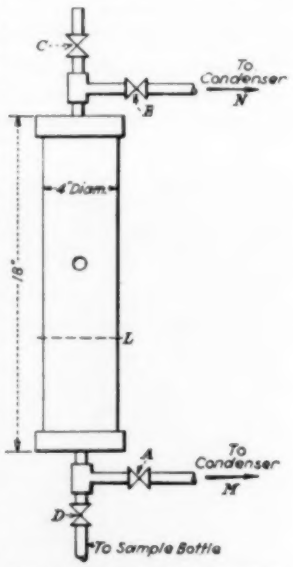


Fig. 8 Pressure-Equalizing Device

(With *N-M* connections made valves *C-D* are closed and *A-B* opened. Water is allowed to circulate through the container *O* for several minutes and then valves *A-B* are closed.

To draw off a sample for determining the O_2 content valve *C* is opened to break the vacuum, then valve *D* is opened to allow the water to overflow a sample bottle until the level in *O* has dropped about 1/3 of its depth [*L* in sketch], then the bottle is corked.)

In carrying out the various alterations necessary to produce a change in the condition of the circulating water or its flow it should be definitely understood that only one alteration on a particular installation may be made at one time and its results recorded. If more than one alteration at a time is made the cause of any change in results cannot be definitely established.

Air Quantity Data for A.S.M.E. Condenser Tube Research Sub-Committee

Company..... Date.....
Plant..... Condenser no.....

	1 Pressure in. hg. + or −	2 Dissolved O_2 cc./liter	3 Circulating water temp. ° F.
Pump suction			
Pump discharge			
Inlet first pass			
Inlet second pass			
Exit second pass			
Entrained air (atm. pr.) entering with c.w. at pump discharge		cu. ft. per min.	
Suction head		ft. water	
Condenser vacuum (in steam space)		in. hg.	
Quantity circ. water		g.p.m.	
Circ. pump speed		r.p.m.	
Circ. pump power		b.hp.	
Remarks:			

Increased Kilowatt Output of Adjustable-Blade Propeller Turbines

By C. R. MARTIN,[1] MILWAUKEE, WIS.

The commercial development of the adjustable-blade propeller-type hydraulic-turbine runner has been in a slow and careful manner, with the idea first of solving its mechanical difficulties and later of taking full advantage of the possibilities of high efficiency. The principal problem of the engineer in introducing a new design is to keep to a safe and reasonable program of development such as will not incur serious financial losses. In addition to the mechanical difficulties of the adjustable runner, there also have been many problems of regulation to be solved as well as the determination of a correct design of draft tube. This paper covers the gradual development and placing in service of the adjustable-blade runner in the United States to the time when the European design of automatic adjustable-blade runner was brought to this country.

The objectionable feature of the fixed-blade type of propeller runner is the peaked efficiency curve and the low efficiency at part-gate openings. By adjusting the runner blades one can operate at all the peak points of efficiency, reducing the disadvantage of part-gate operation. It is now possible to obtain a peak efficiency on a propeller-type runner that is within 1 to 2 per cent of the peak efficiency of Francis-turbine runners. The blades of a Francis runner are not adjustable, resulting in one-peak efficiency. The adjustable runner, therefore, with the possibility of adjusting its peak over a considerable range in load, makes it possible to obtain an average plant efficiency as high as that of the Francis runner.

THE adjustable-blade propeller-type hydraulic-turbine runner has been generally recognized during the last year as representing a marked advance in hydraulic-turbine design. The commercial application of the adjustable-blade runner dates back for several years, both in this country and in Europe. However, it is only within comparatively recent years that more than one manufacturer in the United States was in a position to enter into contract for this type of equipment. The development has been slowly and carefully made in order to determine the best design and method of operation for obtaining the maximum kilowatt-hour output with the adjustable-blade propeller-type runner.

Early in the history of the propeller runner, Forrest Nagler[2] determined many of the characteristics of adjustable-blade propeller runners. This work was accomplished by building runners with different tilts of blades which gave the essential hydraulic characteristics of a runner with flat and steep blade settings. The difficulties in building an adjustable-blade runner of sufficient strength for commercial operation were considered too great at that time, and nothing further was done toward solving the problem until 1917. At that time J. F. Roberts[3] produced a design of an adjustable-blade propeller-type runner which was considered acceptable for manufacturing purposes. The World War intervened, and the work of commercial application was delayed until 1920.

The principle of vane adjustment has been appreciated for many years, as is evident by pump impellers used by the Sanitary District of Chicago at the Thirty-ninth Street Station and installed in 1904. Fig. 1 shows a sketch of the construction used on these propellers. The vanes were made with a flange with slotted bolt holes so that the tilt could be increased or decreased to vary the quantity of water pumped and the head against which the pump operated. It is interesting to note that screw pumps made with vanes similar to those employed in the propeller-type-runner turbine were installed as early as 1888.

In 1921 and 1922 there were a large number of installations made with propeller-type runners. Among these was the Green Island Plant, purchased by the Ford Motor Company for an installation at Troy, N. Y. Fig. 2 shows the runners as furnished with these turbines. It is to be noticed that the runner blade and hub are made in four parts, and so far as is known to the author, this is the first installation in the United States of a separate-blade axial-flow turbine runner.

Shortly afterward the Northern New York Utilities Company purchased for the Sewell's Island Plant two adjustable-blade-runner turbines which required that the unit be shut down and the flume unwatered in order to make an adjustment of the runner vanes. A sketch is shown, Fig. 3, which will illustrate the construction used on these runners. In 1924 the Inca Mining Company purchased an adjustable-blade runner which had blades bolted to the hub. This was a comparatively small runner, and slots were left in the runner flange for bolting the blades to the hub and for adjusting the angle of the runner blade.

The first commercial application in America of the adjustable-blade propeller-type runner where the blades could be moved from the turbine pit, and without requiring that the flume be unwatered, was built in 1925. The adjustment was made by means of the Henderson arrangement, U. S. Patent No. 1699499, which consisted of an adjusting mechanism located between the turbine and generator-shaft flanges. The design included a method for locking each runner blade securely in the runner hub to insure that the runner blade could not rock back and forth in the hub due to changeable vacuum conditions in the high-efficiency draft tubes.

The foregoing gives a brief history of the adjustable-blade runner in the United States prior to the advent of the automatic adjustable-blade runner. Each successive step in the development has been made with the idea of retaining the element of rigidity. The success of the runner has been demonstrated, and is being proved by every-day operation. The question of how far the adjustable feature can be used in solving present-day requirements and the various devices employed in its adaptation will be the interesting work of the future. It is logical that more trouble will be experienced with pitting of propeller-type runner

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[2] Hydraulic Engineer, Allis-Chalmers Manufacturing Company, Life Mem. A.S.M.E., who presented a paper before the A.S.M.E., December, 1919, on "A New Type of Hydraulic-Turbine Runner," describing the development of a fixed-blade propeller-type runner.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[3] Hydraulic Engineer, Power Corporation of Canada, Montreal, Quebec. Mem. A.S.M.E.

blades[4] than has resulted with various types of Francis runners. Five years' experience involving 32 units of capacities varying from 400 to 12,000 hp. and for heads from 8 to 33 ft. shows no serious pitting difficulties with adjustable-vane runners. There have been no mechanical difficulties in connection with the adjusting mechanisms. However, as the vanes are automatically adjusted the likelihood of pitting will probably increase, and greater expense should be incurred in protecting the runner blades against this action of the water.

The characteristic of the higher-speed fixed-blade propeller-type runner (N_S about 150) is such that the shape of the efficiency curve is more peaked than with any other type of hydraulic-turbine runner. A group of curves, Fig. 4, illustrates how the shape changes from a comparatively flat to a much steeper curve for a range of specific speed from the impulse wheel at 4½ specific speed to the propeller wheel with its specific speed of 200 or more. It was this low efficiency at part-gate position compared to the higher efficiency of the Francis runner which led to the early experiment of Mr. Nagler, and it was found at that time that the solution hydraulically lay in the adjustable-blade runner.

The first Holyoke check of the adjustable-blade runner was made in May, 1923, Holyoke test Nos. 2904–6–7–8–9. Results of this test are shown on the curve sheet, Fig. 5, and represent the horsepowers and efficiencies plotted for a unit speed of 340 r.p.m. The horsepower figures have been reduced to represent the power that could be obtained on a 10-in.-diameter runner under 1-ft. head. This curve is intended as an illustration only of the results that can be obtained by adjusting the blades of a propeller-type runner and is not intended to show present-day runner characteristics. Each of the five curves represents an efficiency test with the blades set in a fixed position, at the unit speed of 340 r.p.m. The dotted line drawn across the top of this curve represents an envelope of maximum-efficiency points that could be obtained by an infinite number of runner-blade adjustments. Reading from left to right, the five curves show an increased angle of runner-blade tilt. The first curve represents the runner blade set in almost a horizontal position. The other curves represent a tilt of from 5 to 7½ deg. each from the horizontal, or additional tilt below the flat setting. The total angularity of movement as covered by the curves was approximately 22½ deg. On this runner an efficiency of 88½ per cent was obtained at the best point and at a horsepower of 0.168. Assuming full load as the maximum point of efficiency under the maximum runner tilt, then the full-load horsepower of this runner is 0.26, with an efficiency of 82 per cent. With this horsepower rating of 0.26 the efficiency at half load was 86 per cent. The difference in efficiencies between the various curves and the dotted line, which is the envelope curve, shows the gain in efficiency that can be obtained with the adjustable-blade runner. If this design of runner was used with runner blades cast integral with the hub and set in a fixed position, the maximum horsepower rating for best efficiency would be limited to 0.18 hp. One of the striking features of the adjustable-blade runner is that one of the same diameter as a fixed-blade runner can be rated approximately one-third greater in horsepower. Generally speaking, the efficiency of a fixed-blade propeller-type runner is 65 per cent at half-load. That of an adjustable-blade-runner at half-load is 86 per cent.

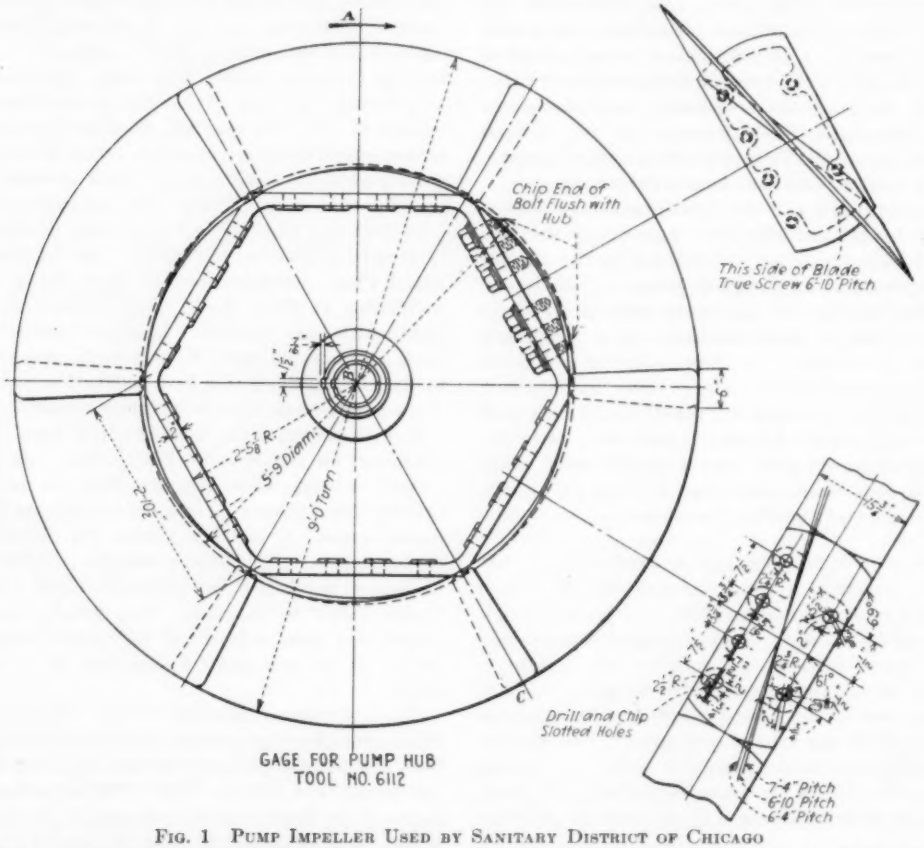


FIG. 1 PUMP IMPELLER USED BY SANITARY DISTRICT OF CHICAGO
(This impeller was used with an axial-flow type of pump operating against an effective head of 7 to 11 ft. and delivering 600 to 700 c.f.s.)

[4] See Trans. A.S.M.E., 1929, Paper HYD-51-9, "Permissible Suction Head of High-Speed Propeller Turbines," by Arnold Pfau.

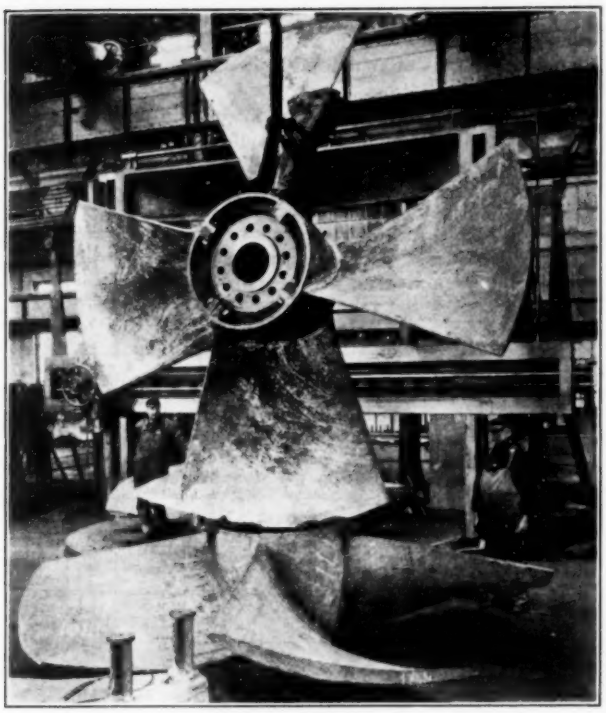


FIG. 2 FIXED-BLADE PROPELLER-TYPE TURBINE

(Each blade is made as a separate part. This turbine is used with a single-runner vertical-shaft hydraulic turbine to develop 2000 hp. under 13 ft. effective head, at a speed of 80 r.p.m. The runner diameter was 156 in. and the assembled weight 17,000 lb. The runner was made of cast steel.)

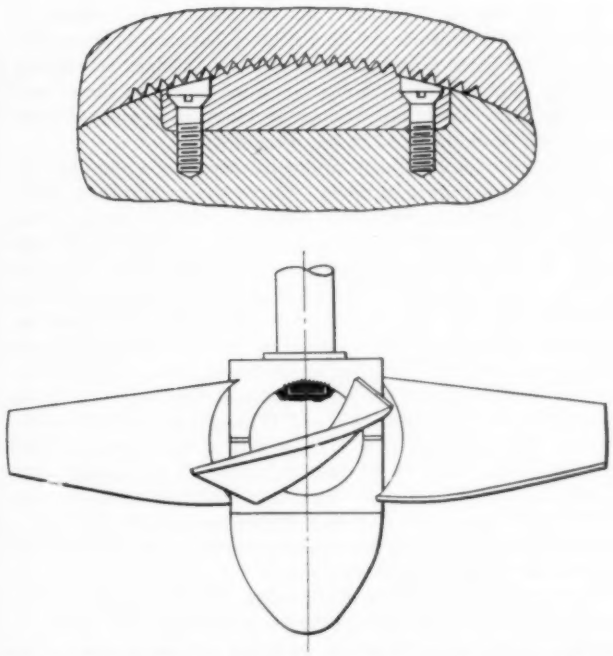


FIG. 3 EARLY DESIGN OF ADJUSTABLE-BLADE PROPELLER-TYPE RUNNER

(The runner hub is divided at the horizontal center line, and the adjustment is accomplished by dropping the lower part of the hub, together with the vanes. Each vane is tilted by an amount of one or more teeth in the adjusting block. At the time when this runner was built the problem of smooth operation had not been completely solved, and large pulsations in draft tubes were quite common. It was thought at that time that the runner blade must be securely locked in the hub; later experience has shown that this feature requires very careful consideration.)

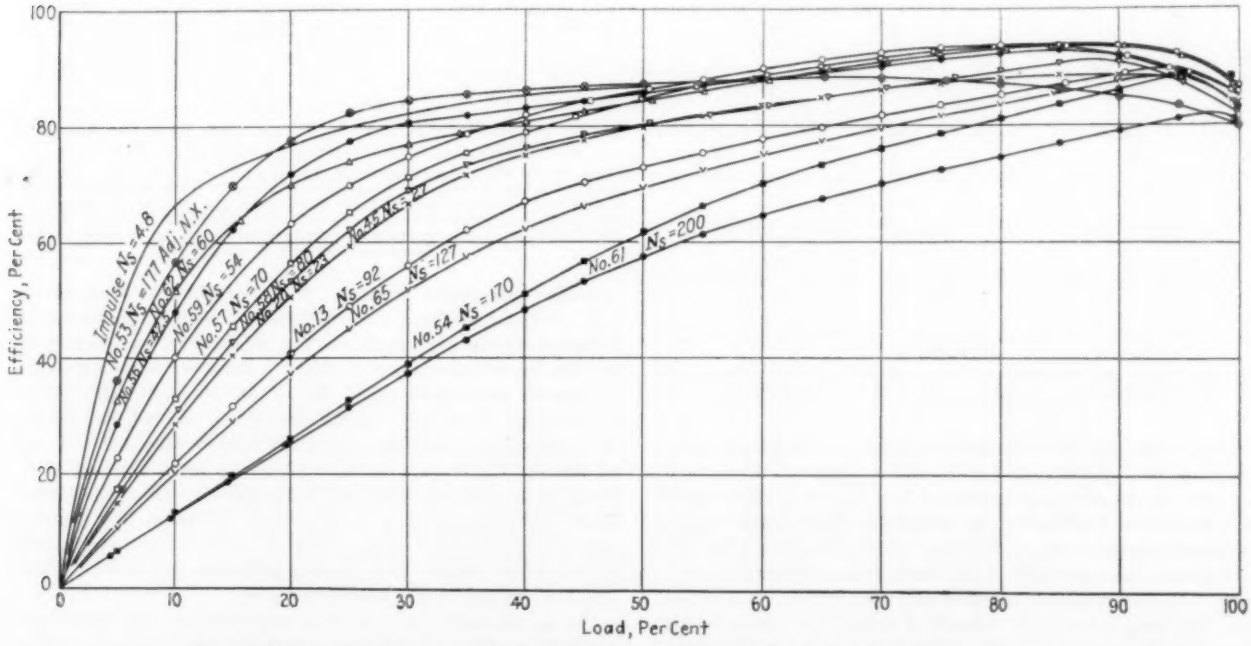


FIG. 4 EFFICIENCY CURVES FOR HYDRAULIC-TURBINE RUNNERS OF DIFFERENT SPECIFIC SPEEDS

(The peaked efficiency curve of the fixed-blade propeller-type turbine falls in line with the efficiency curves of lower-specific-speed hydraulic-turbine runners. The impulse wheel has the flattest type of efficiency curve, with a specific speed of approximately 4½ for best efficiency. At an increase of the specific speed on the Francis type of runner, the efficiency curves become less flat. A corresponding decrease in the flatness of the efficiency curves is found from the Francis runner to the fixed-blade-propeller type of runner; however, the adjustable-blade propeller runner gives a curve which approximates the flatness of the impulse wheel.)

In order best to show the value of the increased efficiency, Figs. 6, 7, and 8 illustrate the efficiencies of four fixed-blade-runner turbines and three adjustable-blade-runner turbines consuming the same total water flow under the same head and the output from the four fixed-blade-runner turbines as compared with the three adjustable-blade-runner machines.

Fig. 7 shows the combined turbine efficiencies of a plant having four fixed-blade-runner turbines. Curve 1 is an efficiency curve of one machine operating alone; curve 2 is the efficiency of two machines, each equally loaded at all times; curve 3 is the efficiency of three machines equally loaded; and curve 4 of four machines equally loaded. The efficiencies are all plotted against brake horsepower. To this sheet have been added the discharge and head curves, based on the recorded flow and headwater variation of the Des Moines River at Ottumwa, Iowa. In addition to the low efficiencies at part-gate openings, this curve also shows that, with a flow between 1150 and 1375 second-feet, there is no gain in output to the plant. Also with a flow between 2200 and 2350 c.f.s., there is no gain in output of the plant. The fixed-blade-runner turbines have therefore two inherent disadvantages—namely, that of low efficiencies at part-gate openings and that of not being able advantageously to use all of the low flow of the stream in producing power.

Fig. 8 represents a series of efficiency, effective-head, and discharge curves plotted against brake horsepower for three adjustable-blade-runner turbines, utilizing the same maximum quantity of water as could be used with the four fixed-blade-runner turbines. The horsepower developed with the three adjustable-blade machines is greater than with the four fixed-blade-runner turbines due to the higher full-load efficiency of the adjustable-runner units. It is to be specially noticed that the

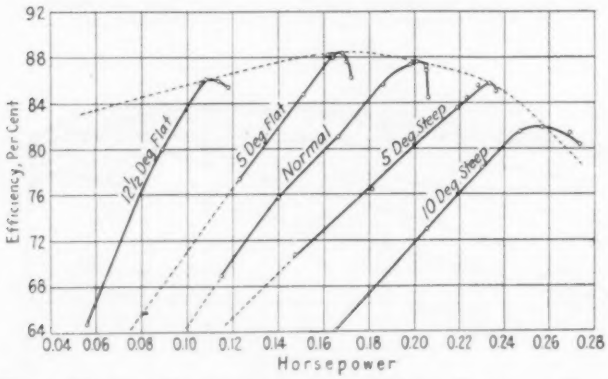


FIG. 5 HOLYOKE TEST OF AN ADJUSTABLE-BLADE PROPELLER-TYPE RUNNER

(The horsepower and speed characteristics of this runner as shown on the curve have been reduced to the equivalent of a 10-in.-diameter runner under 1 ft. head.)

discharge curve for the adjustable-blade-runner machine is almost a continuous uniform curve without any gaps. This is to be expected, as the efficiency curves all overlap, or in other words, the efficiency at half-load on an adjustable-blade-runner turbine is somewhat greater than at full load, which is in accordance with the inherent characteristics of adjustable-blade runners.

Fig. 6 shows a standard duration and head curve. In addition, there has been added the theoretical horsepower output; also the output from the three adjustable-blade runners as determined from the characteristics as shown in Fig. 8 and the output from the four fixed-blade runners as determined from the characteristics as shown in Fig. 7. The gain in output of the three adjustable- over the four fixed-blade-runner machines, as determined by the power-output curves of Fig. 6, is equal to 1,341,232 b.hp-hr.

The actual output of three adjustable-blade-runner turbines equals 22,600,000 b.hp-hr.

The actual output of four fixed-blade-runner turbines equals 21,258,768 b.hp-hr.

This shows a gain of 6.3 per cent increase in output for the three adjustable-blade runners. It should be noticed that in the example given the diameter of each of the four fixed-blade-runner turbines is the same as the diameter for each of the three ad-

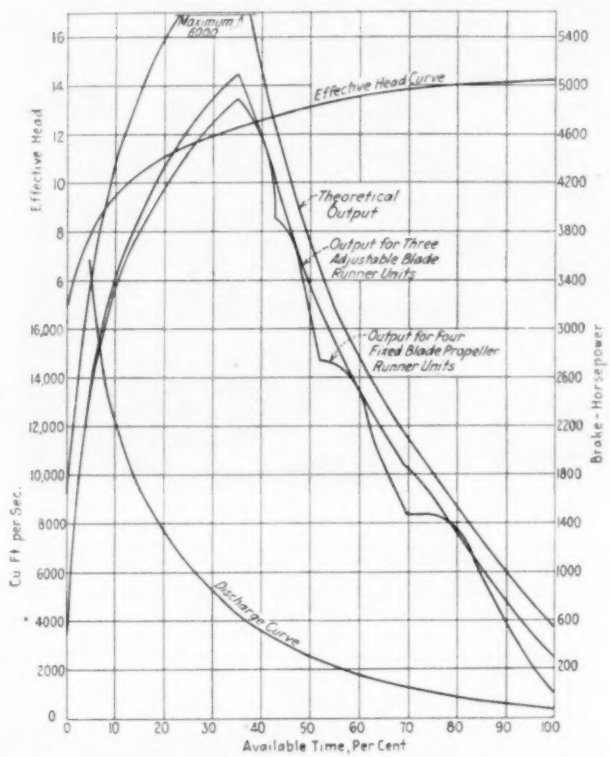


FIG. 6 STANDARD DURATION AND HEAD CURVE, WITH THE ADDITION OF GRAPHICAL REPRESENTATION OF THEORETICAL OUTPUT

(This shows the output for three adjustable-blade-runner units and the output for four fixed-blade propeller-runner units. The increased brake-horsepower output of the adjustable-blade-runner units over the output for four fixed-blade-propeller-runner units is 1,341,232 b.hp-hr., or approximately 950,000 kw-hr. This increase represents a gain of 6³/₁₀ per cent for the adjustable-blade-runner units.)

justable-blade-runner turbines. If draft tubes were built on the basis of linear proportion to the runner diameter, the width of flumes would be the same for the adjustable-blade-runner as for the fixed-blade-runner turbines. However, it is necessary to furnish somewhat wider flumes for the adjustable-blade-runner draft tubes than for the fixed-blade turbine, due to the fact that unless sufficient draft-tube area is provided with an adjustable-blade-runner turbine, a point is reached where additional water discharged through the turbine by tilting the runner blades downward creates such a large discharge loss that no gain in power results. This brings up a very important feature in draft-tube design—namely, that sufficient area must be provided to eliminate high backwater losses.

During the past five or six years considerable data have been obtained in the use of the adjustable-blade runner. Some of these experiences illustrate the advantages of that type of runner, and some features which should be given further consideration.

One of the early plants consisted of three 1200-hp. units, under 9 ft. head, which replaced a number of vertical turbines, gear-connected to a horizontal generator. The output from the

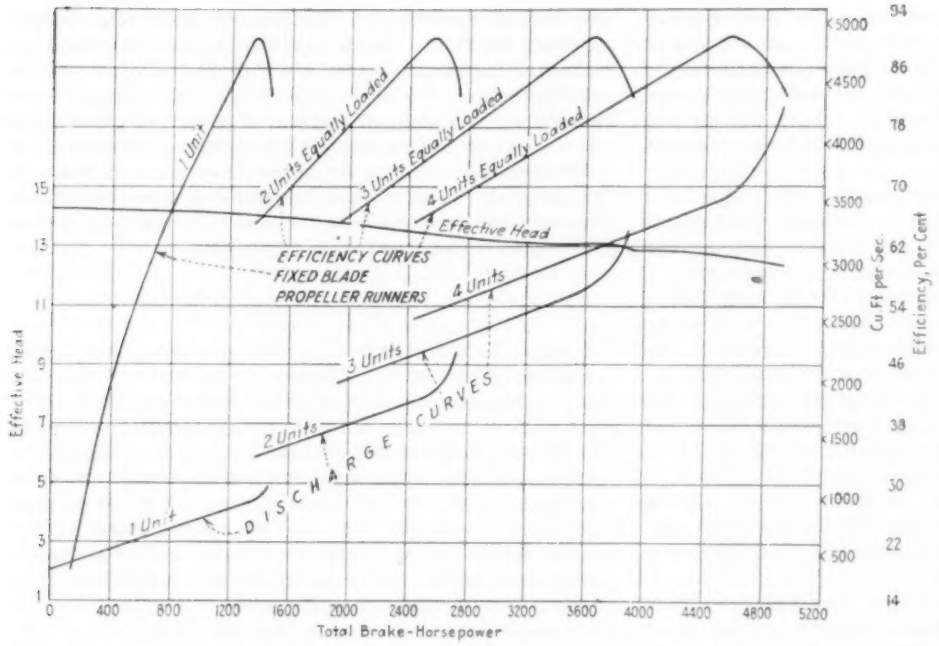


FIG. 7 EFFICIENCY, DISCHARGE, AND EFFECTIVE-HEAD CURVES PLOTTED AGAINST TOTAL BRAKE-HORSEPOWER

(This shows the efficiency of a fixed-blade propeller-type-runner plant for all conditions of discharge up to its full capacity. Notice the gaps between the discharge curves for one and two and for two and three units. These gaps represent portions of the flow which cannot be utilized by the fixed-blade-propeller runners.)

such as occur during heavy floods, a point finally is reached beyond which the power output of the turbine decreases much faster than the ratio of the three-halves power of the heads. In other words, for a runner with a given blade tilt, as the head decreases the unit speed of that runner increases until runaway speed is reached, at which speed the runner no longer produces power. If the blade tilt of the runner is flattened, the runaway speed is increased; in other words, the head can be reduced still more before the unit speed of the runner reaches the runaway speed. It would be desirable to design the runner for opposite conditions so that increasing the runner tilt would increase the runaway speed. Unfortunately, however, this type of runner does not lend itself to such a feature of design. In the adjustable-blade-runner design this inherent disadvantage, during high backwater and resulting low-head conditions, can be offset by flattening the runner-blade tilt and thus improving the runner-speed characteristics.

vertical-gear-connected units averaged 8,000,000 kw-hr. The output from the adjustable-blade-runner plant, which utilized the same water and which was set in the same overall width of power house, has exceeded 12,000,000 kw-hr. per year. The increased output has been due both to the greater installed capacity and higher overall efficiency. The plant in question receives its water from a large lake with regulated discharge, and the runner blades are set, generally speaking, on Monday morning for a definite flow, and they operate for the week under this flow condition. There is practically no change in runner-blade setting during the week, except to regulate for a sudden heavy rainfall or some other requirement of lake-level regulation. One year later four 1100-hp. units, under 8½ ft. head, were installed on the same river. This later plant was an entirely new development, but was limited as to width of power house. However, the same condition of fixed flow prevailed throughout the week. During the spring floods they are troubled with excessive backwater, and it was found that during these periods the output of the plant under a head of 3 and 4 ft. was increased materially by flattening the runner blades from the full-tilt position.

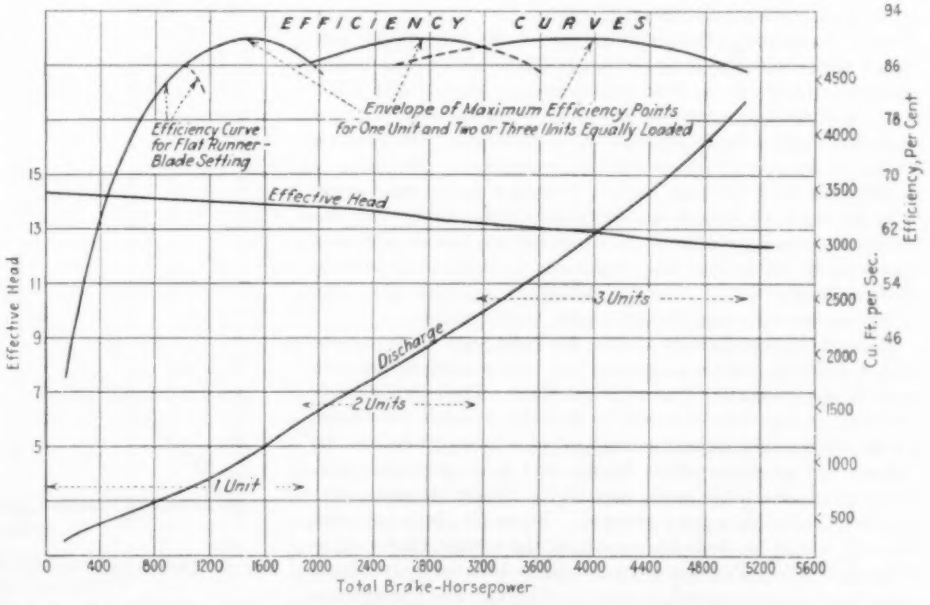


FIG. 8 EFFICIENCY, DISCHARGE, AND HEAD CURVES PLOTTED AGAINST TOTAL HORSEPOWER FOR THREE ADJUSTABLE-BLADE-RUNNER TURBINES

(These turbines are able to utilize over 90 per cent of the flow within the plant capacity, with an average efficiency of approximately 88½ per cent.)

This is due to the fact that with extreme head reductions

Another example of an adjustable-blade-runner installation is a plant consisting of three 3750-hp. turbines under 28-ft. head. This plant operates as a run-of-river plant, using the water as it comes down the river. Immediately downstream from this plant are three high-specific-speed Francis-runner turbines which

were installed at essentially the same time. In checking plant records it is found that during low-flow periods a greater output is obtained per foot of head on the adjustable-blade-runner installation than can be obtained from the high-specific-speed Francis-runner installation. These plants clearly demonstrate the advantage of the increased efficiency that can be obtained at the flat runner-blade positions.

A plant consisting of six 5000-hp. units, at 30 ft. head, was equipped with adjustable-blade-runner turbines. This plant, after operating for a period of more than one year, finds that the adjustable-blade runner gives them the following advantages:

1 The additional capacity that can be obtained by tilting the runner blades enables them to keep one unit as a spare. This is very important, as they are very apt to have some difficulty with the electrical apparatus during storm conditions.

2 This plant operates from a storage reservoir and the load is fixed from day to day, the load being changed by varying the number of units except during the high-flood periods. During the night, or from 10 p.m. to 6 a.m., it is not desirable to carry a great amount of load on the hydro system. During this period one of the units has its runner blades set to a flat position, and only sufficient power is delivered to the system as is required for regulation purposes.

3 The increased capacity which can be obtained by more steeply tilting the runner blades increases their peak-load output from a normal operating load of 3000 kw. per unit to a maximum of 4000 kw. The capacity of the plant normally is from 16,000 to 18,000 kw. This capacity has been increased to 23,000 kw. under flood-water condition, and when called upon to carry maximum peak load under full operating head, the peak capacity could be increased to 24,000 or 25,000 kw.

In a recent plant the requirements of the adjustable-blade runner were limited to using the runner-blade tilt for maintaining constant capacity as the head was decreased due to floods. The capacity of the generator was limited to the capacity of the wheel at most efficient blade setting. It is questioned whether this saving in generator cost is justified, considering the value of generator capacity for peak loads. There is a case in mind where it is proposed to furnish an adjustable-blade-runner unit for 2000 kw. capacity, under 21 ft. head, but to provide sufficient runner-blade tilt so that this output can be maintained with an effective head of 15 ft. This is a rather extreme case, but it can be met successfully with the adjustable-blade runner.

Most of the installations that so far have been made operate with a controlled water supply, or the load conditions are such that the blades can be adjusted at any time. The plants already discussed require that the unit be shut down before the blades are adjusted. The adjusting mechanism is located between the turbine and generator shaft flanges and is so arranged that, after the adjustment is made, each of the runner blades is automatically locked in a fixed position. No doubt there are cases where it would be desirable to adjust the runner blades automatically without shutting down the unit. Isolated hydro plants must be adapted to control the speed. This involves a sacrifice in efficiency which is unavoidable due to the variation of the output. Here regulation is of greater importance than the maintenance of highest overall efficiency. By far the greater number of low-head plants in this country are a part of a power system which has additional plants, hydro or steam, operating in parallel, to take care of the fluctuating load. The low-head plant usually can be operated from the point of view of obtaining highest overall efficiency, thereby justifying such refinements in the interrelation of guide-vane opening and runner tilt as will produce the best economy for each respective operating head. More and more of the streams are being regulated, and as time goes on, the point of complete regulation will no doubt be reached on the principal power streams. The disadvantage of automatically regulating the runner vanes, especially in connection with the guide-vane movement, is that it is difficult to lock the blades after the change. The period is just being entered of adjustable-blade-runner use, and new features of design will open up as this new type of runner is employed in more actual developments.

The hydraulic utility of the adjustable-vane runner has been demonstrated. However, the unwarranted number of fixed-blade-runner failures points to the need of even greater caution in extending the adjustable-vane-runner field.

Discussion

CARROLL F. MERRIAM.[5] The author presented some interesting curves indicating the comparative operating efficiency of plants having all fixed-blade propeller turbines and all movable-blade propeller turbines; but it would be interesting to consider the effect of combining these two types of units in a single plant. To illustrate the promising possibilities in this respect, the writer has assumed as an example a broad efficiency curve that might be expected for a movable-blade unit, such as the author has described, and an efficiency curve that might be expected for one having fixed blades. In order to simplify comparison, these curves have been drawn to have the same maximum efficiency and the same output and discharge at maximum load.

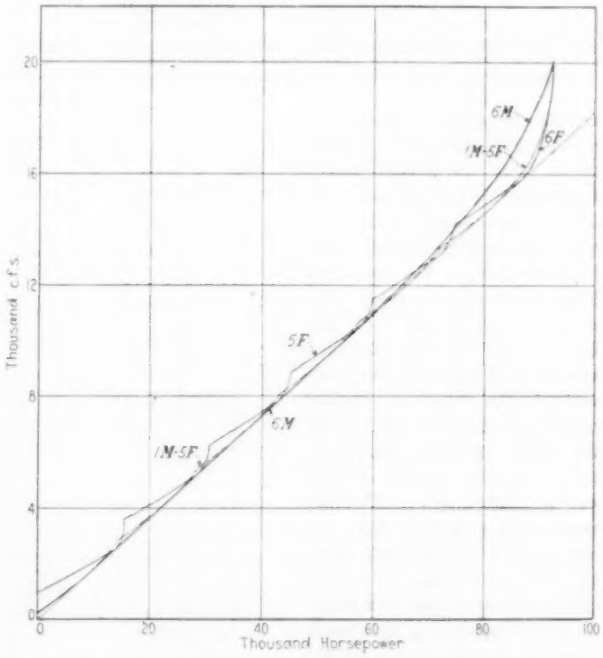


FIG. 9 DISCHARGE VERSUS HORSEPOWER—NO ALLOWANCE FOR RUNNING SPARE CAPACITY

On the accompanying diagram, Fig. 9, there have been plotted curves marked 6-*F* and 6-*M*, showing discharge versus horsepower, for plants having six units each of both types of wheels. These curves correspond to those which are shown by the author in Figs. 7 and 8. There has been added a similar curve marked 1*M*-5*F*, for a plant having one adjustable and five fixed-blade turbines.

It has been pointed out by the author that the objectionable feature of having all fixed-blade units is the presence of inefficient

[5] Pennsylvania Water & Power Company, Baltimore, Md. Mem. A.S.M.E.

zones or gaps between the curves for successive groups of units. These zones are almost entirely closed by the use of movable-blade units. It should be noticed, however, that the substitution of one movable-blade unit for one having fixed blades eliminates the greatest portion of these areas, and also that successive substitution would have little further effect. The fixed-blade units can secure better efficiency than the movable when the available water is nearly equal to the full capacity of the plant. The substitution of one movable-blade turbine may somewhat reduce this advantage, but by no means in proportion to the gain from increased efficiency at lighter total station loads.

One distinct advantage in the fixed-blade propeller turbine, as compared with the movable blades, is the lower first cost. For this reason, it is very desirable to design hydro power plants in which either type might be used, to secure the maximum benefits which are commensurate with the additional cost of movable-blade over fixed-blade units.

Another important element to be considered is the amount of running spare capacity required for system protection, which frequently makes it necessary to add another unit, even though the actual load could be carried more efficiently without the additional machine. When it is necessary to make liberal allowance for heavy swings or for sudden failure of the largest machine on the system, it may be found economical to select more units of the movable type. Fig. 10 has been drawn up to show the

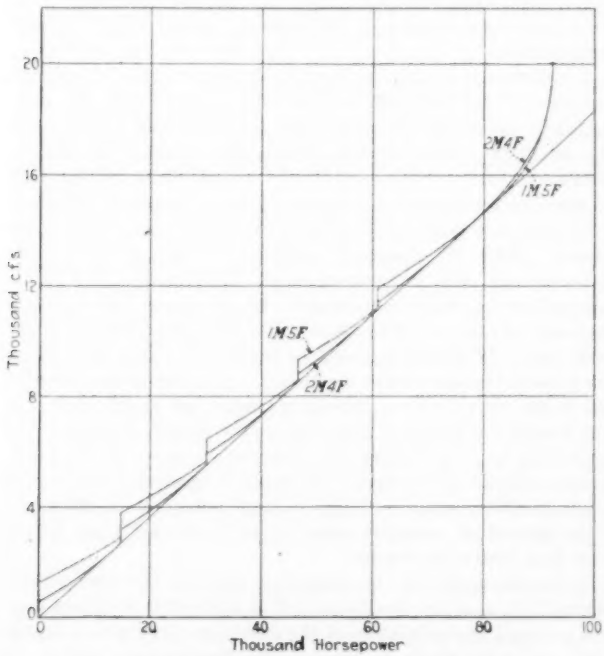


FIG. 10 DISCHARGE VERSUS HORSEPOWER—11,000-HP. ALLOWANCE FOR RUNNING SPARE CAPACITY

comparison between discharge versus output curves for two six-unit plants with one and two movable-blade units, respectively. It will be seen that in the case of the former there still remain large inefficient gaps which can for the most part be closed by a second movable-blade machine.

ROBERT W. ANGUS.[6] This valuable paper calls attention to the use of the propeller-type turbine with adjustable blades as distinguished from the corresponding turbine with fixed blades. If the blades of the propeller-type runner are adjusted to suit the load or head, or both, very much flatter efficiency curves are obtained than are possible with the fixed-blade propeller units; and the author shows in Fig. 4 that under lower loads the efficiency of the adjustable unit is better than that of Francis units, although these types change places at full load.

The author appears to have based his conclusions in the paper on the turbine always running at its best efficiency, which implies that the blade adjustment is always made to suit the power and head available. Such a condition is only possible, however, if the blades are automatically adjustable during the operation of the turbine and are therefore adjusted by the governor and not by hand. It is evident that if the machine has to be shut down to make the adjustment, or even if the adjustment must be made by hand while the machine is running, the results shown in the paper will not be correct, because under such conditions the machine will not always run at best adjustment. Hand adjustment is very valuable in looking after changes of head due to floods, etc., as the adjustment need only be made at intervals.

The automatically adjustable propeller turbine, or Kaplan turbine as it is commonly called, has been very slow in making headway in this country and, so far as the writer knows, only one manufacturer now has the rights to make this type of turbine in America. On the other hand, when the writer made a very extensive trip to the hydraulic power plants in Europe last year he found many plants using the Kaplan turbine, and the number of such turbines is relatively increasing, although the earliest units have not been in use for many years. Since different servomotors are in use on the distributor and runner guides, the relation between the movements of the two sets of guides may be altered if it should be necessary to meet requirements of varying heads, speeds, etc., and usually the adjustment of the runner blades is much slower than the distributor guides.

An inspection of the mechanism necessary to operate the runner blades indicates that it is not altogether simple, and the Kaplan turbine is more complicated than the simple propeller turbine. For power plants with fixed heads and for which the propeller type of turbine is suitable it would therefore be advisable to have some of the units with fixed runner blades and to operate these continuously at the load corresponding to best efficiency, while some Kaplan turbines could be installed to take up the variations in load. Such a plan would give the simplest units, while at the same time producing maximum efficiency and long life of the turbines. If the head has large variations, naturally all but the Kaplan turbines would have hand-adjustable blades.

One of the most interesting plants now being built for a low head is that at Ryburg-Schworstadt on the Rhine River a short distance east of Basel. This plant connects the German and Swiss frontiers, and the fact that the two countries were involved probably accounts for some novel features connected with it. The weir is on the Swiss side and the power house and transformer station are on the German side of the river, and the designs are for four vertical turbines of 35,000 hp. each, although the output falls to about one-half of this for the lower heads.

Since the units are of unusually large size, considering the head of 35 ft., requiring about 10,000 c.f.s. each, there was a special engineering organization entered into for the construction of the turbines, three of the largest turbine builders in Europe forming a joint company for the purpose. All of the units are to be vertical Kaplan turbines with a runner diameter of nearly 23 ft., so that when completed the plant will be of unusual interest, and the results from it will be of great value in connection with this type of turbine. The information available shows that the draft-tube diameter at the top is 23 ft., and hence the vertical

[6] Professor of Mechanical Engineering, University of Toronto, Toronto, Ont., Canada. Mem. A.S.M.E.

component of the velocity at full load is slightly over 24 ft. per sec., the corresponding velocity head being 9 ft., or 25.7 per cent of the head of 35 ft. It is interesting to note that quarter-turn draft tubes are being used and that the runners are set at approximately normal tailwater level.

The paper is of real value in laying stress on a type of turbine which is essential to the success of many low-head plants.

ARNOLD PFAU.[7] The author, in the closing paragraphs of his able paper, points out that automatic adjustment of the runner blades would be desirable in certain cases, particularly with isolated plants which must be adapted to control the speeds.

He points out that this involves a sacrifice of efficiency. May the writer be permitted to dwell a little on this particular point? Where speed control is of importance, efficiency must take a secondary place. Vice versa, where maximum kilowatt output is the prime requirement, the speed control is of secondary importance. Thus it becomes evident that, in the first case, automatic control of guide vanes by the governor for quick adjustment of the output in accordance with the load change and of runner blades for maintaining as high as possible an efficiency should be adopted, whereas the second case justifies a refinement in the interrelation of guide-vane opening and runner tilt, for the purpose of obtaining highest economy for each respective operating head.

For a fixed head the law of interrelation between guide-vane opening and runner-blade tilt is a fixed one for obtaining the envelope of maximum efficiencies. This law, however, varies as the head varies, unless the speed can be changed accordingly, which, however, is commercially impossible. It is therefore evident that the fixed interrelation obtained with governor control cannot result in highest economy under variable heads, whereas an individual adjustment of the runner blades permits of such a refinement, so that a substantial increase in kilowatt output can be obtained during each extended period of existing operating head.

GEORGE A. JESSOP[8] AND BEAUCHAMP A. SMITH.[9] The author has called attention to what, at the present time, is the most interesting and important development in the hydraulic-turbine field.

The first known instance of the attempt to use an adjustable-blade turbine is the Ludlow United States patent No. 67,994, granted in 1867. In this design the blades only were adjustable. There were no water-control gates on the turbine, but stationary guide vanes were used.

About the time the Ludlow design was brought out, there were also many patents issued showing axial flow runners with adjustable gates of the cylinder, wicket, and other types. Apparently the advantage to be obtained by adjusting the buckets and the gates simultaneously to maintain a predetermined relationship was not realized, and no attempt was made to design or build such turbines. The first application of this principle was made by Dr. Victor Kaplan, of Brünn, Czechoslovakia, who entered patent applications in Europe in about 1913 and in this country in 1914. Due to the disturbance in business conditions caused by the war, no attempt apparently was made either in Europe or in this country to make use of the Kaplan designs in a practical way until about 1920. Shortly after this date, a contract was entered into by an European builder for two automatically adjustable-blade units operating under 20-ft. head and developing 100 hp. each. This installation was rapidly followed by others of various capacities and operating under various heads. One European builder alone has furnished 85 turbines of this type developing a total of 187,400 hp. under heads of from 7 to 48 ft. and in sizes up to 40,000 hp. During this same period the same manufacturer has built only 19 fixed-blade axial-flow turbines with a total of 79,000 hp. At the present time European practice is to install automatically adjustable-blade turbines in practically every instance where the head conditions are suitable.

In 1927 exclusive rights under the Kaplan United States patents were secured by an American manufacturer, who immediately proceeded with the commercial development. The first automatically adjustable-blade runner was sold in 1928 and was rapidly followed by other contracts. Since that time 16 adjustable-blade units operating under heads of from 10 to 40 ft. and developing a total of 37,366 hp. have been sold. Of this number only one, and this the first one, has been of the manually adjustable type. It is significant that this last-mentioned turbine has been the means of securing an order for three automatically adjustable-blade turbines for the same installation. It is interesting to note that for this plant, an old one with 20 units now installed, there has been purchased, first, a fixed blade, then a manually adjustable, and now there are on order, as mentioned, three fully automatic adjustable axial-flow turbines.

Sufficient experience has now been had in this country to show that the operation of this type of turbine will be entirely satisfactory.

A study of the author's Fig. 5 will clearly reveal the disadvantages of the manually adjustable compared with the automatically adjustable-blade turbine. As has been pointed out, the envelope shown by the dotted lines shows what can be obtained with an infinite number of runner-blade adjustments. It must be remembered, however, that for maximum efficiency with each gate position there is a definite position of the runner blades. With the manually adjusted turbine, therefore, the envelope can only be approximated. For example, if 0.26 hp. is considered as maximum capacity, let us assume that there are as many as four adjustments made in going from full to 40 per cent load. Of necessity, then, the results will follow the 10-deg. curve until the horsepower is reduced to the maximum shown by the 5-deg. curve, with a consequent large loss in efficiency. If the blades are adjusted when the power output is slightly less than 0.24 hp., the turbine can then be operated on the 5-deg. curve until the maximum power output is reached at the normal blade position, again suffering a large reduction in efficiency. This method of operation must be continued until the 40 per cent load position is reached.

A consideration of the operating features involved clearly shows that, as a practical matter, it is impossible to obtain even the outlined approximation of the envelope curve with a manually adjustable-blade turbine. The turbine can seldom, if ever, be shut down to change the blade position at the exact power output required to obtain maximum economy, thus requiring operation for a longer time on the low efficiency curve. This will cause especially large leases if a margin of power is required for speed regulation.

By using a turbine having automatically adjustable blades which are kept in proper relation to the gates, the results will always follow exactly on the envelope curve shown by the dotted line. The turbine will thus operate with maximum economy without the necessity of shutdowns for adjustment with the consequent expense and reduction in kilowatt-hour output.

Referring to Fig. 6, it may be noticed that the curve showing the output for the three adjustable-blade turbines can be obtained only if the adjustment is fully automatic, and the best

[7] Consulting Engineer, Allis-Chalmers Mfg. Co., Milwaukee, Wis. Mem. A.S.M.E.

[8] Chief Engineer, S. Morgan Smith Company, York, Pa.

[9] Secretary, S. Morgan Smith Company, York, Pa. Jun. A.S.M.E.

that can be done with the manually adjustable turbine is an approximation of the results shown.

A careful comparison shows that the cost of a medium size, fully automatic adjustable turbine is about 6 per cent more than for a properly designed manually operated turbine. This difference in cost increases somewhat for smaller sizes and decreases somewhat for large sizes. In either design, the hub mechanism should be of the same rigidity and strength. The additional equipment required for the fully automatic adjustable type is simple in design and construction and is accessible for inspection and care.

The author has suggested that it is necessary to lock the blades in a fixed position to prevent any possibility of rocking back and forth. Properly designed blades have the operating and supporting stem so placed that the forces on each side of the stem are practically balanced. Just as the governor holds the wicket gates in a fixed position, even against a large unbalanced pressure, so the blades are held in the proper position by the runner servomotor, which is controlled by a distributing valve connected to the gate mechanism and a compensating mechanism between the runner servomotor piston and this valve.

If an adjustable-blade turbine is installed with a larger capacity than would otherwise be used, it follows that the flume must be larger to handle the increased quantity of water without increasing the loss of effective head. The draft tube can always be made to suit the unit spacing required by the flumes. Properly designed elbow draft tubes prevent any possibility of the piling up of water in the tube and thus eliminate all danger of building up back pressure to seriously reduce the effective head on the runner.

The author has suggested that there will be an increased likelihood of pitting with the automatically adjustable-blade turbine.

In our seven years' experience with the fixed-blade turbine not a single case of serious pitting has developed, although the axial-flow turbine has been installed under heads as high as 60 ft.

If the blades are automatically adjusted and kept in the correct relationship with the gates, pitting will be decreased and stability of operation increased, because the gates and buckets are maintained continuously in the proper relationship to produce the best and most efficient conditions of flow. With the manually adjusted runner, the pitting will be reduced to the extent that the flow conditions are improved by the adjustment of the blades.

The fully automatic adjustable turbine has no limitations as to advantageous use only with a controlled water supply nor is it necessary to confine its use to those plants where the operating conditions are such that the unit can be shut down and the blades adjusted at any time.

Operation of modern systems calls for steady loads on the steam plants and variable loads on the hydro plants. The fully automatic adjustable turbine is suitable in every way for such service, whereas the manually adjustable runner has obvious serious disadvantages, if, indeed, it can be used with any economy at all.

It is necessary to state that the S. Morgan Smith Company has exclusive rights under all the United States Kaplan patents. The installations are as follows:

TYPE X-4 RUNNER INSTALLATIONS, ADJUSTABLE BLADES

	Purchaser	Place	No.	Head	Speed	Unit Power
1	Metropolitan Edison Co.	York Haven, Pa.	1	26	200	2970
2	Central Pr. & Lt. Co.	Devils River No. 9	1	33	277	1900
3	Wisconsin Pr. & Lt. Co.	Rockton Develop.	1	10½	100	750
4	Nova Scotia Pr. Comm.	Tusket Falls	3	20	225	1052
5	Utilities Pr. & Lt. Co.	Valentine, Neb.	1	29	600	320
6	Wheatsworth, Inc.	Hamburg, N. J.	1	29	600	310
7	Metropolitan Edison Co.	York Haven, Pa.	3	26	200	2970
8	Connecticut River Dev. Co.	McIndoes, Vt.	2	29	150	5120
9	Utilities Pr. & Lt. Co.	Lebanon, Mo.	2	40	327	2030
10	San Joaquin L. & P. Co.	Merced Falls, Cal.	1	26	128.6	4750

J. W. LINK.[10] In the very interesting paper the author has confined his comments largely to telling the net results of the good points of the adjustable-blade turbine as expressed in kilowatt-hour output, and there is little, if anything, that need be added to this aspect of the case, except to say that the general conclusions seem to be sound.

If a departure from the subject as expressed in the title is permissible, it may be well to bring out some points which, while they may be mentioned or implied in the paper, may not be given the emphasis they deserve.

The propeller-type turbine, but especially the adjustable-blade design, is a distinct contribution to the progress of water-power development. This is particularly important at this time, because of the more or less successful competition of the steam turbine in the field of prime movers.

The propeller-type turbine makes possible the development of some water powers which would otherwise be economically impracticable, but it is not suited for all situations.

The adjustable-blade turbine is peculiarly adapted to very low-head situations, but more especially to those having a large variation in head and water supply, because these machines not only develop more output at part gate, because of better efficiencies, but also because they will continue to turn out power under heads so reduced as to render other types of hydraulic turbines ineffective.

The author has stated that the advantage of the adjustable-blade turbine lies in the increased kilowatt-hour output, combined with lower overall costs. This statement is probably a little too broad, for, while the high specific speed practicable in a propeller-type turbine may tend to reduce the cost of the generator and exciter and in some cases may also tend to reduce the cost of the power house, there are many situations where the dimensions of the power house will be determined by the size of the water passages and there may be no reduction in cost by reason of using the adjustable-blade turbine. If, however, the high specific speed of the propeller turbine should effect a saving in cost of a power development, such saving would probably be produced by the fixed-blade as well as the adjustable-blade turbine.

There are some situations where the generating units can be operated advantageously at fixed loadings, and in such cases a fixed-blade propeller-type turbine would probably give as large an output as the adjustable-blade turbine, and it is probable that the Francis turbine would give a larger output provided the head was practically constant.

However, conditions change, and even if it should be best to operate a plant today under a fixed load, in five or ten years it may be desirable or necessary to operate in some other way; and the great advantage of the adjustable-blade turbine lies in its adaptability to changeable conditions of operation, of head, and of stream flow. In other words, it is a more flexible machine, as regards operating conditions, than any of the other types of hydraulic turbines.

In considering the possible advantages of the adjustable-blade turbine, it should not be forgotten that it has its limitations. In its present state of development it is peculiarly a low-head turbine and probably finds its largest field of usefulness in situations where the head is 30 ft. and under, but especially where there is a large percentage variation of head.

If this machine can be successfully developed for higher heads, it will doubtless have many, if not all, of the advantages that it has for the lower heads, but the maximum range of heads under which it will operate successfully has yet to be determined.

It is also probable that without automatic control of the blade

[10] Hydraulic Engineer, Byllesby Engineering and Management Corporation, Chicago, Ill.

settings the adjustable-blade turbine will be entirely impracticable for use in a plant where full automatic operation is desired.

In plants where it is necessary to vary the load frequently during each 24 hours because of regulating requirements or because of frequent daily fluctuations in water supply, manual adjustment of the turbine blades would be a considerable inconvenience if not an absolute impossibility. There are comparatively few situations where low-head plants would be subjected to such conditions, and if only one or two settings of the blades in 24 hours would be required, manual control would be entirely feasible, because the time required for stopping a machine (if it happens to be loaded when the adjustment is to be made), making the adjustment, starting the machine, and putting it back on the line is only a few minutes.

The author calls attention to progressive stream regulation as favoring the adjustable-blade turbine and voices the optimistic opinion that complete regulation will eventually be a reality on the principal power streams of the country.

This is a consummation devoutly to be wished, but the physical conditions are usually unfavorable to this accomplishment.

However, if complete regulation were accomplished, presumably it would mean constant or nearly constant stream flow and practically uniform head, and such conditions of operation would appear to reduce rather than enhance the advantages of the adjustable-blade turbine.

Author's Closure

Each of the discussions brings out additional features in connection with propeller-type runner turbines. These points can be summarized as follows:

Mr. Merriam's discussion emphasizes the possibility of a combination of adjustable- and fixed-blade runner units in order to effect economies in a power development. From the standpoint of present-day practice this is very probably true, but as we proceed with the development of the adjustable-blade runner it is possible that at some future date we may be able to take full advantage of the adjustable-blade runner and build a plant having all adjustable-blade runners at approximately the same price as could be done with a combination of fixed- and adjustable-blade propeller-type runners.

Under the discussion offered by Mr. Robert W. Angus, Mr. George A. Jessop, and Mr. Beauchamp A. Smith, the advantages of the automatic adjustable-blade runner have been presented as an addition to the paper on the manually adjustable runner. The author's experience has been confined to actual power-plant operating data obtained from the manually adjustable-blade runner. He has, therefore, offered this experience in the hope that it may be of value to the hydraulic engineering profession in the design of power plants involving both manual and automatic adjustable-blade propeller-type turbines.

Mr. Arnold Pfau brings up the very interesting subject of interrelation between guide-vane opening and runner-blade tilt for varying head conditions. This feature of design is the most important point to be considered in connection with the adjustable-blade runner. Generally speaking, each plant that may be built has its own peculiarities of intake- and draft-tube requirements. It is the author's experience that the relation of guide-vane opening to maximum efficiency in the actual plant varies considerably from the relation established in model test flumes. If we are to obtain the peak efficiency on the adjustable-blade runner, we must therefore determine the characteristics of the wheel under the actual power-plant setting.

The discussion of Mr. J. Wm. Link gives an interesting sidelight on the propeller-type turbine from the standpoint of the user. The point of changing conditions under which a hydro plant may be required to operate in future years is certainly a very forceful argument in favor of the adjustable-blade runner.

Changing Requirements in Hydraulic Turbine Speed Regulation

By FORREST NAGLER,[1] MILWAUKEE, WIS.

Flyball control of units such as are now operated in the plants of large systems may be of secondary importance for normal conditions. It is a control for large or sudden changes, a shutdown device for emergency purposes, and it effects certain conveniences in operation. It is of real use only in emergency conditions, it is entirely ineffective when a unit is on the load limit, and it is entirely ineffective when the gate position is controlled by frequency under gradual load change conditions. It comes into play in case of line troubles or when load changes are faster than the frequency control can meet.

Practically complete governing equipment can be built without a large flyball, and with probable ultimate simplification of governing equipment. It would consist of frequency control operating the gate through regulating cylinders and distributing valve, a load limit positioning device for that valve, just as is done nowadays, except that no main-relay connections would be required, no dashpot necessary, and all question of flyball drive eliminated. Emergency stop and emergency start would be obtained simply by making secondary contact at an abnormally high or low frequency point that would take the unit off the line or at least close it to "speed-no-load" condition.

It is not suggested that governors be eliminated. Many of their elements are very satisfactory. The oil-pressure systems now in use are probably the best form of speed-reducing mechanism for the purpose, reducing 600- to 1200-r.p.m. motor speed to as low as 1 r.p.m. It is suggested that they no longer be governors from a normal speed-control standpoint with its possible wide range of gate movement. They should be control mechanisms designed for the duties that they are to accomplish. The design of such controls would rapidly modify governors as they are designed today, and would also begin to modify turbine structures themselves, simplifying and cheapening them, a large portion of their cost being due to details which are not necessary or even desirable for the production of the maximum number of kilowatt-hours for a given stream flow or for a given amount of storage water.

IT IS ADVISABLE in any field of design where conditions change rapidly occasionally to visualize the changing needs and requirements of any particular division of the equipment. Such conditions apparently exist today in connection with speed governors on hydroelectric units. These governors are frequently quite foreign in their action to the main purpose of the plant in which they are installed, being one of the major causes if not the major cause of loss of kilowatt-hours, interruptions to service, damage to parts, and increase of maintenance cost. Conventional governor action is a relic of requirements which no longer exist, as there is no necessity for extensive and continuous gate movement on units of large systems to effect speed control.

The earliest hydraulic unit, probably the earliest prime mover, was connected to a mill stone or water lift. Such a load required no speed control of the water-wheel gates, as the load was of the self-regulating variety to a sufficiently close degree. The same condition existed to some extent in the later mechanical mill drives. When these became electrically driven, a speed control became necessary because of parallel operation and voltage regulation. When various electrically driven mills were first

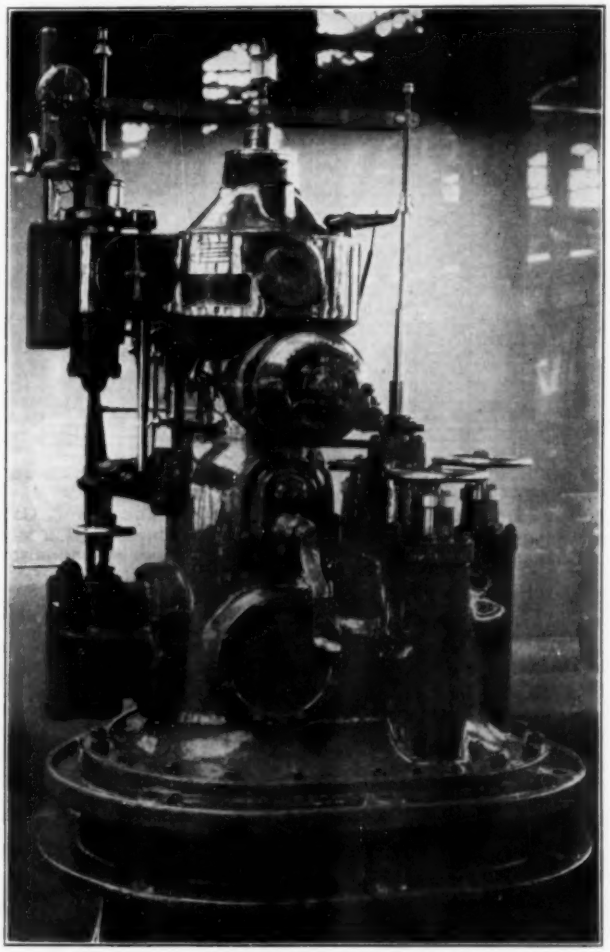


FIG. 1 MODERN ARRANGEMENT OF HYDRAULIC-TURBINE GOVERNOR WITH MOTOR-DRIVEN FLYBALL AND FULL AUTOMATIC ATTACHMENTS. ONE OF EIGHT REGULATING THE 13,000-HP. PROPELLER TYPE UNITS IN OHIO RIVER PLANT OF LOUISVILLE HYDRO-ELECTRIC CO.

(There are two schools of designers of governors. One considers the governor stand as a column which supports externally the various elements so as to provide the maximum of accessibility. The other treats the stand as a case which contains the various working parts. The former type is typically illustrated above.

The load limit device is at the upper right, its finger bearing on a collar of the valve stem. When it is in operation, the flyball, its motor, the upper floating lever, the entire dashpot mechanism, and synchronizing motor attachment at the upper left are inoperative.)

[1] Hydraulic Engineer, Allis-Chalmers Mfg. Co. Life Mem. A.S.M.E.

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

grouped in one system and combined with lighting and traction loads, the necessary refinement in control of voltage and frequency more than offset any advantage in diversity factor of load changes, so that regulating requirements were still severe. With more extensive system interconnection, the increase in total system flywheel effect, and the very small load changes resulting from wide diversity of load have altered conditions so that the governor on any individual unit may have a negligible effect on the system. As a result, and for gain in other directions, most of them are run "on the block," that is, with the flyballs inoperative, and as an emergency shutdown means only.

There is at present in this country a total capacity of hydraulic turbines under governor control far in excess of any possible load

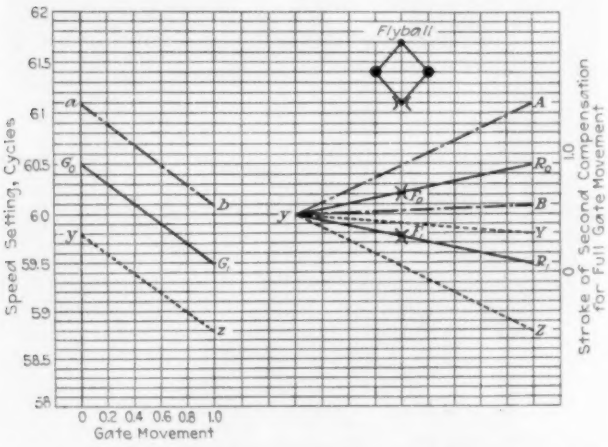


FIG. 2 DIAGRAM OF RELATIONSHIP BETWEEN FREQUENCY, GATE OPENING OR LOAD, THE NO LOAD TO FULL LOAD SPEED RANGE OF A GOVERNOR, AND ITS ADJUSTABILITY THROUGH THE SYNCHRONIZING MOTOR

(For proper division of load between units a normal range G_0 to G_1 is required from no load to full load. This may vary from one to three cycles and the shape of the connecting curve may be varied to effect desired relationships between units. Without altering this range the synchronizing motor is used to shift it on the frequency diagram from some such position as R_0 R_1 to AB or YZ resulting in speed ranges a-b or y-z. Ordinarily, this is done by the operator to correct frequency if he finds it low or high. By this correction full load speed may even be made higher than no load speed as indicated by the location of point b at 60.1 cycles and y at 59.8 cycles.

Control by frequency meter if extended to the full capacity range of the unit results in "flat" regulation—i.e., points y and b and also all intermediate load points are brought to 60 cycles.

Load-limit control blocks the effect of the flyball on the regulating valve V for all speeds below normal in the range F_0F_1 wherever it may be shifted on the frequency scale, but leaves it operative for speeds above normal. Similarly, an "anti" load-limit device[2] has certain utility for limiting the reduction of gate opening at overspeeds so as to enforce participation on other governors.)

changes that require continued speed regulation. About the only duty that the governors now installed have to fulfil is that of emergency shutdown when the load is lost completely through electrical disturbance. Under such circumstances speed regulation no longer exists. This is evident from the fact that even our best governors under such conditions permit the speed to vary 20 or 30 per cent, and that condition is not acceptable with any power system. In other words, when this speed variation exists, it makes no difference to the system what the speed of the unit may be because the unit is no longer connected to the system.

The only condition which has come to the author's attention requiring governors on large modern units is that in which the main load of the plant is through a long transmission line, service through which may be interrupted by some such cause as lightning. If under such conditions a plant, say of 100,000 kw., instantaneously loses 90,000 kw. when a line goes out, but has to hang on to a 10,000-kw. local load, governors may help the situation and be a necessity. The local load is subjected under that condition to perhaps a 20 to 40 per cent speed rise, and probably to a very positive voltage rise, so that even then it may be questionable whether the local load might preferably have been dropped entirely and later picked up without imposing such abnormal voltage or frequency conditions. Such a combination, however, is about the only logical set-up of conditions that might require complete speed governing.

[2] Patent pending, Pfau.

INSTABILITY OF HYDRO PLANTS

Instability of hydro units (steam units also) is becoming a large operating problem. By instability is meant that quality of unit which permits it to fall out of step electrically, thereby losing its load even though tied to the line. Present governors operated

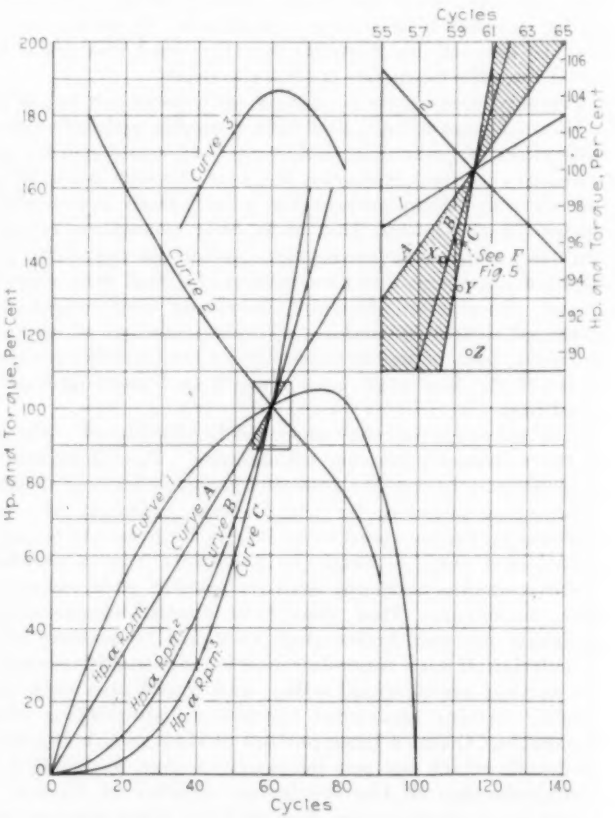


FIG. 3 SOME FEATURES OF SYSTEM CHARACTERISTICS AFFECTING REGULATING OF HYDRO-ELECTRIC UNITS

[Curve 1 shows the speed-capacity relationship of an average water turbine, its runaway speed of 66⅔ per cent being reduced by remaining windage and friction. The corresponding torque is shown in Curve 2, and the efficiency in Curve 3. Curves A, B, and C are load curves varying respectively as the first power, the square, and the cube of the speed. Most loads come within this range. A represents a load such as a sawmill. Curve C is representative of a centrifugal pump or blower load, while B is a composite of the two.

Points X and Y were obtained on systems of about 100,000 kw. capacity by blocking all governors and suddenly opening up the circuit breaker of a unit carrying about 5 per cent of the entire load. The remaining units slow down until the reduction of load at the lower speed brings back a balance between developed and absorbed capacity. System X is moderately self-regulating, while Y is more so. The governing problem on Y would be simpler than on X.

Z was obtained on a system of over 500,000 kw. capacity. All governors could not be blocked, so the end point which for 5 per cent load change is not reached for 15 or 20 sec., has been modified by some active governors opening up.

It is significant in tests of this kind that blocking all governors frequently results in less system frequency variation than with governors active. A reduction from 0.25 cycle to 0.1 cycle has been noted. Three separate systems tested by the author show this general tendency. (Compare B with A, Fig. 5.)

In tests of this kind the 15 to 20 seconds to reach a new stability point is an indication of the flywheel effect or regulating constant of the system. The ultimate speed attained for stability indicates the character of the load assuming all governors blocked. (Fig. 5, F.)]

normally make this situation worse, by increasing the turbine torque at subnormal speeds at a time when it should be reduced. Corrective measures may even be required in the form of a load-reducing coil that takes the control away from the flyballs.

Speed Regulation

Various overall types of regulation are experienced with hydraulic turbines.

1 All turbines are self-regulating to some extent, even without gate movement. The quality is no longer usable because the usual speed range, 20 to 50 per cent from no load to full load, is too great and because the efficiency loss is prohibitive. (See Fig. 3.) Special design can reduce the range, though not to the fractional per cent now desired, but can do but little with the poor economy except through gate closure.

2 A second class of speed regulation is that indicated in the 20 to 40 per cent speed rise guaranteed with instantaneous drop of full load. This results from the excess horsepower while the governor closes the gates upon large rejection of load.

3 A third type is the no load to full load speed change required to permit governors to divide load between units without hunting while on flyball control. This is 2 to 3 per cent. (See Fig. 2.) Range *H* to *L* in curve *C* Fig. 5 is a part of this range.

4 The fourth type, usually called frequency regulation, suppresses the effect of the third class of regulation. (See Fig. 5 *D*.)

The speed change of any rotating mass or system caused by excess or deficiency of generated horsepower is dependent on its flywheel effect and speed. One figure called *regulating constant* expresses its capacity completely.

$$\text{Regulating constant} = \frac{\text{wr.}^2 \times \text{r.p.m.}^2}{\text{hp.}}$$

This constant for the usual hydro-electric unit ranges from about 4,000,000 to 10,000,000. It is simply an index of the capacity of a rotating mass to store foot-pounds of kinetic energy as contrasted to the foot-pounds per second of free energy contained in the excess or deficient horsepower that changes its speed.

indicate that the improvements are frequently more radical than these figures indicate. This gives, roughly, a measure of changing condition as regards regulation of the second class. This speed change is temporary only, as it is corrected by the conventional governor, and by the self-regulating quality of the load. (See curves *A*, *B*, and *C*, Fig. 3.)

The third type of regulation, the speed change from no load to full load, is the permanent after effect of a load change. It is

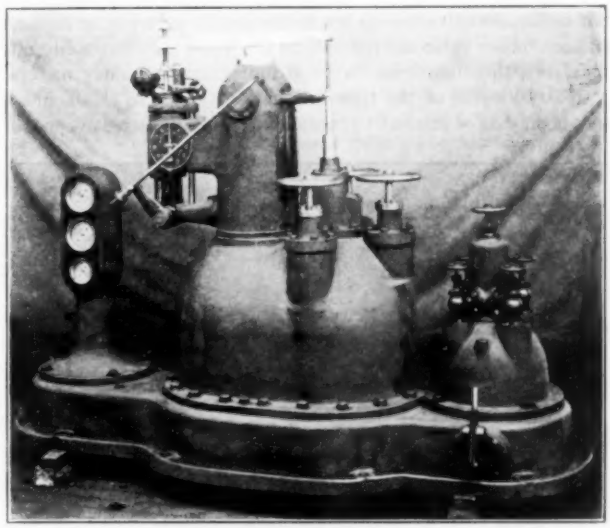


Fig. 4 Governor Stand Typical of Those Used on the 54,000-Hp. Conowingo Units, the 45,000-Hp. Martin Dam Units, and the 70,000-Hp. Niagara Unit, Speed Control Coming From a Shaft Type Flyball as Shown in Fig. 6

(The load limit or block, the rod of which crosses the gate dial, is not motor operated in this particular governor.)

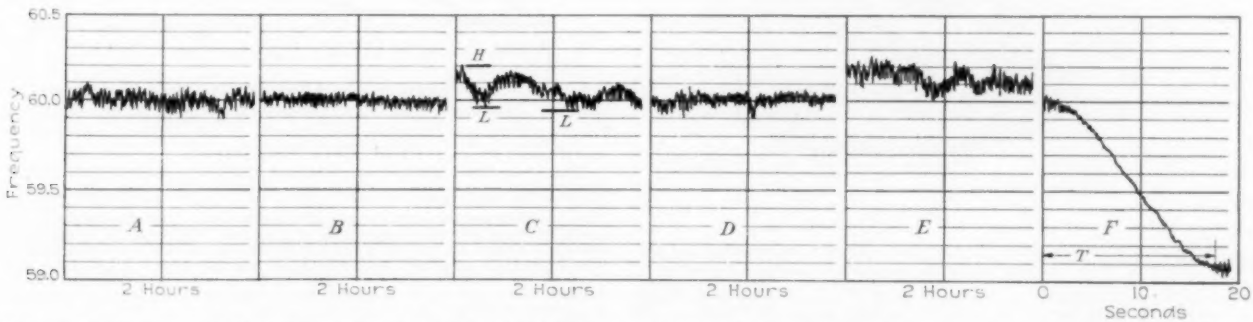


Fig. 5 Representative Frequency Charts Based on Six Systems in Various Sections of the Country. Small Systems Are Below 100,000 Kw. Capacity, Medium Systems Around 100,000 Kw., and Large Systems Around 500,000 Kw. Approximately Two Hours' Time in Each Section

(*A*—Good regulation on medium low-head system and large high-head system, both on governors.
B—Medium system with governors blocked, but no large load swing. Typical of all units set lightly against load limits.
C—Average of two large systems on governor control, but with hand correction of frequency swing resulting from gradual load changes. The usual method for greatest economy of water.
D—Similar system on frequency control. Average of large and medium systems. Wasteful of water if carried out on individual units or plants.
E—Small system without frequency control, but with varying numbers of its governors blocked, up to three-quarters of system load. No significant change evident.
F—Increased time scale indicating the effect of a load increase of 4 per cent of the system load on an average system when all governors are blocked. Frequency control on the load limit would slowly (about 30 sec.) correct the low cycles. The time *T* from opening of circuit breaker is dependent on flywheel effect of the system. The drop in frequency depends on the character of the load.
If load changes are within the operating time of frequency control, the frequency may be held within half a tenth of a cycle, whereas it requires a most excellent governor to hold it within one-tenth cycle without the secondary correction. Quick changes widen the range of both systems as it does that under hand correction. Such changes are not normal on large systems.)

As contrasted to a unit alone on a load like a lighting load and a regulating constant of perhaps 5,000,000, a large system may now have equal or less load changes and a regulating constant of 100,000,000. A load change that would have produced 20 per cent (12 cycles) speed rise with the former, will only produce about 1 per cent (0.6 cycle) with the latter. Actual field results

necessary only to permit units on flyball control to divide the load without trading it between units. (See Fig. 2, range R_1 to R_0 or G_1 to G_0.) Operation of a governor "on the block" or load limit is customary, and under such condition all flyball action and this third type of speed change become inoperative until manual control or excessive load drop may bring them in.

Frequency Control

The fourth type of control is designated as frequency control. In general, governors as now constructed may be considered to be sensitive to speed changes of $^1/_6$ to $^1/_2$ of 1 per cent and to regulate in parallel satisfactorily, with a no-load to full-load speed difference of 2 to 3 per cent. Expressed in cycles, the former requires 0.1 cycle to 0.3 cycle and the latter 1.0 cycle to 2.0 cycles. Refinement of frequency control affects both of these figures, but in a different manner. It affects the former, in that such insensitiveness as lies in the flyball, its drive, or its connections to the valve are reduced to the $^1/_{10}$ of a cycle readily obtainable with commercial forms of indicating frequency meters, or to 0.001 cycle of the time-integration type. In this connection, it may be of interest to compare roughly the sensitiveness of



Fig. 6 Details of Flyball Arranged for Direct Mounting on the Main Shaft of the Unit, but Mounted for Shop Test

(The forked bell-crank lever at upper right transmits the vertical motion of the flyball collar to the governor stand. This type of flyball is extremely powerful. Two types of design exist in this connection. One type utilizes a light flyball, and builds up the force and travel necessary to move the main governor valve by means of an intermediate relay valve, the stroke of the flyball collar not being mechanically connected to the main valve. The alternate design uses a powerful flyball motion which moves a pilot valve directly in the main valve, so that the position of the latter is at all times mechanically determined by the flyball.

The flyball in above view develops over 50 lb. pull for 2 per cent speed change. The motor-driven type of Fig. 2 develops 30 lb. Both are representative of the latter type of design described above.)

various forms of speed indicators. They may be outlined as follows:

Type of device	Approximate Comparative Accuracy: Thousandths of 1 per cent	Fractional cycles (based on 60)
Slave pendulum	1	0.0006
Chronometer	2	0.0012
Time integration method (Warren)	2	0.0012
Electrical frequency meter	20	0.012
Mechanical tachometer	20	0.012
Hydraulic governor insensitiveness	100	0.06
Hydraulic governor no load to full load	2000	1.2

These figures are only roughly comparative. The pendulum may be more accurate than 0.0006 cycle, and the chronometer different than shown, but the former is probably superior, hence its position.

From the valve to the turbine gates, and including insensitiveness of the valve, there is no difference between independent frequency control and regular governor control, but probably less than one-half of ordinary governor insensitiveness is attributable to that portion of the chain of operation. For speed regulation purposes, except of the third type, these general limits may indicate that frequency control, as such, will effect no radical improvement over previous governing equipment through flyballs alone.

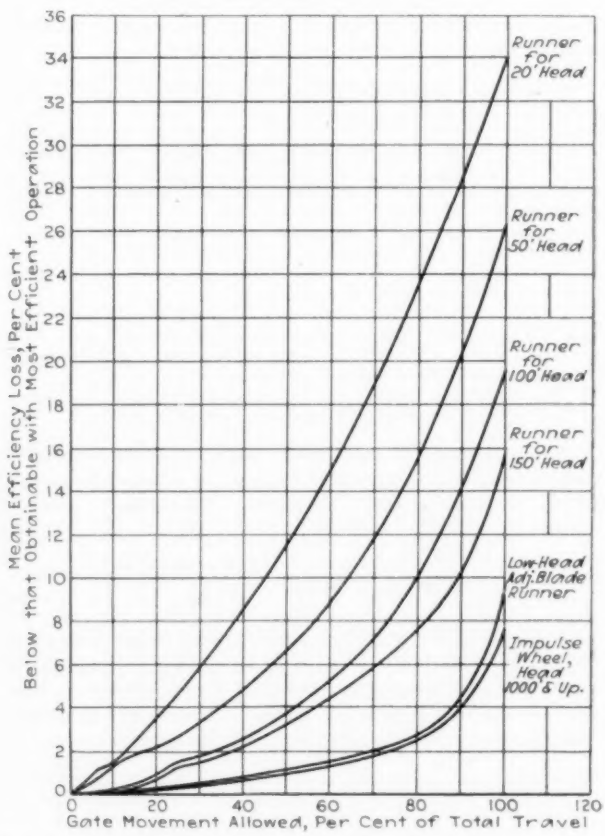


Fig. 7 Curves Showing, for Practically All Types of Hydraulic Turbines, the Percentage of Power Lost by Continuous Flyball Regulation Due to Reduction of Efficiency From That Obtainable

(These may vary slightly for runners of different makes, but the general characteristic is correct and checks out with observed plant performance. Every hydro plant might advantageously have one of these curves drawn from the efficiency curve of its particular type of unit to emphasize the wastage of kilowatt-hours during periods of governor regulation. For example, with such a curve an average plant would suffer a loss of 6.7 per cent in kilowatt-hours if the governors were allowed to regulate over a 50 per cent gate range as, for example, from 50 to 100. This is taken directly from the 50 ft. head curve above. It is simply the reduction in the average ordinate of the efficiency curve over a given range, below the peak ordinate.

Wider ranges of regulation and consequently greater losses are only too frequent. Such losses are further increased if more time is spent at the extreme gate openings of any particular range.)

The third type of regulation, the no-load to full-load difference, becomes of importance when particular apparatus, such as papermill drives and telechron clocks, are considered. The 1.2 to 1.8 cycles variation from no load to full load required completely to shut down a unit operating in parallel is more than is desirable for such modern requirements. For example, a one-cycle maintained variation, applied to a clock system, means about one-half an hour a day gain or loss in all synchronous clocks connected to

that system. This is merely mentioned as an indication of the value of frequency control down to fractions of a cycle.

From a governor standpoint, in order to secure a definite division of load between units operating in parallel or to prevent uncontrolled hunting or switching of load between units, some such a figure as two cycles variation between no-load and full-load speed is practically essential. This necessitates the governor having two ranges, one to permit of operation in parallel and the other to reduce or rather temporarily shift that range so as to enable maintaining constant frequency regardless of load. The quick-acting flyball control (full stroke in about 3 sec.) is then left as now designed, and the frequency control which is relatively slow acting (about 30 sec.), but very much more sensitive, added to it. It does not reduce the flyball range, but shifts that range so that at full load the flyball is centered at the upper end of its operating range, and at no load, centered at the lower end of its operating range. (See Fig. 2*b* to *y*.) The effective length of its range is unchanged so far as dividing load is concerned. The result of this shifting of its range is practically to hold constant speed.

Such a device is easily applied to any governor having a synchronizing motor. Engineers have worked on this as applied to the load-limit device, but as has been demonstrated by the Alabama Power Company, it may be carried out very inexpensively and very effectively, simply by connecting the raising and lowering circuits of the synchronizing motor to the red and green light connections, ordinarily supplied on certain forms of frequency meters. This arrangement does exactly what an operator does if he finds his frequency high or low. He corrects it by shifting the gate position by means of his synchronizing motor which is operated from the push-and-pull switch on the switchboard. If the unit is in operation on load limit, as perhaps 90 per cent of all units now are operated, he should do it by means of his load limit, particularly if that load limit has switchboard control which parallels that of the synchronizing motor. Engineers of the company with which the author is connected prefer the load limit arrangement because that is the only effective one, unless the synchronizing motor connection is operated so far as to take the unit off of the load-limit. This is mentioned as bringing out the fact that frequency control through the synchronizing motor is impossible, or at least is greatly delayed in effectiveness if the unit is on the load-limit device, and units are desirably and usually operated on that basis.

The slowness of the intermittent type of contacts made from the usual mechanically actuated frequency meter permits of the elimination of relay motion just as its elimination is permitted when an operator goes to the board and pulls the synchronizing motor switch to raise or lower frequency.

Kilowatt-Hour Output

The measure of financial performance of a hydro plant is found in the percentage of the kw-hr. inherent in the flow and head that it transforms into kw-hr. on the outgoing line. It should be directly comparable with the simplest bookkeeping of a flour mill, charging the bushels of grain bought and crediting barrels of flour. Hydro plants are initially financed on that sort of an analysis and should be so operated, though requirements of quality of service frequently prevent.

With the tremendous and increasing interconnection of systems, even the largest unit has but small effect on the whole system, so that units may be taken off or put on without noticeable disturbance. Pulling out a 40,000-hp. unit instantaneously makes hardly a noticeable effect on the largest systems, so that taking it off gradually should be entirely feasible. On that basis the maximum kw-hr. production is obtained by running each unit at its best point or with gates completely closed. Run-of-river plants cannot be shut down in case of lack of load without wastage of potential kw-hr., but that situation is rapidly being corrected by the development of offpeak loads exemplified by the installation of "pumpers," or hydraulic storage batteries. Plants which have some storage even though it be but daily or hourly should be the regulating plants, doing such regulation by starting and stopping units, all other plants to regulate very slightly, if at all. Governors designed to enforce such a regime would have an automatic shutdown device operable to close entirely the gates of successive units as their capacity at or just below their best point became excessive, or to open them up again in case of demand. A region of instability would thus be created over such gate ranges as are uneconomical. An upper limit just above the best point would be fixed by the load-limit device which would be backed off in case of excess demand of power and flow in the river capable of meeting it.

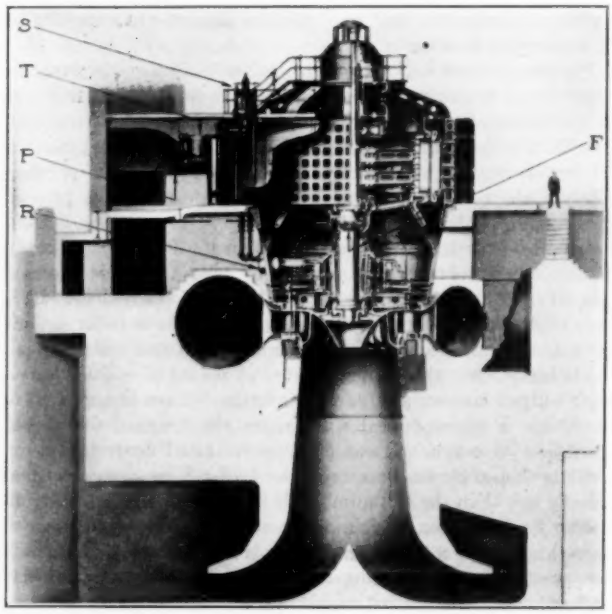


Fig. 8 Modern Large Capacity Unit Illustrating the High Energy Shaft Type Flyball *F*, the Control Stand *S*, the Regulating Cylinders *R*, the Oil Pressure Pump and Tanks, and the Inter-Connecting Piping

(All these elements are present in the conventional modern governor. The cylinders with the oil pumps and tanks are doubtless the most practical speed-reducing and accumulator means available. The other elements are affected more by new requirements of frequency and load control. This unit is installed at Niagara and has carried about 85,000 hp. with a head slightly in excess of 200 ft.)

Governing equipment to meet these conditions should be designed for the following:

1 Opening the gates to start the unit.
2 Controlling the gates to permit the unit to be synchronized with the line.
3 Opening the gates to the point dictated by conditions of flow available, load demand, or as is ideally desirable, to the point of maximum efficiency only.
4 Adjusting the gates to meet any change in conditions.
5 Emergency shutdown of the unit in case of electrical troubles such as instability, loss of load, lightning, etc.
6 Normal shutdown when the unit is no longer desired.

Not a single one of these requirements necessitates the quickness of control typified by the normal two- or three-second governor. In the majority of instances 10 sec. or even 60 sec. operating time is entirely feasible, except for condition (5), and neither that nor requirement (2) demands the use of a flyball

nor does the latter demand the use of quick action. On the contrary, most automatic plants require a deliberate slowing down on the motion as the unit goes through that gate opening which gives approximately synchronous speed. In both this feature of slowing the governor action for synchronizing, in blocking its gates positively by a load-limit device, and in several other ways, the fast governor action around which the most intricate designs are built, are definitely suppressed or eliminated entirely.

Frequency control, except on the basis of continually starting or stopping units, may be quite uneconomical. Mr. McCrea in the *General Electric Review* of June, 1929, illustrates such control down to 0.04 cycle as contrasted to 0.16 cycle on hand adjustment of governor control. From a frequency standpoint the results are gratifying, but to secure them the turbines were apparently operated over such a wide range of gate opening as to reduce the kw-hr. possible from the water used by over 10 per cent. This might mean the loss of the entire net operating profit and illustrates what may happen behind what appears to be a splendid showing.

Frequency control, if carried to the limit, often encountered in large plants, requires operation of one or more units in the plant as synchronous condensers at zero gate opening. This particular condition is the fly in the ointment in load control for maximum kilowatt-hours. The usual operation encountered is that wherein all units but one are blocked with the valve pressed against the load limit by perhaps half a cycle pressure (see Fig. 2), but with one or more units operating on the flyball to take the load swings in so far as they are able by reason of their kilowatt capacity. This is done because it is easy and convenient, but is basically wrong. Generally speaking, all of the units in a plant should be operated at equal gate openings if the units are identical. Technically, they should be operated at points of equal slope on their output curves, plotted on discharge. Even this system of operation is uneconomical when units are designed for proper operation as a synchronous condenser. Load control between units is then difficult, because if the load can be carried on materially less than the full number, it is more economical when dividing load between such a series of units to operate some of them at zero gate opening. This is true if the guide vanes are equipped with special sealing devices, and if the units are properly and automatically vented, simple draft-tube venting not sufficing for high-head units with specific speeds below about 50. This condition introduces some complication in frequency control, just as it does in load control, but is a complication that must be met when kilowatt-hours are the vital consideration. Unfortunately, considerations of load dispatching, convenience in operation, availability of market, and similar considerations, only too frequently preclude the operation of units for the maximum number of kilowatt-hours. It is not generally realized that this condition exists, nor is the remedy usually appreciated, except as a result of analysis of those characteristics of a hydro unit which seldom are written into a contract or shown on the guarantee curve for that unit.

Discussion

ARNOLD PFAU.[3] The paper is an able and timely contribution to the study of a subject which is undergoing radical and basic changes. The fact that we must deal with new requirements brought about by the merger of power systems into one large complex has set problems before us the proper solution of which will require radical changes in the governing means of these various prime movers.

[3] Consulting Engineer, Allis-Chalmers Mfg. Co., Milwaukee, Wis. Mem. A.S.M.E.

What we need in order to meet the new requirements of speed control of large systems may be crudely expressed in the words, "A governor which wakes up sooner, but does not have to get so busy to catch up with its task." To better illustrate this,

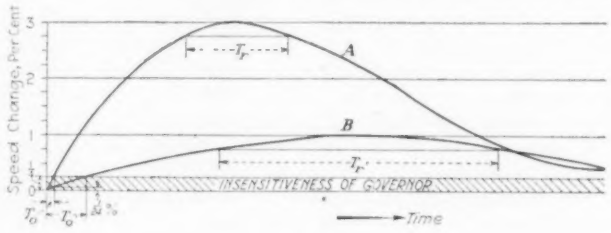


FIG. 9

let us consider two speed-change curves A and B, Fig. 9.

Curve A represents the speed rise in a small power system of, say, 20,000 kw. caused by a load rejection of 10,000 kw., or 50 per cent of its load. This speed rise may be 3 per cent.

Assume an insensitiveness of the governor of $^1/_4$ per cent from the time of the beginning of the load rejection until the control means (servomotor operating the guide vanes, etc.) receive their first motion. Here the ratio between insensitiveness and total speed rise is 0.25 to 3, or 1 to 12. The delay at the start is T_o sec. and may be T_r sec. for a reversal of the speed within the $^1/_4$ per cent range of insensitiveness. Assume a flywheel constant $c = \dfrac{WR^2\,(\text{r.p.m.})^2}{\text{hp.}}$ of the system of $c = 10{,}000{,}000$. Thus the momentary relative speed rise is:

$$S = \frac{800{,}000 \cdot \varphi \cdot T}{c}$$

where $\varphi = 0.5$, so that

$$S = \frac{400{,}000}{10{,}000{,}000} \cdot T, \text{ or } 0.04 \cdot T$$

Curve B represents the speed rise in a large system of, say, 1,000,000 kw. caused by a load rejection of 50,000 kw., or 5 per cent of its total load. Here the speed rise may be only 1 per cent, because the flywheel constant of that large system with all its rotating elements of prime movers and industrial driven parts may be 100,000,000, so that

$$S = \frac{800{,}000 \,.\, 0.1}{100{,}000{,}000} \quad T = 0.008 \cdot T$$

If the same kind of governor is used—i.e., with an insensitiveness of $^1/_4$ per cent—then the ratio between insensitiveness and total speed rise is 0.25 to 1, or 1 to 4, or three times as severe as it was with the small system, although a load rejection of only one-tenth (5 per cent as against 50 per cent) in intensity took place. Thus the delay becomes T_o' sec. and T_r' sec., which is also correspondingly more pronounced.

From the diagram (Fig. 9) it is evident that if the delay T_o were to be maintained also for curve B (that is, for the large system) the insensitiveness of the governor would have to be about $^1/_{24}$ per cent instead of $^1/_4$ per cent. In the small system the $^1/_4$ per cent insensitiveness represents, say, one-twelfth of 10,000 kw., or about 835 kw. In the large system it amounts to one-quarter of 50,000 kw., or 12,500 kw., which is fifteen times the former amount. The governor of the large system has "overslept," so to speak, in an amount of about 11,665 kw. It is obvious that this delay cannot be made up, because a correspondingly quicker movement of the control means of the turbine may become prohibitive on account of penstock conditions, etc.

It therefore seems to the writer that the question of sensitiveness is one which should receive the greatest attention, and he cannot but feel that a participation of as many prime movers in the control of the speed of the system will prove to be the most effective solution, provided that—

(1) their governors are highly and equally sensitive
(2) they operate at a moderate rate of gate motion
(3) they are normally limited in their action to within that range of output which still assures a satisfactory efficiency.

S. Logan Kerr.[4] The writer wishes to commend the author for his very clear summary of the present trend in hydraulic-turbine speed regulation. This fact of changing requirements is not as generally appreciated as it should be, although operating engineers of the various power companies have had this fact drawn to their attention through the new requirements in operation due to the extensive interconnections of power systems and their attendant problems.

The very interesting relations shown in Fig. 3 illustrate the self-regulating characteristics of power systems. This is a very interesting graphical representation of the statement which the writer made at a meeting held in Utica, New York, of the Empire State Gas and Electric Association on the general subject of "Automatic Frequency Control." It would be of interest to quote from the minutes of this meeting the writer's remarks since they are particularly pertinent to Mr. Nagler's paper and are not generally available in published form.

"There is an increasing need for constant speed, since many industries depend on constant speed to control their products. In selecting apparatus to control speed or frequency there are three distinct elements which must be considered:

(1) Rate and magnitude of load changes
(2) Capacity available and characteristics of governors
(3) Characteristics of load with respect to speed.

"Electrolytic and electric furnace loads are practically independent of frequency. Centrifugal pumping loads vary as the cube of the speed. Other rotating loads have more or less flywheel effect. Thus a system with a large proportion of electrolytic or electric-furnace load is more difficult to regulate than a system with considerable proportion of rotating or centrifugal pumping load, since the flywheel effect of rotating loads and the speed-power characteristics of centrifugal pumping loads tend to stabilize the system to a considerable extent. The increasing number of interconnections has made it necessary to regulate the frequency within much closer limits than before. Interconnections have also introduced the problem of where to control frequency. Steam units are capable of quicker changes in load than hydraulic units, since there is less inertia in flowing steam, whereas the inertia in flowing water columns is very considerable, particularly in long pipes. On the other hand, extreme variations affect economics, particularly in steam plants. Any governor, operating in parallel with other governors, must have an inherent speed drop from no-load to full-load to insure stable operation with the other units. The purpose of frequency-control equipment is to hold the speed at normal instead of allowing it to follow the inherent speed-drop line. This is accomplished by changing the governor setting or shifting the governor characteristic line in proportion to the increased load."

The author's enthusiasm for the "load limit" feature of a governor is understandable, but the writer cannot look upon this very simple device as the best answer to all operating problems. The load limit is very effective in limiting the maximum output of the unit and in avoiding continuous departures from a desired point of operation. In order to supply the commercial type of service, it is necessary to consider the variations in demand and to follow them rapidly if this elusive thing called constant frequency is to be attained.

The usual operator in the average hydroelectric station must readjust the "pressure" on the load-limit device each time he adjusts the device itself or else the flyballs will not act at the proper increment of speed above normal. This device also has the disadvantage that it prevents the unit from picking up additional load when the demand increases.

Also, the day has not yet come when the conventional governor action can be relegated to the past. Loads do change, and unless they cease to change rapidly, the governor will still be required to control the speed, while the correction of the frequency to normal can be safely left to ultrasensitive but slow-moving automatic devices.

In the last two paragraphs of the paper, the author mentions some of the practical phases of the problem: "Frequency control, except on the basis of continually starting or stopping units, may be quite uneconomical." And in the last paragraph he mentions the problem of load control, outlining its limitations and the difficulties in the way of operating generating stations for the maximum number of kilowatt-hours. The economic division of the load between units, the maintaining of the economic number of units in service, and the use of units as synchronous condensers are all cited as well as "considerations of load dispatching, convenience in operation, availability of market, and similar considerations [all of which] only too frequently preclude the operation of units for the maximum number of kilowatt-hours."

It is just these considerations that go to make up the daily life of the power-system operator. It is just these considerations that must be taken into account if commercial operation is to be conducted with the greatest economy and with the highest standards of service.

The fundamentals of speed regulation must be considered in conjunction with the characteristics of the prime mover, of the load being supplied, and local conditions must also be considered in order to achieve successful operation. One further element which is essential in all of these considerations is that of economy in the broadest interpretation, this being the economy of the system as a whole with respect to the cost of the energy produced.

A new element has entered the field of power generation—namely, the inseparable tie-up between hydraulic engineering, mechanical engineering, and electrical engineering. To consider all phases of the problem of electric-power generation of the most successful type, it is no longer possible for the hydraulic or steam engineer to neglect or ignore the electrical side of the problems, nor is it possible for the electrical engineer to look with disdain upon the work of the hydraulic and mechanical engineers. These elements are so closely interlocked in the final analysis of the problem that it is impossible to omit any single element which may affect the success of the combination.

The fundamentals of speed regulation are becoming more widely understood, the fundamentals of power distribution control are in the process of development, and the fundamentals of economic loading of units and of stations are becoming more widely appreciated. In order to secure the maximum utilization of the generating equipment with the maximum economy, it is necessary to meet certain requirements:

(1) The characteristic of the unit must be determined and the ideal scheme for economic loading established.

(2) The degree of speed regulation possible and the degree

[4] Research Engineer, I. P. Morris & De La Vergne, Inc., Philadelphia, Pa. Assoc-Mem. A.S.M.E.

of speed regulation desirable must be determined from the study of the individual characteristics of the system from the point of view of both the prime mover and the load characteristics, as well as the combination of the two which may be called "the self-regulating characteristics of the system."

(3) With this established, a form of frequency tolerance must be determined based upon these factors and with consideration to the system load demand and the requirements of the customers being supplied with power.

(4) The individual system operating problems must be studied so that their effect on the individual system is not transmitted to broader fields through interconnection.

(5) The schedule of operation must be determined to include consideration of power transfer between systems through interconnections and of the operating schedule required by the various types of generating stations, both steam and hydraulic.

(6) With all of these factors considered, a still further study must be made to determine which of these functions of an ideal schedule can be accomplished by manual means and how many must utilize some other method such as automatic control.

It has been found from actual studies that it is not usually possible to maintain rigid operating schedules when dependence is placed on individual operators. To maintain such a rigid schedule would require the constant attendance of one or more men at the control board to adjust the load between units following any change in the system demand. Load dispatching of the highest order of intelligence would be necessary even to approach the ideal schedule. With automatic control equipment, however, the economic operating schedules can be made an integral part of the functions of the control equipment and various supplementary functions be included to take care of local conditions.

This same principle can be applied to the control of the proper number of units in service and to a combination of plants whereby certain plants can be selected for frequency-control purposes and other plants kept at constant output and the combined efficiency of the entire group of stations maintained at their higher value.

This sounds like a message from "Utopia" or at least some hazy ideal which might be attained in the far-distant future when all ideals will come true. However, this is not the case, as this object has already been accomplished with an installation of automatic control equipment which has been in service for a period of more than a year.

It has been the privilege of the writer to undertake the studies as outlined in this discussion in connection with the Norwood Development of the Carolina Power & Light Company, in which a 60,000-kw. hydroelectric generating station is controlled automatically for a number of different purposes. The equipment which is in operation at this plant controls an economic division of load between units of dissimilar characteristics and maintains constant station output when desired without impairing the availability of the governing equipment for emergency service. At periods when so desired, the control is set to hold constant frequency on the system with the maintenance of the economic load distribution between units simultaneously, and on occasions it controls the distribution of power to various customers through separate tie lines independent of the output on the remaining portion of the station. The control equipment literally performs the functions of an ideal operator at all times and as a result has been termed the "automatic operator."

The writer visited the Norwood plant recently and inspected the equipment. The control has been in commercial operation continuously during one year, and the total outage time was $1\frac{1}{2}$ hours during the year, being due to causes other than the control equipment. There have been no replacements, no repairs, and no renewals required, and in addition the operating efficiency of the station has been maintained at an exceptionally high level. Through the use of the control equipment, operating losses were reduced from approximately 6 to 10 per cent to the order of 2.5 or 3 per cent when the control was first installed and more recently to less than $\frac{1}{2}$ of 1 per cent for appreciable periods.

Analysis shows that the losses which are now experienced are due to only having extra units in service when carrying the system reserve capacity or failing to have the proper units available for operation at the required time. All losses due to improper loading of the units have been completely eliminated.

There are also under construction at present an "automatic operator" for the Waterville Development of the same company, two installations for the Washington Water Power Company (one for the Long Lake Plant and one for the Post Falls Plant), and one installation for the Morony Development of the Montana Power Company.

The field of automatic operation of generating systems is a very large one and has given promise of the most gratifying returns when studied with due consideration to all of the factors involved. The savings made, due to the economic operation of the units and systems, are amazing, and the application of automatic control equipment to a number of the phases of the problem has been shown to be not only possible but highly desirable.

E. E. Woodward.[5] What is the saving in eliminating the governor? Except with very small turbines, power means must be provided to open and close the gates. Electric motors may be used for this purpose, but hydraulic pressure is generally used on account of the ease, exactness, and quickness with which the gates can be adjusted; and because of having stored energy in the pressure tank, the turbine may be stopped or started when electric current is not available.

We must, then, have servomotors, pumps, tanks, piping, valves, restoring rods, manual or remote control of the valves with gate-opening indicator. All we need to make a complete governor is a flyball governor and an anti-racing device. This effects a saving too small to be warranted.

What is the loss in eliminating the governor? Speed regulation is not needed on some relatively small, self-synchronizing, automatic plants which feed into a large system. Here the load may be full or part-gate as determined by the limit stop or by a float at the headwater. Even then the speed governor may occasionally be needed if the plant is called upon to carry a local load if separated from the main system.

All power units must at times be started and connected to the line. If the power is needed badly, as is often the case, any delay will be more or less expensive both to the power company and to the power users.

Back in the time when turbines were to some extent self-regulating, it was not difficult to open the turbine gates to about the right position and have the machine synchronized in a short time. It is different with the modern high-speed runner with draft-tube setting. At no load and with a fixed gate, the speed swing is often as much as 5 or 8 per cent. It would be difficult to estimate what the loss would be if such a unit was without speed regulation. To get it into the line might be possible, but would certainly be difficult, slow, and unsatisfactory. Such a condition is probably the most difficult one that a governor has to meet, but results are being obtained so that synchronizing can be effected with comparative ease and dispatch where it seemed impossible to get into the line without the governor.

[5] Woodward Governor Company, Rockford, Ill.

HENRY E. WARREN.[6] This interesting paper suggests an idea which would have been rank heresy a few years ago and even now may be opposed by the majority of designing engineers. Hydraulic governors for prime movers have been bought for the last thirty years on the basis of performance under conditions rarely met with in practice. Having been deeply involved in the business of designing and building such governors for many years, the writer can fully appreciate the efforts made by the manufacturers, which were actively encouraged by power-plant engineers, constantly to increase the possible speed of motion of the turbine gates. A governor which would open or close the gates in two or three seconds could command a much higher price than another which would require twice that time interval.

It is perfectly obvious, as the author points out, that in actual operation an expensive, quick-acting governor with complicated valve mechanism never has an opportunity to show what it can do. The load and speed changes are so gradual that slow movement of the turbine gates is to be preferred, and the features of sensitiveness and stability are alone important. The real question is how slowly can the turbine gates be closed after circuit breakers have opened without endangering equipment by overspeed and overvoltage. If the generator and turbine are perfectly safe at runaway speed and the electrical apparatus is protected against overvoltage, the writer can see no reason why the gate stroke need be made in less than 10 or 15 seconds. Certainly in system operation this gate speed is adequate to give frequency regulation far beyond that which is obtained or sought today.

The author is undoubtedly correct in pointing out that the sensitiveness of the centrifugal elements of governors now in use is too poor to render much aid in present-day requirements. Accepting his figures of 0.1 and 0.3 cycle as the limit of machine sensitiveness, the examination of frequency records of a large system, such as that of the New England Power Company, discloses that for days at a time there is not a single occasion when the frequency is in error as much as 0.3 of a cycle, and sometimes there are days when there is no error as great as 0.1 of a cycle. Obviously such results could not be secured by governors alone nor does it seem probable that the centrifugal elements played any part in bringing about this degree of speed regulation. It seems quite possible that during these periods of system operation the frequency would have been exactly as good if the centrifugal elements of the governor had been eliminated and they had been used only as convenient gate-setting devices. Of course, the writer does not advocate the elimination of the centrifugal elements, because there are times when by reason of circuit breakers opening or the necessity of running the units independently the speed-control function of the centrifugal elements is necessary; but he agrees with the author that these are secondary functions and the design of the governors may well be modified accordingly.

With regard to automatic frequency regulation, the writer would be inclined to offer an amendment to the author's statement that "frequency control, except on the basis of continually starting and stopping units, may be quite uneconomical." This statement may be true if automatic frequency control is applied only to a single unit on a large system or to a very small percentage of the units on a system; but if simultaneous frequency control is applied to a large number of units on the same system, so as to control all of them together in proportion to their output, especially where those controlled units are in storage plants, automatic frequency control may be as economical as hand control.

This statement assumes, however, that in any case the kind and number of units to be operated in various plants will be intelligently planned in accordance with the anticipated load and actual water and steam conditions. The writer feels quite sure that on large systems the comparatively small percentage of fluctuations in load during the greater portion of the day can be taken care of automatically with great precision and without loss of efficiency if from one-third to one-half of the total generating capacity is under simultaneous control; the remaining generating capacity may be running with fixed output.

This is believed to be true because the short-time percentage fluctuations in load on large systems are for the most part less than 10 per cent and generally less than 5 per cent. Therefore, this would involve a range of generator output, if half the capacity was automatically controlled, of from 10 to 20 per cent on the controlled generator. Such a range should be well within the efficient zone of most modern turbines. Therefore, this kind of regulation should not prevent the most efficient use of the water.

The very great advantage of automatic frequency regulation simultaneous at many points on a system is the tendency of such regulation to reduce greatly the fluctuations in the tie-line loads. Obviously regulation at a single point on a system, whether manual or automatic, involves very large energy fluctuations in the tie lines which diverge from that point; because all changes in system energy which may occur at the most distant points must be corrected at the central regulating point, and the increase or decrease in energy must in the beginning be transferred through all the intermediate tie lines to the point where the change in load has occurred. After correct frequency has been restored by increase or decrease in power at the central regulating point, it is then customary to readjust the tie-line loads by supplementary corrections at other stations.

In the case of automatic regulation at many points on a system, a correction for increased or diminished load at any point on the system is divided proportionately among all the automatically regulated units. This means that the change in output of any one unit is only a small fraction of the total correction, so that if the turbines are operating in the region of their most efficient performance they may continue to remain in that region. Since each regulated station is correcting for only a small portion of the total change in load, the increased or diminished power flows to the point where the load change occurs, over many tie lines, and the change in any one tie line is generally very much less than the total amount of the load change. Therefore, less adjusting of the tie-line loads will be necessary where multiple frequency control is in use. In fact, since there is a tendency for load changes to occur successively at different points on the system, the small tie-line variations are likely to be distributed quite widely over the network and to counteract each other.

When major load changes occur, as at seven, twelve, one, and five o'clock, multiple automatic frequency regulation will be able to handle the situation if the output of the controlled units is sufficiently large to give the most economical results at other times. Obviously, however, it will be desirable to add or subtract at such times some of the units which are running on fixed output so that all units on the system may continue to operate with the most efficient output.

Such readjustment of units as may be necessary to bring about this economical performance can be carried on by hand oftentimes according to a schedule prepared in advance. It would not seem worth while to add to the cost and complexity of the equipment by automatic devices merely for the purpose of shifting the output of a generator three or four times a day from its minimum to its maximum value, or vice versa. Undoubtedly there are certain plants which by reason of size or

[6] Warren Telechron Company, Ashland, Mass.

hydraulic conditions should not be equipped with automatic frequency control apparatus.

The solution of the problem of simultaneous automatic frequency control at many points on a system depends principally upon three factors:

(1) The ability to measure frequency with a degree of absolute precision which is very materially higher than the ordinary operating range of the frequency. For example, if its operating range is ± 0.1 of a cycle, the automatic control device should be able to measure the absolute value of the frequency to a precision much better than 0.01 of a cycle; otherwise, the different frequency controllers will not work in unison, and there will surely be a tendency to trade power.

(2) The automatic control devices must be so perfectly coordinated that they will not only act in unison, but will also bring about proportional correction. That is to say, if a total increase in load of 10,000 kw. comes on a system and there are five regulated stations, each one of these five stations must increase its output by its proportionate share of the total amount. If the stations were equal in size, each one of them should take on 2000 kw.

(3) Multiple automatic frequency control devices must be reasonably free from a tendency to shift load from one station to another. This requirement will practically be met if controllers are perfect with respect to the other two requirements.

There are several possible ways of providing multiple automatic frequency control which will comply with the foregoing requirements. Perhaps the most obvious method is by the use of interconnecting signal wires or a radio broadcasting system which will transmit controlling impulses from a central frequency-measuring point so as to add or subtract simultaneously proportionate amounts of power at many points on a system. Such an arrangement would be ideal as regards coordination, but the expense of installation might be prohibitive.

The second method is that now in use by certain companies where controllers of the frequency-meter type are so arranged as to be slightly biased by generator output. As a result the small inherent error of these meters is neutralized by the amount of the bias at some point in the generator output. This means that such a controlled unit will gradually assume a definite output value on a system which is running at a fixed frequency value. The position which this unit will assume on such a system may be easily altered by manual adjustment of the amount of bias. Controlled units of this type should be stable and should respond practically in unison on small load fluctuations. They do not tend to maintain a frequency curve that is exactly flat, but rather one which has a slight sloping characteristic.

The third method is by the measurement of errors in the integrated frequency determined over very short time intervals by a very special type of master clock. This method has extremely high precision and tends to maintain a perfectly flat frequency curve. The different units are also effectively coordinated. The tendency to transfer power from one station to another is not entirely eliminated, but is reduced to a very low rate, so that it is believed to be practically unobjectionable.

Any one of these three systems can be operated in parallel with the others, but the type of control which maintains a flat frequency curve will probably tend to do a larger proportion of the work of regulating on large load changes.

In the writer's opinion one or more of these three methods of automatically regulating frequency simultaneously at many points on all large systems will eventually come into general use, and the result will be an improvement in the system efficiency —a very marked reduction in tie-line difficulties and an increase in the size of the network which may be operated satisfactorily.

J. P. Jollyman.[7] The operating experience of the Pacific Gas and Electric Company confirms the author's conclusions. This company's system contains 33 hydro plants, with 77 units of all types and sizes, and 5 steam plants. Interconnections are maintained with several other hydro systems.

Nearly all the hydro units are operated on a block load set by load-limit devices or by hand-controlled needle nozzles. In these plants the governors act on a rise in speed of about 0.5 cycle, and they may be required to govern a section of the system which has become isolated. When conditions are normal, only one hydro plant governs, and frequently only one unit is used.

Several of the smaller plants have been re-equipped for semi-automatic operation. The governors have been replaced with controllers operated by penstock pressure and equipped with a motor-operated load limit. These controllers adjust the gates to the water available and close the gates in the event of overspeed. The units must be restarted manually, because the generators are not suitable for automatic synchronizing. Shutdown on overspeed is accomplished by an overspeed trip operating electrically, backed up by a mechanical overspeed trip.

One plant recently built is equipped for full automatic operation. The turbine controller, in addition to the functions previously mentioned, brings the unit up to synchronous speed after the transmission line to the plant is energized. When synchronizing has been accomplished, the controller adjusts the gates to the water available.

The turbine controllers are simpler than governors for small units and medium or high heads, and are therefore preferable for automatic or semi-automatic plants.

The company is considering the application of automatic frequency control to the system, but has not selected the type to be employed.

Each important plant is furnished with information showing the loads for each unit that give the best overall efficiency. Motoring for synchronous condenser effect is best accomplished with impulse waterwheel units. High-head turbines with close runner clearance cannot be safely operated with the casing drained unless provision is made to admit a small amount of water for cooling the runner.

E. B. Strowger.[8] Since definitions may sometimes clarify the treatment of a subject, the writer wishes to submit the following definition of the word "sensitiveness" as applied to governors before discussing this illuminating paper: Sensitiveness of a governor is the degree of speed change required to cause the turbine gates to move in response to changes in speed on the main unit.

The author's conclusion that, except for the correction of frequency due to the inherent speed drop of governing equipment, frequency control, as such, will effect no radical improvement over previous governing equipment through flyballs alone, is an important one to recognize. In other words, with given load and generator characteristics, the small fluctuations in frequency from the average temporary value will still be of about the same order of magnitude with frequency control equipment as without such equipment. Frequency control substitutes the insensitiveness of the frequency control equipment, which for the time-integration type may be of a very small order of magnitude—say, from 0.01 to 0.001 of a cycle—for the insensitiveness of the flyball element, which probably is of the order of 0.05 to 0.15 cycle. The insensitiveness of that part of the

[7] Hydroelectric and Transmission Engineer, Pacific Gas and Electric Company, San Francisco, Cal.

[8] Assistant Hydraulic Engineer, Niagara Falls Power Company, Niagara Falls, N. Y.

governing equipment from the floating lever to the turbine gates remains in any case. This includes that due to pilot-valve lap and that due to lost motion in the floating-lever connection to the flyball element, if this can vary the floating center of the lever for any given speed. This point is mentioned since in practice there are old governors on systems with which the use of frequency control apparatus has been contemplated. Due to avoiding an outage these governors have not been taken down for repairs, and to obtain good results in the application of frequency control devices a thorough overhauling should probably first be made, especially as regards the elimination of lost motion.

The writer has encountered views on the part of men concerned with the operation of hydroelectric plants which were at variance with these statements and were based solely upon the fact that the frequency control apparatus keeps sending impulses to the synchronizing motor until the position of the pilot valve and therefore of the turbine-gate opening has changed enough to bring the frequency to the desired value. It should be remembered, however, that the pilot valve must be restored to neutral again, which involves the taking up of the existing lost motion in the opposite direction to that of the first corrective action. This may result in a tendency of the frequency to swing within the range as determined by the insensitiveness, as just explained.

The best that can be done, then, by means of frequency control in the direction of decreasing the band on the frequency chart which represents the governor's insensitiveness is to change it from, say, the range of 0.1 to 0.3 cycle to the range of 0.05 to 0.15 cycle, which is little different in order of magnitude. It may therefore be said that the limit of tolerance in sensitiveness is controlled, not necessarily by the sensitiveness of frequency control apparatus, but by the sensitiveness of present hydraulic apparatus. It must be realized, however, in making this statement that small quantities are referred to and that the small departures of the pen on the frequency chart from the temporary average value are not entirely due to insensitiveness of the governing equipment. These momentary variations are also dependent upon:

(1) The momentary amount of load changes on the system and the manner in which these load changes are divided among the regulating units.

(2) The momentary amount of excess or deficiency of input energy to each of these units over or under the demanded output.

(3) The amount of connected Wr^2.

(4) The speed-torque relationship of the load.

These momentary fluctuations are shown on the author's Fig. 5. The excess or deficiency in energy transfer from the penstock to the turbine is manifested by a rise or fall in speed of the rotating parts. If in a given system the units having the shortest penstocks and the largest values of Wr^2 were set with the narrowest regulation in order to take the load swings, then the variation in system frequency would tend to be at a minimum. If, on the other hand, the units with the long penstocks and the smallest values of Wr^2 were set with the narrowest regulation, then the variation in system frequency would tend to be at a maximum.

The speed change caused by a load change on any system is given by the following formula. The plus sign applies to the problem of loads off, the minus sign to that of loads on, and in selecting the value of Wr^2 the load should be taken into consideration as well as the generating equipment.

$$\frac{N_2 - N_1}{N_1} = \sqrt{1 \pm \frac{3{,}229{,}000}{N^2 \times Wr^2} \Sigma\,(\text{hp.} \times \Delta t)} - 1$$

where W = weight of rotating mass in pounds
r = radius of gyration in feet
N_1 = normal value of speed in r.p.m.
N_2 = maximum or minimum speed in r.p.m.
$N_2 - N_1$ = change in speed in r.p.m.
$\frac{N_2 - N_1}{N_1}$ = relative change in speed expressed as a fraction of normal speed.
(hp. $\times$ Δt) = integration of excess or deficiency in horsepower with respect to time.

It will probably be interesting to note that at present the Wr^2 value of the generating equipment and the load of the Buffalo, Niagara & Eastern system is approximately from 500 to 600 million sq. ft-lb. The largest value of Wr^2 in any one generator on the system is 70,030,000. If we consider that this value of, say, 550,000,000 for the Niagara system has increased from about 150,000,000 in 1918, then the speed rise obtaining both in 1918 and at present for an amount of excess energy of 100,000 hp-sec. would be 10 per cent and 3 per cent, respectively. This amount of excess energy would correspond roughly to the dumping of 40,000 hp. in 5 sec. from units running at 100 r.p.m., which units were set with a narrow regulation in order to assume the regulation for the system. These figures substantiate those of the author in regard to changing conditions of regulation which have taken place in the last few years.

Although the small fluctuations in frequency are not affected appreciably by frequency control apparatus, the suppressing of that variation due to the inherent speed drop of the governors may result in a big improvement in frequency service, especially where the manual system is not followed up by the operators with vigilance and where the characteristics of the load are such as to be adverse to self-regulation.

The author gives data showing the possible operating efficiency loss due to uneconomical loading caused by controlling frequency to close limits. The improvement of frequency service beyond the point where it begins to interfere with economical load division is probably unwise provided the service is already within commercially satisfactory limits. In general, economical load distribution should be given first consideration and speed regulation second consideration. Any method of frequency control which, by its application or by the application of supplementary devices, interferes least with the maximum production of energy from a given resource is to be highly commended.

Author's Closure

Mr. Warren's discussion is about as complete a comment on this general subject as could be presented. One or two thoughts occur that may be of interest.

While 0.1 to 0.3 cycle may represent average insensitiveness of existing equipment, it should not be forgotten that the inactive band involved in such figures need not always be used in its full width before present governors come into play. Some governors on any existing group may be at or near the top of the band, so that on an average certain changes may be met with considerable refinement of control. It is only on complete reversals that the entire band becomes effective.

Frequency control within the range mentioned by Mr. Warren as "the efficient zone of most modern turbines" does not involve the kind of error that creeps into the operation of many plants, but, even though it does not, it is still less efficient in the production of kilowatt-hours than operation of any unit at a given point only. Any turbine unit has a maximum point of efficiency which is a point only and not a band. Any operation away from that particular point results in reduction of the kilowatt-hours that could otherwise have been produced. Whether this is a sufficient

reduction to warrant some sacrifice in frequency control is entirely a matter of economy and quality of service, but the fact remains that any regulation whatsoever is accomplished at the sacrifice of kilowatt-hours. Fig. 7 is presented in an effort to picture that penalty of regulation for practically all existing types of turbines.

Mr. Strowger's comment on the Wr^2 value of the Buffalo, Niagara & Eastern System is of special interest. It is a more comprehensive picture of the tremendous increases of flywheel effect that are changing the regulation picture. It could only be some such stabilizing influence that makes it almost impossible to note the effect of dropping off loads as high as 30,000 kilowatts from large systems or, as was actually tried, the sudden increase of a similar amount resulting from cutting off a unit carrying that load.

Mr. Woodward's comments are extremely practical. There is no question whatsoever that a standard governor can be installed about as cheaply as any control mechanism. It is not the cost of equipment, but rather the extent to which that equipment may contribute to improper operation that plays such an increasingly important part in plant operation.

The author would not agree with the inference in the closing paragraph of Mr. Woodward's discussion. High-speed runner units are all too frequently unstable, but the correction of the instability from a hand-control gate to a governor-control gate is not always noteworthy. Frequently the governor action may give accumulative swings that are entirely prohibitive. Frequently the complete blocking of the governor will result in a smaller amplitude of swing or a slower swing than with a governor. This is only at the synchronizing point where extremely high-speed units are difficult to control under certain conditions of length of flume and draft tube that give a water-pendulum time somewhere nearly approaching governor time. It hardly seems possible to anticipate and design for such conditions.

Mr. Kerr's comments as to automatic loading units to the economical point are significant as pointing to the increasing attention that is being paid to operating units at their best gate openings. There is no question but such operation of units that are maintaining a frequency is proper. Regardless, however, of the perfection of such control, any operation other than at the point of best efficiency involves a loss that could be avoided. How much simpler and more productive of kilowatt-hours it would be if every unit on a big system could be operated at only one point. This would simply mean that any unit would be operating at that point or have its gates entirely closed. With systems large enough, the steps between numbers of units would not be reflected in a very considerable change in frequency. Such operation would be ideal from the standpoint of efficient use of water. The moment that intermediate control of units to correct the step-frequency changes becomes imperative, Mr. Kerr's suggestions as to the means for proper division of load between units or proper loading of units becomes the logical solution. It is the feeling of the author, however, that the greater simplicity of hand control under definite operating instructions to run units only at their best point, combined with the increasingly negligible effect of starting and stopping units, presents the most fertile field of increasing the kilowatt-hour output of most existing plants. As interconnection becomes greater, the tendency in the direction of that simplicity can hardly fail to become more pronounced.

Determination of the Actual Theoretical Head of a Centrifugal Pump

By MICHAEL D. AISENSTEIN,[1] OAKLAND, CALIF.



THE purpose of this paper is to show how the theoretical head of a centrifugal pump, as calculated by means of the classical theory, can be made to agree with experience. The study presented is based upon the actual performance of a large number of centrifugal pumps. The following symbols and nomenclature are used (see Fig. 1):

d_1 = diameter at inlet of the impeller in feet
d_2 = diameter at outlet of the impeller in feet
K = coefficient of head loss
c_1 = absolute velocity of water at entrance of impeller, feet per second
c_2 = absolute velocity of water at exit of impeller, feet per second
w_1 = relative velocity of water at inlet of impeller, feet per second
w_2 = relative velocity of water at outlet of impeller, feet per second
v_d = velocity in volute casing
u_1 = peripheral velocity of impeller at inlet, feet per second
u_2 = peripheral velocity of impeller at exit, feet per second
θ = water angle
h = total actual head in feet
H_{th} = imaginary total theoretical head
H = actual total theoretical head
e = efficiency per cent
N_s = specific speed in r.p.m.
N = revolutions per minute
δ = casting angle (vane angle).

The equation of the total theoretical head of a centrifugal pump is given by

$$H_{th} = \frac{u_2 c_{u2} - u_1 c_{u1}}{g} \quad [1]$$

where $c_{u2} = c_2 \cos \theta_2$ and $c_{u1} = c_1 \cos \theta_1$.

The following assumptions are usually made when deriving the fundamental Equation [1]:

1 The liquid is incompressible
2 The liquid is inviscid
3 The liquid is frictionless
4 All passages are filled with liquid
5 The liquid follows the angles of the vanes.

To obtain the actual head it is necessary to subtract the losses from the theoretical head:

$$h = H_{th} - \text{losses} \quad [2]$$

and consequently the hydraulic efficiency is

$$e_h = \frac{h}{H_{th}} \quad [3]$$

The pump efficiency is

$$e = e_h \times e_v \times e_m \quad [4]$$

where e_v is the volumetric efficiency, which is the ratio of the short-circuited liquid past the wearing ring to the total quantity, and e_m is mechanical efficiency, which takes care of the power absorbed by the disk friction, stuffing boxes, and bearings.

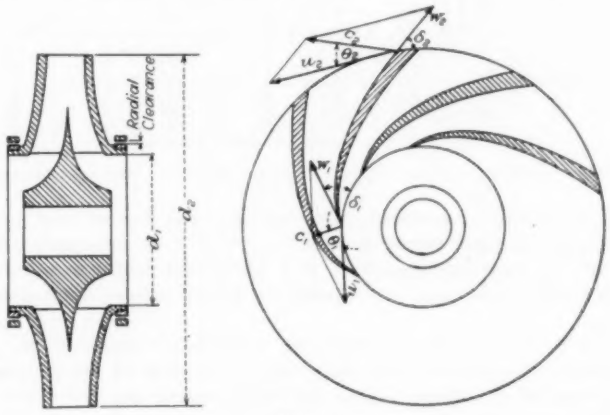


FIG. 1

It is a well-known fact that the hydraulic efficiency when using h from actual test and H_{th} calculated from the physical dimensions of the pump, by means of Equation [1], is very often less than the pump efficiency, which includes the volumetric and mechanical efficiencies.

Many discussions have arisen over this fact, and attempts[2] have been made to discredit the sound fundamental Equation [1] on the ground that it gives too large values of H_{th}.

Since u_2 and u_1 are functions of the outer and inner diameters of the impeller, they are fixed quantities. c_{u1} is small, and the product $u_1 c_{u1}$ has a relatively small effect upon H_{th}, therefore we have to look for an error in the calculations of c_{u2}.

Most of the authors on hydraulics agree that the velocity c_{u2} as calculated from dimensions and angles of the impeller is never obtained, since the assumption that water follows exactly the casting angles does not hold true.

Experiments indicate clearly that the theoretical head for the same outer diameter of an impeller depends upon the number of vanes, the curvature of the vanes, the ratio of the diameter of the impeller eye to the outside diameter of the impeller, the width, and the shape of the blade, and the design of the suction elbow and the discharge casing.

As an illustration see Fig. 2 where are plotted two head-

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[2] "Centrifugal-Pump Economics," by A. F. Scherzer. Trans. A.S.M.E., vol. 51 (1929), paper FSP-51-15.

capacity curves from actual tests which show the effect of the number of vanes in the impeller. Both impellers were of the same design, the only difference being that one had 8 vanes and the other 16. Therefore c_{u2} is not a simple function of angles and radii, and a correction factor should be introduced which will modify the value of H.

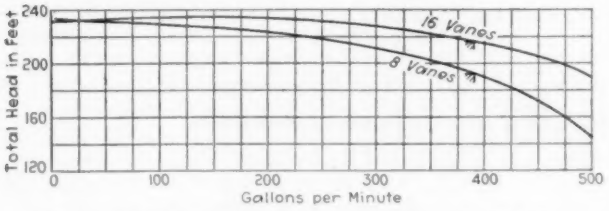


FIG. 2 HEAD-CAPACITY CHARACTERISTICS OF A $2^1/_2$-IN. CENTRIFUGAL PUMP

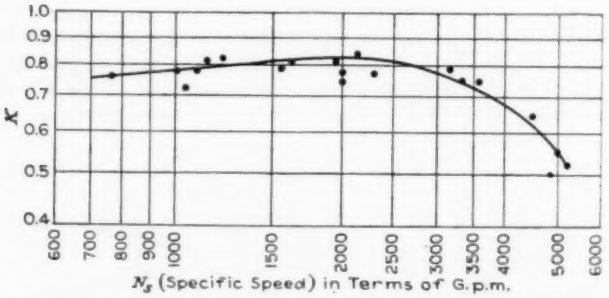


FIG. 3 RATIO $K = c'_{u2}/c_{u2}$ PLOTTED AGAINST SPECIFIC SPEED

As already mentioned, there are so many factors that influence H—and correspondingly c_{u2}—that it is impossible to take up each one separately; and this led the author to search for some term which would embrace all the factors influencing c_{u2}.

Very fortunately we have the expression "specific speed," which possesses the advantage that it involves all the peculiarities of the design of the centrifugal pump and determines the type of impeller.

The "specific speed" (N_s) in terms of gallons per minute is the speed at which a model of a pump when reduced proportionately in all dimensions from the original will deliver one gallon per minute at a head of one foot.

$$N_s = \frac{N\sqrt{\text{g.p.m.}}}{h^{3/4}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [5]$$

Every designer of centrifugal pumps is familiar with the fact that any change whatsoever in design of the impeller results in a change of the specific speed.

For this reason the author determined from actual tests and physical dimensions of a number of pumps the ratio $K = c'_{u2}/c_{u2}$ and plotted this against the corresponding specific speeds. (See Fig. 3.) This chart enables the designer to make a correction for c_{u2} and theoretical head when the head, capacity, and speed—and consequently the specific speed—of a pump are given.

After c'_{u2} is determined, the whole velocity diagram at the discharge of the impeller can be easily calculated.

Analyzing the curve shown in Fig. 3 it is seen that K increases gradually with the specific speed, and after reaching a maximum between $N_s = 1500$ and $N_s = 2800$, drops more abruptly.

At higher specific speeds, say, from 3000 to 5000, the impeller is very pronouncedly of the mixed-flow type, with the ratio of inlet to outlet diameter approaching unity; this results in relatively poor guidance of the liquid and the coefficient K decreases.

On the other hand, as the specific speed decreases the casting angles become smaller and the vanes have a much greater curvature.

Moreover, because of the relatively small angles the number of vanes decreases. This affects the guidance of the liquid and the coefficient K again decreases.

It is worth while mentioning that the highest efficiency is obtained when the specific speed is such that the value of K is a maximum.

This suggests that because of better guidance the losses become smaller and consequently the efficiency of the pump is improved.

Mechanics of Hydraulic-Turbine Pressure Regulation

By ARNOLD PFAU,[1] MILWAUKEE, WIS.

Problems of pressure control or the pressure variations resulting from the action of the governor complicate calculations of hydraulic turbines. This paper describes mechanical means whereby desired field results can be obtained under existing conditions of operation and of pipe-line characteristics. Calculations, formulas, and examples of speed and pressure control take up such points as closing gates, passing through position of static balance of turbine gates, formulas for speed regulation, effect of pressure variations on speed, and calculations of pressure rises in pipe lines.

THE problems of speed and pressure control of hydraulic turbines are closely interlinked. The problem of speed control is simple when no pressure variations are involved, and the practical methods of calculation are sufficiently accurate for commercial application. It is the problem of pressure control, or the pressure variations resulting from the action of the governor, which complicates the calculations.

Many excellent methods of calculation have been published and formulas have been given for use in practice. Most of these calculations, in order to be possible at all, or to avoid very complicated mathematical problems, are based on certain assumptions, or limitations. If one limitation precludes in practice another limitation, also embodied in the calculation, it is impossible to expect a result which is directly applicable in practice. For instance, it is practically impossible to simultaneously meet all three of the following conditions: namely,

1 That the closure of the turbine gates is a straight-line function with respect to time and no cushioning effect is present at the end of the stroke nor "dead time" at the beginning;

2 That the time required to effect a partial stroke of the gates is a direct proportion of the time required for the full stroke. That is, the closure from half-gate to zero is effected in one-half the time required for the closure from full gate to zero, and that a closure from one-quarter of full gate to zero is made in one-quarter of the full gate time and so on; and

3 That the discharge through the orifice and hence the average velocity in the conduit is considered to be a direct linear function of the opening of the orifice, there being no variation in the coefficient of discharge.

This paper is not intended to depreciate the value of present methods of calculation, but merely endeavors to describe practical mechanical means whereby desired field results can be obtained under actual existing conditions of operation and of pipe-line characteristics.

To explain these means properly a brief analysis of the principles of functioning of a servomotor is made, and a few suggestions are added.

Hydraulic-Turbine Governors

Rating. It has become an accepted practice to rate hydraulic-turbine governors, or servomotors, in foot-pounds. This is incomplete in two ways:

1 The time element is omitted, and

2 The maximum force available at any intermediate position of piston stroke is not indicated.

1 *Time Element.* It is evident that a servomotor producing the rated foot-pounds in one second is superior to one requiring 10 seconds or more for a full stroke, as was the case with the mechanical governors of the older kind which produced a rotary moment with one or more complete turns for a full stroke. Furthermore, it seems that there is no precise definition of the term "governor time," for which three different values can be distinguished: namely,

a The time T_s required by the servomotor piston to travel over the full stroke S, such as is generally determined by stop-watch measurement of the seconds, and with control-valve piston raised (or lowered) the full stroke Y. It is evident that under the above conditions the servomotor piston travels at its maximum speed, so that T_s is the shortest period obtainable.

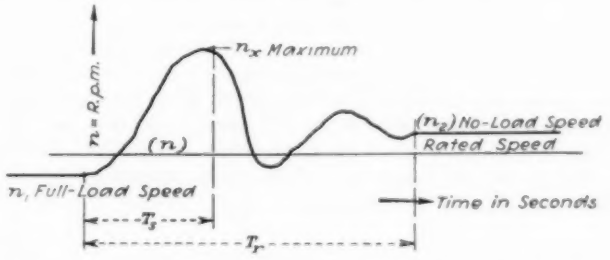


Fig. 1 Curve of Turbine Speed n as a Function of Time (Imperfect governor action because of oscillation.)

b The time T required from the beginning of the load change ΔN until the servomotor piston has completed its full stroke S. This period is longer than T_s because it includes the dead time of the flyballs (t_o, as explained later), and due to the action of the relay or restoring mechanism it does not permit of constant maximum valve lift Y, consequently the servomotor piston does not move at its maximum speed.

c The time T_r required from the beginning of the load change ΔN until the new constant speed has been attained in accordance with the new load. This period is the longest of the three, because it includes that additional time over T which is occupied by possible speed waves Fig. 1 (hunting with decreasing amplitudes), or by a gradual reduction, Fig. 2, from a larger speed variation to a finer adjustment, as obtains with a so-called second-relay compensating dashpot.

Maximum Force. A governor may be capable of producing the total foot-pounds required for a full-stroke operation, yet it may prove unfit if it lacks that maximum force necessary at an intermediate point of stroke to overcome the resistance of the turbine operating mechanism.

It is proposed here to rate a governor in foot-pounds per second, implicitly, giving the three values of

1 Stroke of servomotor piston, in feet

2 Pounds pressure upon the servomotor piston, and

3 Time in seconds for completing one full piston stroke.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

The symbol for this correct rating may be written.

$$E = \text{ft-lb. per sec.} = SP/T_s \quad \ldots\ldots [1]$$

In order to properly discuss the essential features of a servomotor (or fluid pressure) governor, a brief analysis of its fundamental performance may be given, by referring to Fig. 3.

The essential elements of the governor proper are:

1 The flyballs, or any other speed-responsive means.

2 The control valve admitting fluid under pressure to one side of the servomotor piston and draining the fluid from the opposite side.

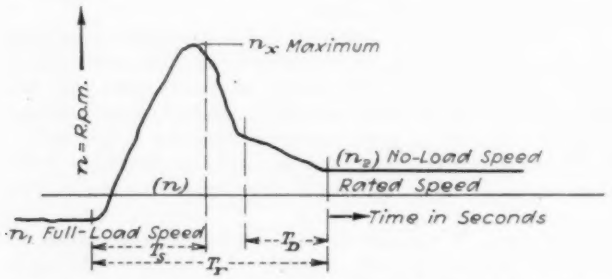


FIG. 2 CURVE OF TURBINE SPEED n AS FUNCTION OF TIME (Satisfactory governor action because of no oscillation. T_D is time of corrective action of secondary relay compensation.)

3 The relay or restoring mechanism which restores the control-valve piston F, to mid, or dead-beat position, so that the gate motion of the turbine again comes to a stop.

Space does not permit dealing with all the essential elements of the governor, but only with the control valve proper and the servomotor in relation to the control of the flow of water through the turbine, as affecting the speed of the unit and the pressure in the pipe line, if one is employed.

The process of regulation will be easily understood by referring to Fig. 3. Any speed rise

$$\Delta n = n_x - n_1 \quad \ldots\ldots [2]$$

causes the speed-responsive means to lift the floating lever ABC at joint B. Since the servomotor is in in its mid- or dead-beat position, C is temporarily a fixed point, so that the lever end A raises the control valve piston F, thereby admitting oil pressure p_a through port I, to the closing side of the servomotor piston E, at the same time allowing the oil from the opposite side of the piston to drain through port II. The servomotor piston then starts to move in the direction which closes the turbine gates, at the same time raising lever end C independently of any change of position of pivot B, so that end A is lowered and the control-valve piston finally restored to the mid- or dead-beat position in which both ports are closed, thereby completing what may be termed the initial and essential cycle of servomotor motion.

It is evident that if the servomotor can move sufficiently fast so that C will be raised almost immediately after B is raised, this will allow end A and along with it the control-valve piston to practically maintain its mid-position. In other words, the servomotor follows the motion of the speed-responsive means without delay, so that no over-regulation takes place; that is, there is no hunting or wave in the speed curve during the load change

$$\Delta N = N_x - N_1 \quad \ldots\ldots [3]$$

This kind of regulation is termed "aperiodic" (or waveless) regulation.

If, however, due to lack of flywheel effect, to intensity of load change, or to other causes which will be mentioned later, the speed rises so rapidly that the servomotor cannot follow synchronously, the control valve piston will be raised considerably, resulting in over-regulation, which produces one or more waves in the speed curve during, and even after, the period of load change.

The number of speed waves should be reduced to a minimum as this affects particularly the voltage of a hydroelectric unit. Therefore either the servomotor should be able to move the turbine gates as fast as possible in order to attain the aperiodic regulation, or the speed-responsive means should be so designed that it cannot move faster than at a rate which will allow the servomotor to follow synchronously.

A quick motion of servomotor may be permissible with turbine control which does not affect the operating head or pressure. This condition is met with an open-flume turbine setting having moderate velocities in the head and tail races and a short draft tube, or with impulse wheels where the jet is diverted from the buckets so that the flow of water in the pipe line is not affected. Where the control of the flow by the governor causes variations in the operating pressure of the turbine, the time (and the rate, as will be shown later) must be adjusted accordingly.

It is outside the scope of this paper to discuss in detail the means for preventing the flyball collar (at pivot B) from moving faster than the servomotor piston and relay. Briefly stated,

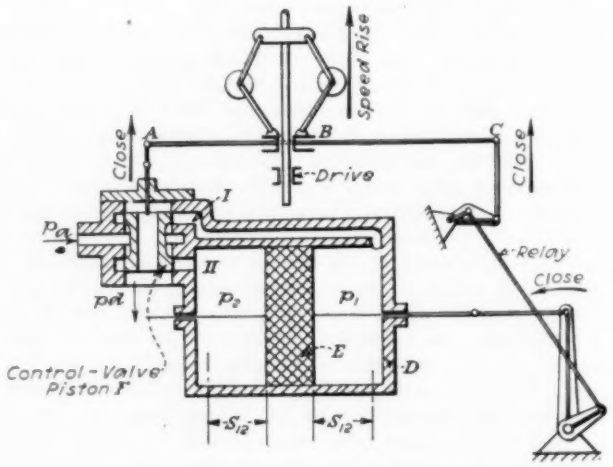


FIG. 3 DIAGRAMMATIC SKETCH OF GOVERNOR

this is accomplished either by employing highly static flyballs (i.e., having a large difference in speed between end positions of flyball stroke) with a so-called flexible relay or double compensation to bring about finally a closer range of speeds between no load and full load, or by using less static flyballs and a compensating device (counter dashpot) which forcibly retards the motion of the flyball collar B temporarily during the process of initial servomotor motion, and which really amounts to nothing more than a temporary increase of the static condition of the flyballs as in the first case. The principal mechanical difference or disadvantage is that the pivot joints of the flyballs and relay are put under a load in the second case, whereas they remain unloaded in the first and thus neither is sensitiveness impaired nor is there any excessive wear of the joints of the relay and floating lever.

GOVERNOR TIME T_s

This is to be understood as the time which the servomotor piston requires to complete a full stroke S, and thus represents

the time required for full stroke at a *constant*, average piston speed[2]

$$C = \frac{S}{T} \quad [4]$$

This time T does not furnish any indication as to the so-called rate of servomotor motion or piston speed ds/dt for intermediate portions of the piston stroke S, and which, as will be shown later, seriously enters into the problem of speed and pressure regulation.

The rate of motion of the servomotor piston is fixed by the quantity of fluid delivered per second under pressure to one side of the servomotor piston and drained from the opposite side. This may be more explicitly discussed by reference to Fig. 3.

Let S = total servomotor piston stroke, feet
y = lift of control-valve piston d, inches
p_a = fluid pressure available at inlet of control valve
p_d = discharge pressure at control valve
p = $p_a - p_d$ = total available fluid overpressure
p_1 = pressure on closing side of servomotor piston
p_2 = pressure on opening side of servomotor piston
$p_a - p_1 = p_{v1}$ = overpressure in port I
$p_2 - p_d = p_{v2}$ = overpressure in port II
D = diameter of servomotor piston, inches
A = area of servomotor piston, square inches

$$A = \frac{\pi D^2}{4} \quad [5]$$

d = diameter of control-valve piston, inches
a = area of control-valve piston, square inches

$$a = \frac{\pi d^2}{4} \quad [6]$$

f = port area of control valve, square inches

$$f = \pi d y \quad [7]$$

The control-valve piston being cylindrical and the ports I and II being equally closed in mid- or dead-beat position,

$$f_I = f_{II} = f \quad [8]$$

m = coefficient of velocity of the fluid under pressure
q = quantity, in cubic feet per second, of fluid under pressure passing through ports I and II
Q_D = total servomotor-piston displacement, cubic feet

$$Q_D = \frac{\pi D^2 S}{4 \times 144} = \frac{D^2 S}{183.34} \text{ cu. ft.} \quad [9]$$

V_1 = actual velocity at which the fluid under pressure passes through the control-valve port I, feet per sec.
V_2 = actual velocity at which the fluid under pressure passes through control-valve port II, feet per second.

$$q = \frac{Vf}{144} \quad [10]$$

Subscripts x designate values for intermediate servomotor piston strokes S_x.

The conventionally accepted rated energy E in foot-pounds is

$$E = p_a \times A \times S \quad [11]$$

[2] It will be seen later that the actual governor time T_g is the period from the beginning of the full-load change $\Delta N = N$ until the servomotor has completed its full stroke S, and that this time T_g is longer than T.

The effective energy per second at an intermediate position of the servomotor piston is:

$$E_x = \left(p_{1x} - p_{2x}\right) A \frac{dS_x}{dt} = (p_{1x} - p_{2x})\, AC_x \quad [12]$$

The corresponding displacement of fluid under pressure in cubic feet per second is:

$$q_x = \frac{A}{144} \times \frac{dS_x}{dt} = \frac{AC_x}{144} \quad [13]$$

However, the relation

$$C_x = C = \frac{S}{T} \quad [14]$$

obtains only provided the servomotor travels at a constant piston speed C, a condition which never prevails because the resistance offered by the gate mechanism of the turbine is not constant but varies over the entire stroke.

The quantities q_x therefore vary, and

$$Q = \frac{AS}{144} = \int_{t=0}^{t=t_s} q_x dt \quad [15]$$

The value of the displacement q_x can be expressed differently:

$$q_x = \frac{V_x f_x}{144} = \frac{V_x \pi d y_x}{144} \quad [16]$$

$$V_x = m\sqrt{2gh_x} \quad [17]$$

where

$$h_x = \frac{p_a - p_{x1}}{0.434} = \frac{p_{x2} - p_d}{0.434} \quad [18]$$

Since $f_1 = f_2$ and $p_{v1} = p_{v2}$,

$$V_x = \frac{m \times 8.02}{\sqrt{0.434}} \times \sqrt{p_a - p_{x1}} = 12.175\, m \sqrt{p_a - p_{x1}} \quad [19]$$

and

$$q_x = \frac{12.175}{144} \times \pi d \times m \times y_x \times \sqrt{p_a - p_{x1}}$$

$$= 0.2656\, dmy_x \sqrt{p_a - p_{x1}} \quad [20]$$

and since

$$p_a - p_1 = p_2 - p_d = \frac{p_a - p_d - (p_1 - p_2)}{2} \quad [18a]$$

and setting $p_a - p_d = p$, also $p_1 - p_2 = p_z$, so that

$$p_a - p_1 = \frac{p}{2}\left(1 - \frac{p_z}{p}\right) \quad [18b]$$

Equation [20] can be written:

$$q_x = \frac{0.2656}{\sqrt{2}} \times dm \sqrt{p}\; y_x \sqrt{1 - \frac{p_z}{p}} \quad [21]$$

or, regrouped,

$$q_x = \left[0.1878\, dm \sqrt{p}\right] y_x \sqrt{1 - \frac{p_z}{p}} \quad [21a]$$

The values d, m, and p are constant.

The total servomotor displacement in cubic feet is, according to [9],

$$Q_D = \frac{D^2 S}{183.34}$$

and the time T_s for a full stroke S is

$$T_s = \frac{Q_D}{q_x} \quad [22]$$

or

$$T_s = \frac{D^2 S}{183.34 \times 0.1878\, dm \sqrt{p}\, y_s \sqrt{1 - \frac{p_x}{p}}} \quad [23]$$

Regrouping and setting $y' = y/d$,

$$T_s = \left[\left(\frac{D}{d}\right)^2 \frac{S}{m} \frac{0.0290}{\sqrt{p}}\right] \frac{1}{y'} \frac{1}{\sqrt{1 - \frac{p_x}{p}}} \quad [24]$$

The values D, d, s, p, and m are constant for a fixed design and available overpressure of the servomotor, so that

$$T_s = R \frac{1}{y'} \frac{1}{\sqrt{1 - \frac{p_x}{p}}} \quad [24a]$$

Equation [24] contains all the factors involved in connection

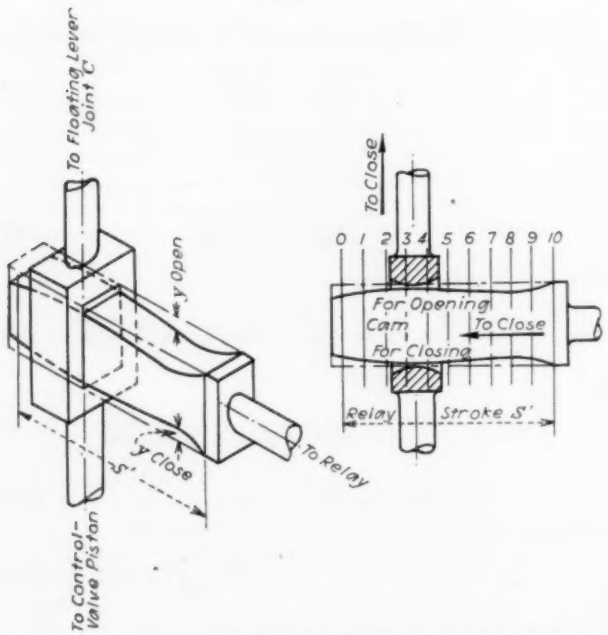


FIG. 4 DIAGRAMMATIC SKETCH OF RATE-LIMITING DEVICE (U. S. Pat.)

with the servomotor characteristic. From these equations the servomotor time T_s is seen to be:

1 Inversely proportional to the valve lift y since $y' = y/d$
2 Inversely proportional to the square root of the available fluid overpressure p
3 Inversely proportional to the *square* of the ratio d/D of the control-valve piston and servomotor piston
4 Inversely proportional to the square root of the ratio of actual overpressure p_x on the servomotor piston (due to resistance from the turbine gates) to the total available fluid pressure p.

These considerations lead to the following conclusions:

Case 1 To obtain largest possible valve lift y, the speed responsive means (flyballs) should be placed *between* the control valve and the relay.

Case 2 Increasing the available fluid pressure p shortens the servomotor time T_s, but only in the one-half power ratio of p.

Case 3 It is important not to use too small a control-valve diameter (d). Reducing this 50 per cent or else increasing the servomotor piston diameter D 100 per cent will increase the servomotor time T_s fourfold.

Case 4 Since p_x represents the overpressure ($p_1 - p_2$) on the servomotor piston necessary for overcoming the resistance of the turbine gates, it is evident that p_x will be <0 if the resistance acts in the direction of the respective servomotor piston motion (for instance, tendency to close gates when closing) and the value p_x/p becomes negative; thus $1/\sqrt{1 + p_x/p}$ shortens the servomotor time T_s.

For a fixed design R, Equation [24a] enables a direct comparison of servomotor time to be made for various intermediate gate positions. Let

$$\left.\begin{aligned} T_{sx1} &= R \frac{1}{y_1} \times \frac{1}{\sqrt{1 - \frac{p_{x1}}{p}}} \\ \text{and} \quad T_{sx2} &= R \frac{1}{y_2} \frac{1}{\sqrt{1 - \frac{p_{x2}}{p}}} \end{aligned}\right\} \quad [24b]$$

then

$$\frac{T_{sx_1}}{T_{sx2}} = \frac{y_2}{y_1} \sqrt{\frac{1 - p_{x2}/p}{1 - p_{x1}/p}} \quad [25]$$

For a constant valve lift $y_2 = y_1$,

$$\frac{T_{sx1}}{T_{sx2}} = \sqrt{\frac{1 - \frac{p_{x2}}{p}}{1 - \frac{p_{x1}}{p}}} \quad [25a]$$

or the servomotor times are in simple relation with their respective ratios of overpressure on the piston to available fluid pressure.

For a constant resistance on the turbine gates the value under the square-root sign is constant, and

$$\frac{T_{sx1}}{T_{sx2}} = \frac{y_2}{y_1} \times \text{constant} \quad [25b]$$

so that the servomotor times are inversely proportional to the respective valve lifts y_2 and y_1. This equation and the general Equation [24a] indicate *that for a design once selected any desired servomotor time T_s can be obtained by properly varying the valve lift "y."*

Since S represents the rate of servomotor piston speed, or the travel T_s per second,

$$\frac{S}{T_s} = \left[\left(\frac{d}{D}\right)^2 \frac{m\sqrt{p}}{0.02093}\right] y' \sqrt{1 - \frac{p_x}{p}} \quad [26]$$

or for a fixed design and fixed value of p,

$$\frac{S}{T_s} = R' y' \sqrt{1 - \frac{p_x}{p}} \quad [27]$$

indicating that by varying the valve lift "y," any desired rate of speed "S/T" of servomotor piston can be obtained; thus,

$$\frac{S}{T_{sx}} = R' y_x' \sqrt{1 - \frac{p_x}{p}} \quad [27a]$$

which applies to any gate position "x."

This can be most conveniently accomplished by a device (rate-limiting device, U. S. Patent No. 1,685,749, granted in 1928) which limits in either direction the stroke of the control-valve piston individually for any intermediate position of servomotor piston. In its simplest form (Fig. 4) it is a cam, operated in synchronism with the servomotor piston. The cam slides through a slot provided in the valve stem, and the surface above and below the cam can be properly determined in the field so as to obtain those valve piston lifts which are required for a desired rate of motion of the servomotor piston.

With such a device any desired rate of servomotor piston motion can be obtained; for instance:

1 A constant rate of piston motion, irrespective of the variable resistance on the turbine gates.

2 A rate which produces a constant rate of change of output of turbine, a feature which would simplify the calculation of momentary speed changes due to sudden change of load in open-flume turbines or impulse wheels, when not subject to pressure variations. All of which will be discussed more in detail in later paragraphs.

3 A rate which produces a constant rate of change of turbine discharge, and which is particularly desirable in cases where pipe-line problems are involved. (Direct control of the flow of water, by the adjustment of the guide vanes of a turbine, or the needle of an impulse wheel.)

4 A rate most suitable for automatic remote control of units—for instance, for *quickly* opening the turbine gates to a point which allows the unit to attain normal speed at no load, then slowing down the motion to permit of sufficient time for automatic synchronizing of the unit, then a quick opening of the gates to pick up full load. Also a desired rate for closing the gates to friction-load opening and a subsequent slowing down for avoiding a serious final pressure rise in the pipe line. Application of the rate-limiting device to control the rate of hydraulically operated butterfly valves, gate valves, etc., in a similar manner has also been successfully made in practice.

5 See Appendix No. 1.

For $p_1 - p_2 = 0$,

$$p_a - p_1 = p_2 - p_d = \frac{p_a - p_d}{2} \quad \text{[18c]}$$

In other words, when no resistance is offered by the turbine gates to the servomotor piston, then the entire pressure difference $p_a - p_d$ can be utilized for forcing the oil through the valve ports I and II, so that q becomes a maximum because, according to [19],

$$V_{max} = 12.175\ m \sqrt{\frac{p_a - p_d}{2}} \quad \text{[19a]}$$

Here $E_1 = E/T = 0$, since $E = PS = 0$, notwithstanding the fact that T is a minimum because $q = Q/T$ is a maximum; but because $p_1 - p_2 = 0$, $P = 0$.

Thus the power in foot-pounds per second is zero.

The other extreme $E_1 = 0$ which occurs when all the over pressure $p_a - p_d$ is necessary on the servomotor piston to resist the reaction from the turbine gates, and

$$p_1 - p_2 = p_a - p_d;\ p_a - p_1 = p_2 - p_d = 0$$

thus $q = 0$, which is the condition when the governor is unable to move the gates, so that $E_1 = 0$, because $T = \infty$ and $S = 0$ (no oil flowing), although $P = A(p_1 - p_2)$ is a maximum.

Between these two extremes $E_1 = 0$ there must be a maximum which represents the most efficient working conditions of the servomotor.

$$E_1 = \frac{E}{T} = (p_1 - p_2)\, A \frac{S}{T} = (p_1 - p_2) AC \quad \text{[28]}$$

since

$$A \frac{S}{T} = q = \frac{vf}{144}$$

Thus

$$E_1 = (p_1 - p_2)\ 0.264\ dmy \sqrt{p_a - p_1} \quad \text{[29]}$$

According to Equations [16], [17], and [18],

$$E_1 = 0.2656\ dmy\ (p_1 - p_2) \sqrt{p_a - p_1} \quad \text{[30]}$$

For a fixed design a given fluid, and for a fixed valve lift y, the values d, m, and y are constant; thus:

$$E_1 = C'(p_1 - p_2) \sqrt{p_a - p_1} \quad \text{[31]}$$

The power in foot-pounds per second is a function of the pressure. E_1 is a maximum when

$$p_1 - p_2 = \frac{2(p_a - p_d)}{3} = \frac{2p}{3} \quad \text{[32]}$$

so that[3] $p_a - p_1 = p_2 - p_d = \frac{p_a - p_d}{6} = \frac{p}{6}$

therefore

$$E_{1\,max} = C' \frac{2p}{3} \sqrt{\frac{p}{6}} = C' \times 0.272\ p^{3/2} \quad \text{[33]}$$

or, explicitly,

$$E_{1\,max} = 0.07226\ dmy\ p^{3/2} \quad \text{[34]}$$

Further,

$$\frac{2}{3} p A \frac{S}{T} = 0.072\ dmy\ p^{3/2} \quad \text{[35]}$$

or, since $A = \pi D^2/4$,

$$7.25\ D^2 \frac{S}{T} = dmy \sqrt{p} \quad \text{[36]}$$

This equation comprises all the essential values of the servomotor and its available fluid overpressure, for the most favorable rating as previously defined and for a constant resistance from the turbine gates.

However, when the resistance is variable the size of servomotor as determined from Equation [36] is not the most suitable. Again referring to Equation [25],

$$\frac{T_{ss1}}{T_{ss2}} = \frac{y_2'}{y_1'} \sqrt{\frac{1 - p_{s2}/p}{1 - p_{s1}/p}}$$

it can be seen that variable forces produce a variable ratio of p_s/p. The smaller the servomotor diameter D, the greater will be p_s, consequently the greater will be the variation in the time T_{ss}.

For example, when the resistance overpressure p_s opposes the servomotor motion in one case and assists it in another, then

$$\frac{T_{ss1}}{T_{ss2}} = \frac{y_2'}{y_1'} \sqrt{\frac{1 - p_{s2}/p}{1 + p_{s1}/p}} \quad \text{[25c]}$$

The ratio between the two times is very pronounced. For a servomotor size D as selected from Equation [36], i.e., where $p_s = 2p/3$,

[3] See author's article in *Die Turbine*, vol. 5, no. 21, Aug. 5, 1909.

$$\frac{T_{sx1}}{T_{sx2}} = \frac{y_2'}{y_1'}\sqrt{\frac{1-{}^2/_3}{1+{}^2/_3}} = \frac{y_2'}{y_1'}\sqrt{\frac{3}{3\times 5}} = \frac{y_2'}{y_1'}\times\frac{1}{2.236} \quad \ldots [25d]$$

Therefore, to obtain a constant servomotor time (or rate) the valve lifts y would differ 224 per cent.

A liberally dimensioned servomotor will in this case hold p_x so low that the ratio $\frac{T_{sx1}}{T_{sx2}}$ or $\frac{y_2'}{y_1'}$ will be closer to unity.

Conclusions. From the foregoing analysis it may be concluded that:

1 For widely variable resistance on the servomotor piston it is advisable to select a liberal size for D in order to avoid the pronounced variation of either valve lifts y or servomotor time T_{sx}.

2 The diameter D of the servomotor piston can be made smallest, and yet the largest amount of foot-pounds per second (E_1) obtained when the resistance forces between the turbine gates and servomotor piston are so transformed that they are nearly constant when acting upon the servomotor piston. (Selection of gate mechanism.)

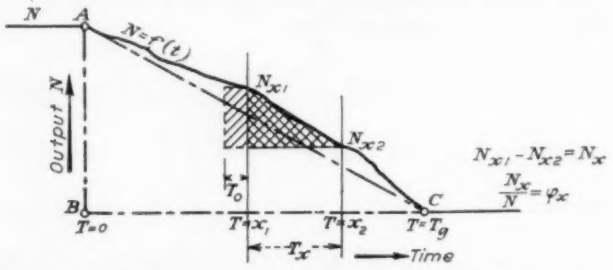


FIG. 5 CURVE SHOWING OUTPUT N DURING A CLOSING PERIOD OF THE SERVOMOTOR AS FUNCTION OF TIME T

Referring again to Equation [24a] and assuming a constant valve lift, we obtain $1/y' = \text{const.}$ Thus

$$T_s = \frac{R'}{\sqrt{1-\frac{p_x}{p}}} \quad \ldots [37]$$

This equation shows the general character of the relation between servomotor time and overpressure on the piston.

In Fig. 8 the times T_s are plotted for values of $R' = 1.7562$. To illustrate the influence produced by a change of either the control-valve diameter d, which does not affect the pressure p_x, or of the servomotor piston diameter D, which affects the pressure p_x, three additional T_s-curves T_{II}, T_{III}, and T_{IV}, are plotted.

T_{II} shows the values of T_s for a control-valve diameter half of that used in curve T_I. The servomotor times T_s shown are fourfold.

T_{III} shows the values of T_s for double the size of both the control-valve and the servomotor piston diameters. These values show the least variations between $p_x/p = \pm{}^2/_3$.

T_{IV} shows the values of T_s when the servomotor piston diameter is doubled, but not that of the control-valve piston. (Here the lower p_x improves the conditions over the most unfavorable ones, represented by Curve T_I, which has the same ratio of servomotor piston to control-valve piston.)

EFFECT ON SPEED AND PRESSURE CONTROL

A speed change of revolving masses is caused either by a surplus or by a shortage of energy, the surplus increasing the speed and the shortage reducing it. Fig. 5 illustrates in dash-dot line a sudden load rejection N. The servomotor closes the turbine gates at a certain rate so that the output of the turbine may be represented by the full line extending from $T = 0$, when the load was suddenly rejected, to $T = T_g$, when the turbine gates are closed [or closed to a point where the output of the turbine has become 0]. The shaded area below the curve $N = f(t)$ [a function of the time] is the surplus energy which accelerates the rotating masses.

If N were a straight line (the rate-limiting device arranged so that the output decreases at a constant rate $dN/dt = \text{const.}$ Case 3, on an earlier page) then we could correctly apply the well-known flywheel formula:

$$n_x^2 - n_1^2 = \frac{1{,}620{,}000\ (N_x - N_1)\ T_x}{WR^2} \quad \ldots [38]$$

where n_x = highest (or lowest) speed in r.p.m. caused by the load rejection (increase)

$\Delta N_x = N_x - N_1$

n_1 = initial speed at time the load change began

N_x = new load in horsepower

N_1 = load before load change

N = maximum rated full load of turbine

$\frac{N_x - N_1}{N}$ = relative load change = $\phi_x N$

T_x = time in seconds which the governor requires from the beginning of the load change $N_x - N_1$ to the completion of the servomotor piston stroke S_x necessary for adjusting the turbine gates to the new position

WR^2 = moment of inertia of revolving masses involved

For full-load change $N_x = N$, $\phi = 1$; $T_x = T_g$, we have:

$$n_x^2 - n_1^2 = \frac{1{,}620{,}000\ N\ T_g}{WR^2} \quad \ldots [39]$$

Equation [38] is the accurate flywheel formula. For moderate speed changes the well-known simplified formula $S_x = \frac{n_x - n_1}{n}$ can be used, namely,

$$S_x = \frac{800{,}000\ N_x T_x}{WR^2 n^2} = \frac{800{,}000\ \phi_x N T_x}{WR^2 n^2} \quad \ldots [40]$$

or, introducing the flywheel constant:

$$C = \frac{WR^2 n^2}{N} \quad \ldots [41]$$

$$S_x = \frac{800{,}000\ \phi_x T_x}{C} \quad \ldots [42]$$

When the rate of change of N is variable the hatched portion of Fig. 5 deviates from the triangular shape ABC, and its value should be figured more accurately.

$$\Delta E = \frac{NT_g}{2} \neq \int_{T=0}^{T=T_g} N dt \quad \ldots [43]$$

or generally,

$$\Delta E = \frac{(N_{x1} - N_{x2})\ (T_{x1} - T_{x2})}{2} \neq \int_{T=x_1}^{T=x_2} N dt \quad \ldots [44]$$

comprising only the double-hatched area of Fig. 5.

To correctly determine this energy change ΔE, it is necessary to know the character of the curve $N = f(t)$.

The governor time, T_g or $T_{x1} - T_{x2}$ in general, is composed of three essential periods:

$$T_g = T_0 + T_1 + T_2$$

(*a*) T_0 is the so-called dead time, or the time required from the beginning of the load change until the insensitiveness of the speed-responsive means is overcome and the point B (Fig. 3) begins to rise (or drop), thereby placing the control-valve piston out of mid or dead-beat position, and causing the servomotor piston to start moving. T_0 therefore depends upon the quality of the speed-responsive means employed (and to a small extent also upon the acuteness of speed change). T_0 = 0.1 to 0.5 sec., depending upon the quality of the governor. It is evident that this dead time T_0 adds to the energy change ΔE.

$$\Delta E_0 = T_0(N_{x1} - N_{x2}) \quad \ldots\ldots\ldots\ldots\ldots\ldots [45]$$

(*b*) T_1, the time from the beginning of the lift (or drop) of the control-valve piston to the point where the maximum lift Y is obtained. During the period $Y = 0$, to Y = maximum, the servomotor piston is accelerated.

(*c*) T_2, the time it takes the servomotor to complete its stroke S_x after the piston has attained maximum speed (rate of motion).

Since T_2 for Y = maximum is not constant (as previously discussed), we can combine T_1 and T_2 and consider them as the base $T = x_1$ to $T = x_2$ of Fig. 5.

We thus have:

$$\Delta E \text{ total} = T_0\left(N_{x1} - N_{x2}\right) + \int_{T=x_1}^{T=x_2} N dt \ldots\ldots [46]$$

and then obtain:

$$550\left\{\left(T_0 N_x + \int_{T=x_1}^{T=x_2} N dt\right)\right\} = \frac{M}{2}\left(V_x^2 - V_1^2\right) = \frac{WR^2}{2g}\left(\frac{\pi}{30}\right)^2 (n_x^2 - n_1^2). \quad [47]$$

or since

$$V = \frac{\pi D n}{60},$$

$$\frac{1}{N}\left(T_0 N_x + \int_{T=x_1}^{T=x_2} N dt\right) = \frac{1}{550}\left(\frac{\pi}{30}\right)^2 \frac{1}{2g}\frac{WR^2 n^2}{N} \times \left(\frac{n_x^2 - n_1^2}{n^2}\right) \ldots\ldots\ldots\ldots [48]$$

or since $N_x = \phi_x N$, n_1 = about n for close relay setting between no load and full load, and

$$\left(T_0\phi_x + \int_{t=x_1}^{T=x_2} \phi dt\right) = \frac{1}{3{,}232{,}991} C\left[\left(\frac{n_x}{n}\right)^2 - 1\right] \ldots [49]$$

or

$$\frac{3{,}232{,}991}{C}\left(T_0\phi_x + \int_{T=x_1}^{T=x_2} \phi dt\right) + 1 = \left(\frac{n_x}{n}\right)^2 \ldots\ldots [50]$$

n_x/n is the ratio of maximum speed n_x due to load change to the normal speed n. For instance, if

$$\frac{n_x}{n} = 1.25, \text{ the speed rise } 100\,\frac{n_x - n_1}{n} = 25 \text{ per cent.}$$

Thus

$$\frac{n_x}{n} = \sqrt{1 + \frac{3{,}232{,}991}{C}\left(T_0\phi_x + \int_{T=x_1}^{T=x_2} \phi dt\right)} \ldots\ldots [51]$$

This equation reduces to [39], when

$$\phi_x = 1,\ T_0 = 0,\ t_x = T_x,\ n_x = n_2, \text{ and } \int_{T=x_1}^{T=x_2} \phi dt = \frac{T}{2}$$

$$\left(\frac{n_2}{n}\right)^2 - 1 = \frac{3{,}232{,}991}{C}\left(0 + \frac{T}{2}\right)$$

or:

$$\frac{n_2^2 - n_1^2}{n^2} = 1{,}616{,}500\,\frac{T}{C} \text{ or } n_2^2 - n_1^2 = \frac{1{,}620{,}000\,NT}{WR^2}$$

From [51] we note at once that the value

$$3{,}232{,}991\,\frac{\left(T_0\phi_x + \int_{T=x_1}^{T=x_2} \phi dt\right)}{C}$$

is approximately the square of the actual relative speed change $\frac{n_{x1} - n_{x2}}{n}$.

The value $T_0\phi_x$ in [51] is independent of the action of the servomotor, namely, $0.1\phi_x$ to $0.5\phi_x$.

The value $\int_{T=x_1}^{T=x_2} \phi dt$ therefore characterizes the process of regulation.

Example. ϕ_x = 0.5 load rejection; T_0 = 0.2; C = 5,000,000 $\int_{T=x_1}^{T=x_2} \phi dt = \frac{0.5 \times 2}{2} = 0.5$ because assumed T_x = 2 sec., or constant rate of change of N.

Thus:

$$\frac{n_x}{n} = \sqrt{1 + \frac{3{,}232{,}991}{C}\left(T_0\phi + \int_{T=x_1}^{T=x_2} \phi dt\right)} = \sqrt{1 + \frac{3{,}232{,}991}{5{,}000{,}000}(0.1 + 0.5)}$$

$$\frac{n_x}{n} = \sqrt{1 + \frac{3{,}232{,}991 \times 0.6}{5{,}000{,}000}} = \sqrt{1 + 0.387954} = 1.174$$

$$\therefore \text{ speed rise } \frac{\Delta n}{n} = 17.4 \text{ per cent}$$

Illustrative Calculation

The variable resistance of the gates of a turbine reduced to the force P in pounds upon the servomotor piston along the servomotor stroke S may be indicated as shown in Fig. 6.

The total change in force P is 50,000 lb. The stroke S of piston is 1 ft. The gates are balanced (P = 0 at O—here at quarter-stroke from closed position). Assume further that

$p = p_a - p_d$ = 200 lb. per sq. in. oil overpressure
I = overpressure curve for gates in dead-beat position
II = overpressure curve for opening gates, and
III = overpressure curve for closing gates

To hold gates open, P_2 = 20,000 lb.
To hold gates closed, P_1 = 30,000 lb.
To close gates, P_2' = 16,500 lb. at wide-open position
To close gates, P_1' = 34,000 lb. at closed position
To open gates, P_2' = 23,500 lb. at wide-open position
To close gates, P_1' = 26,000 lb. at closed position

Closing Gates. At open position, P_2' = 16,500 lb., and at closed position, P_1' = 34,000 lb. Assume one servomotor cylinder with piston rod projecting through both covers, so that the piston displacement is the same on either side of piston,

$q_{v1} = q_{v2}$. Since generally it is not desired to close the gates at a maximum rate, toward the closed position, $P = 30{,}000$ lb. is taken as the force for figuring the most economical dimensions

$$p_1 - p_2 = 2/3\ p_s = 132 \text{ lb.}$$

$$\text{Piston area } A = \frac{30{,}000}{132} = 227 \text{ sq. in.;}\quad D = 17 \text{ in.}$$

The conventional rating of this servomotor would be

$$E = PS = 227 \times 200 = \mathbf{45{,}400\ ft\text{-}lb.}$$

The actual over pressure to produce $p_2' = 34{,}000$ lb. will therefore be

$$p_1 - p_2 = \frac{34{,}000}{227} = 150 \text{ lb., so that } p_s - p_1 = p_2 - p_4 = 25 \text{ lb.}$$

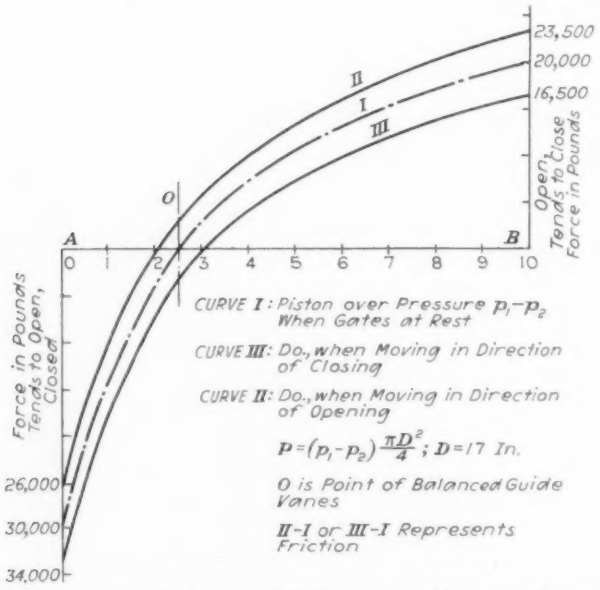


FIG. 6 ASSUMED FORCES ON SERVOMOTOR FOR OPERATING GUIDE VANES

Approaching Closed Turbine Gates. From Equation [20] we obtain the displacement per second in cubic feet.

$$q = 0.2656\ dmy\ \sqrt{p_s - p_1}$$

Assuming $d = 3$ in., $m = 0.45$ for oil, and $y = 0.25$ in.,

$$q = 0.2656 \times 3 \times 0.45 \times 0.25 \sqrt{25} = 0.448 \text{ cu. ft. per sec.}$$

The servomotor cylinder displacement is

$$Q_D = \frac{D^2 S}{183.34} = \frac{289}{183.34} = 1.5763 \text{ cu. ft.}$$

Thus the full-stroke time of the servomotor piston motion is

$$T_s = Q_D/q = C = 1.5763/0.448 = \mathbf{3.51\ sec.}$$

The servomotor piston approaches the end position for closed turbine gates at a rate of speed of 3.51 sec., or from [24],

$$T_s = \left(\frac{D}{d}\right)^2 \frac{S}{m} \frac{0.029}{\sqrt{p}} \frac{1}{y'} \sqrt{\frac{1}{1 - \frac{p_x}{p}}}$$

we obtain:

$$T_s = \left(\frac{17}{3}\right)^2 \frac{1}{0.45} \frac{0.029}{\sqrt{200}} \frac{3}{0.25} \frac{1}{\sqrt{1 - \frac{150}{200}}}$$

$$= 1.750 \times 2 = 3.51 \text{ sec. as before.}$$

Passing Through Position of Static Balance of Turbine Gates (Fig. 6, Point *O*).

$$P_0' = 3000 \text{ lb. for closing gates.}$$

Thus

$$p_1 - p_2 = \frac{P}{A} = \frac{3000}{227} = 13.2 \text{ lb. per sq. in.}$$

and

$$p_s - p_1 = \frac{200 - 13.2}{2} = 93.4 \text{ lb. per sq. in.}$$

Therefore

$$q = 0.2656\ dmy\ \sqrt{p_s - p_1} = 0.2656 \times 3 \times 0.45 \times 0.25 \times \sqrt{93.4} = 0.8662 \text{ cu. ft. per sec.}$$

or

$$T_s = 1.5763/0.8662 = 1.819 \text{ sec.}$$

or again,

$$T_s = 1.756 \sqrt{1 - \frac{13.2}{200}} = \frac{1.756}{0.9664} = 1.819 \text{ sec. as before.}$$

Starting From Wide-Open Gates. Here $P_2' = 16{,}500$ lb. assists in closing the gates. Therefore

$$\frac{16{,}500}{227} = 72.5 \text{ lb. per sq. in.}$$

is the pressure upon the servomotor piston in the closing direction and

$$p_1 - p_2 = -72.5$$

Since $p_1 - p_2 + (p_s - p_1) + (p_2 - p_4) = p_s - p_4$

and $p_s - p_1 = p_2 - p_4$,

$p_1 - p_2 + 2(p_s - p_1) = p_s - p_4$, and we obtain

$$p_s - p_1 = \frac{200 - (-72.5)}{2} = \mathbf{136.25\ lb.\ per\ sq.\ in.}$$

Further,

$$q = 0.2656 \times 2 \times 0.25 \times 0.45 \sqrt{136.34} = 1.047 \text{ cu. ft. per sec.}$$

whence

$$T_s = 1.5763/1.047 = \mathbf{1.50\ sec.}$$

Again,

$$T_s = 1.756 \frac{1}{\sqrt{1 + \frac{72.68}{200}}} = \frac{1.756}{1.167} = \mathbf{1.50\ sec.}\ \text{as before.}$$

The servomotor begins to close from wide-open turbine gates at a rate of 1.50 sec.

Fig. 7 shows the full-stroke servomotor time T_s plotted as a function of the 100 per cent servomotor stroke S.

Vice versa, *if a constant servomotor rate, C,* of say, $T_s = 2\frac{1}{2}$ sec. is to be obtained, the respective valve lifts y_x can be calculated for the three positions in question. Since

$$C = y_x \sqrt{p_s - p}$$

can be calculated and $p_s - p_1$ is a fixed value, we obtain the new required valve lift as follows:

$$\frac{Q_D}{2.5} = q_c = \frac{1.5763}{2.5} = 0.6305 \text{ cu. ft. per sec.} \ldots\ldots [52]$$

$$y_x = y\frac{q_c}{q_x}$$

For Position "A" Near Closed Gates.

$$y_A = 0.25\,\frac{0.6305}{1448} = \mathbf{0.351\ in.}$$

For Balance Point "O."

$$y_0 = 0.25\,\frac{0.6305}{0.866} = \mathbf{0.1819\ in.}$$

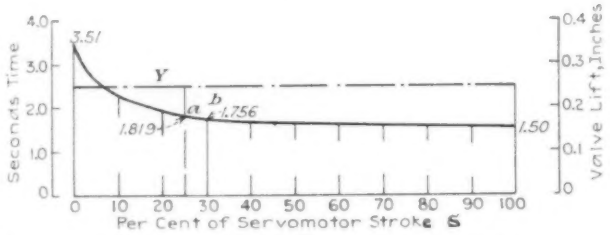


FIG. 7 SERVOMOTOR TIME CURVE AS FUNCTION OF PER CENT OF SERVOMOTOR STROKE

(Based on forces of Fig. 6 for constant valve lift $Y = 0.25$ in. *a*, Point of static balance of gates. *b*, Point of balance when closing.)

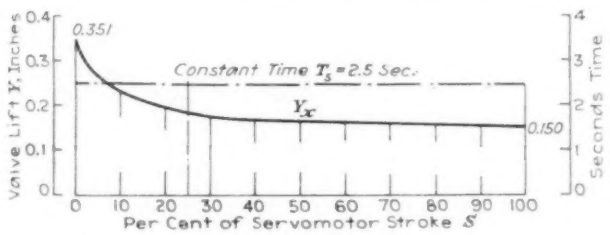


FIG. 7(*a*) CURVE OF VALVE LIFTS Y FOR CONSTANT RATE OF SERVOMOTOR MOTION

(Y_x = variable; $T_x = T_s/10$ = constant. Calculated for $T_s = 0.25$ sec.)

For Position "B" Starting From Open Gates.

$$y_B = 0.25\,\frac{0.6305}{1.047} = \mathbf{0.1502\ in.}$$

as illustrated in Fig. 7(*a*).

In order to obtain a rate of $2^1/_2$ sec. approaching closed gates, the valve lift would have to be increased from 0.25 in. to 0.351 in. This can be very readily accomplished with the rate-limiting device previously described. A constant rate of servomotor piston speed C [2.5 sec., Fig. 7(*a*)] as covered by Case 1 ante does not offer any advantages in regard to speed or pressure control of a turbine, because it results in individual values of $\int_{T=x_1}^{T=x_2} \phi dt$. An adjustment of the rate-limiting device so as to cause valve lifts y which produce a constant rate of change of output ΔN as outlined under Case 2 ante permits of integration of the values

$$\int_{T=x_1}^{T=x_2} \phi_x dt \text{ in Equation [47]}$$

In that case the curve $N = f(t)$ [Fig. 5] becomes a straight line, so that

$$\int_{T=x_1}^{T=x_2} \phi_x dt = \frac{\phi_{x1} - \phi_{x2}}{2}\left(T_{x1} - T_{x2}\right)$$

or for full-load change $= \dfrac{1 \times T_{ss}}{2}$

$T_{ss} < T_s$ because $N = 0$ at friction load N_0 of turbine produced at a stroke S_{rs} of the servomotor piston, and it will take another T_{rs} seconds to fully close the gates.

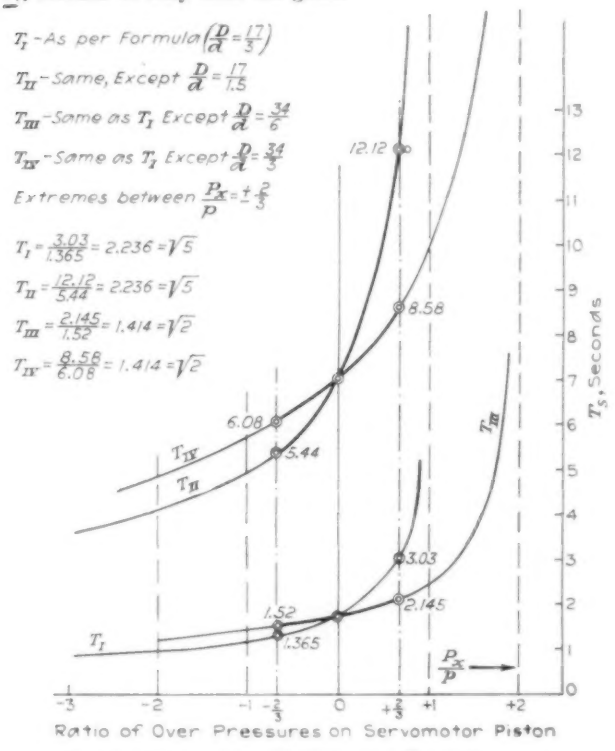


FIG. 8 COMPARISON OF SERVOMOTOR TIMES T_s

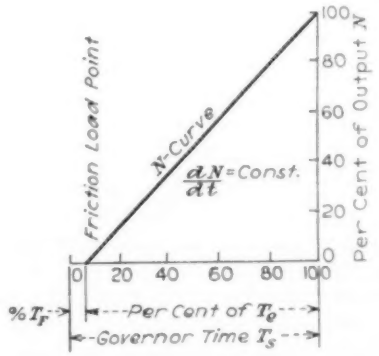


FIG. 9 OUTPUT (N) CURVE AS FUNCTION OF TIME T_s

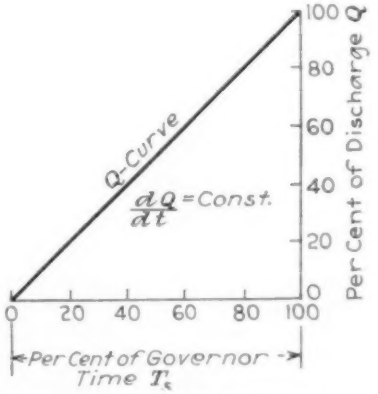


FIG. 10 DISCHARGE (Q) CURVE AS FUNCTION OF TIME T_s

$$\frac{n_x}{n} = \sqrt{1 + \frac{3{,}232{,}991}{C}\left[T_{0\phi_1} + \frac{\phi_1 - \phi_2}{2}(T_1 - T_2)\right]} \quad [53]$$

See also Equation [39] as per Fig. 9.

Where pipe lines are involved, the problem of pressure variations due to load changes becomes important.

An adjustment of the rate-limiting device whereby the servomotor piston travels at a rate which produces a straight-line discharge Q = const. will materially simplify the problem of pressure control. (Fig. 10.)

Here dQ/dt = constant = dv/dt, where v = velocity of water

in pipe line, and the rate of change of velocity remains constant over the whole range of discharge.

In Fig. 11 the output curve N and the efficiency e of the turbine are plotted as functions of the servomotor stroke S. The discharge Q figured from N and e is also plotted. The values of N and e are taken from an actual field test of a 50,000-hp. turbine operating under about 150 ft. head.

In Fig. 12 the full-stroke governor times T_s taken from Fig. 7 are used for obtaining graphically the part-stroke governor times T_x for each 10 per cent ordinate of the servomotor stroke S. The T_x-curve so obtained is again plotted (on a larger scale) in Fig. 13. A constant time rate $T_x' = CT_s$ is the straight line running from $T_s = 1.502$ to 0.

By projecting the 10 per cent points from the T_x' curve horizontally over to the T_x-curve, we obtain in the intersections the new ordinates which, projected to the abscissa line, will give

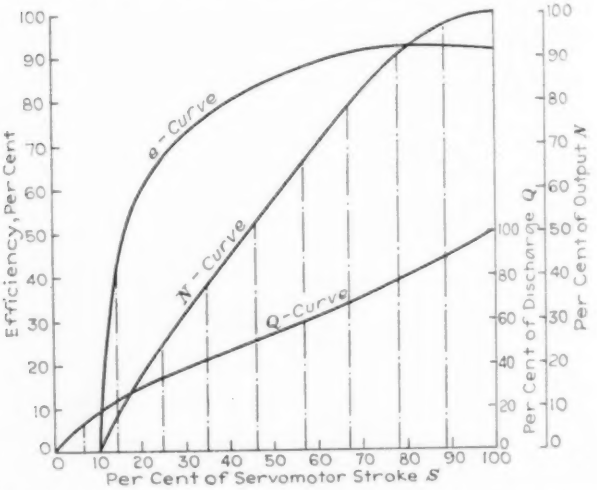


FIG. 11 POWER, EFFICIENCY, AND DISCHARGE CURVES AS FUNCTIONS OF SERVOMOTOR STROKE S

(Taken from actual field tests of a 50,000-hp. turbine at 150 ft. head. Dash-dot ordinates taken from Fig. 13.)

the new increments of per cent servomotor stroke corresponding to the new T_x-curve.

These new abscissa values S_x are plotted in Fig. 11 and furnish the corresponding ordinates in dash-dot lines for obtaining the values N and Q used in Fig. 14 on equal-time elements as abscissas. Thus N and Q are here shown as functions of time. It can be seen that with the exception of the tenth near full load, the N-curve is nearly a straight line, from which we conclude that under the conditions of the time curve, Fig. 7, an N-curve results which permits of almost correct application of the ordinary flywheel formula, if the effect of the dead time T_0 is neglected; or more explicitly, in Equation [51],

$$\int_{T=x_1}^{T=x_2} \phi dt = \frac{\phi_{x2} - \phi_{x1}}{2}\left(T_{x1} - T_{x2}\right)$$

It should be noted that zero load is reached before the servomotor piston attains the closed-gate position. The time to be used in the calculation of speed change is therefore shorter; $T_{s'} < T_s$, as shown in Fig. 14.

The Q-curve as a function of T indicates that the deceleration dv/dt or dQ/dt is greater at the closing end in spite of the fact that the servomotor moves at its slowest rate. The $Q = f(t)$ curve can be used as basis for calculating pressure rises. It will be seen that the pressure rise for changes ΔQ within the first tenth (0–1) of the servomotor stroke is even greater[4] than that caused by a full reduction Q.

The above curves are based on closing rates obtained with a constant valve lift $y = 0.25$ in. Vice versa, we can calculate the corresponding valve lifts for obtaining constant ΔN and ΔQ.

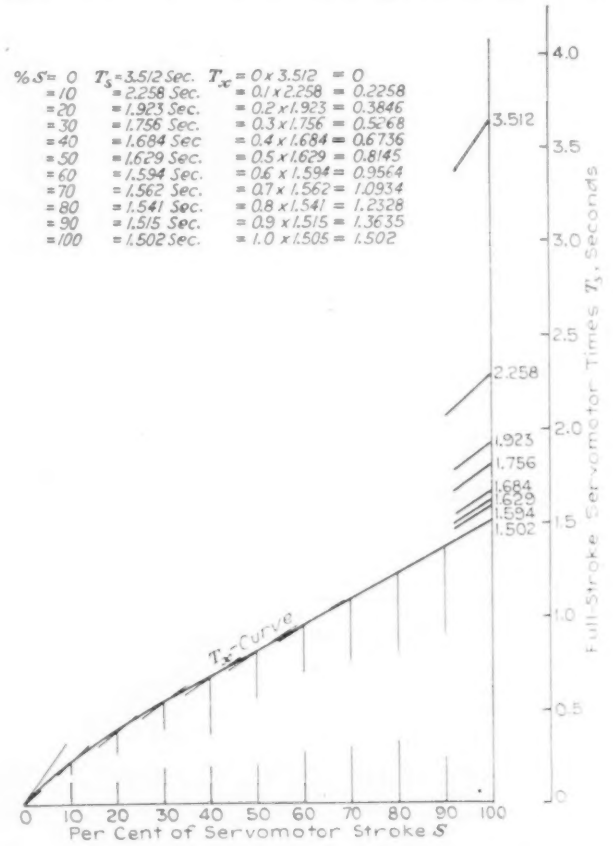


FIG. 12 TIME CURVE T_x (FOR CONSTANT VALVE LIFT $Y = 0.25$ IN.), DERIVED FROM SERVOMOTOR TIMES T_s OF FIG. 7, AS FUNCTION OF PER CENT OF SERVOMOTOR STROKE S

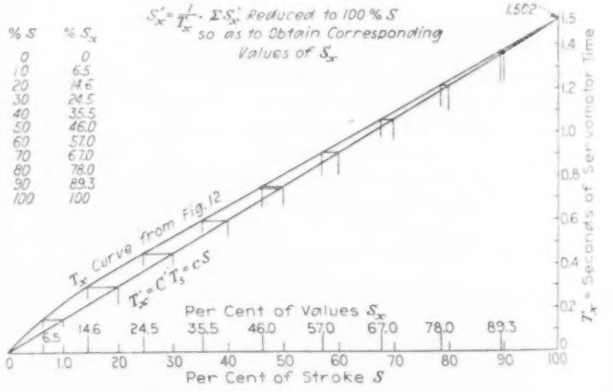


FIG. 13 PER CENT VALUES OF S_x DERIVED FROM T_x-CURVE OF FIG. 12 (ENLARGED SCALE)

In Fig. 14, the abscissas of tenths of N and Q are projected upon the base line, thereby giving those time elements $T/10$ which prevail. These are plotted in Fig. 15. The respective valve lifts are figured from

[4] See "New Aspects of Maximum Pressure Rise in Closed Conduits," by S. Logan Kerr. Trans. A.S.M.E., vol. 51 (1929), no. 6, paper HYD-51-3.

$$y_x = 0.25\frac{T_x'}{T_x} \quad \ldots\ldots\ldots\ldots\ldots\ldots [54]$$

where T_x is $T_s/10$ for the Q-curve and $T_{se}/10$ for the N-curve, and T_x' the respective increment of projection obtained for T_{xN} and T_{xq}, respectively.

The curve marked y_Q shows the valve lifts for a constant change of Q as a function of the time, dQ/dt = const., and the curve marked y_N represents the valve lifts necessary for obtaining constant change of N. The N-curve being almost a straight line requires practically a linear increase of valve lift, slightly increasing because dN/dt is a maximum at the first tenth. It requires a considerable increase toward the wide-open gate position on account of the pronounced deflection of the N-curve. The Q-curve, Fig. 14, indicates a maximum dv/dt at the first tenth and a minimum at the point of inflection about midway. This is evidenced by the upward bulge of the y_Q-curve. It is interesting to note that for the reaction forces P as used from Fig. 6, curve III, a steadily increasing valve lift y would be required for closing constant rate of servomotor piston motion. However, the character of the Q-curve used as the basis in Fig. 11, changes this picture decidedly.

The first practical application of the author's rate-limiting device, Fig. 4, was made in 1924 in connection with the single-jet, horizontal-shaft, double-overhung impulse wheels, rated 2 × 9000 b.hp. under 1260 ft. net head at 300 r.p.m. at the Nakatsugawa Plant No. 1, Japan. The pipe-line conditions (L, V, H,) are within that limit which permits the control of the flow Q, with direct actuation of the needle by the governor. [This being the first impulse-wheel plant where direct control of the flow by the governor was applied and good speed regulation obtained, credit is due the chief engineer of the company and owner of the plant, Mr. Kozo Tada, who accepted the author's conclusions that regulation can be obtained without relief valves or other temporarily water-wasting auxiliaries.]

Without the rate-limiting device the guarantee of 23 per cent maximum pressure rise was fulfilled with a servomotor closing time T_s of $2^3/_4$ sec. for full load thrown off. At smaller load changes, however, pressure rises as high as 33 per cent were observed without the rate-limiting device. The reacting forces on the needle are here the reverse of what was assumed in Fig. 6. The needle has a tendency to open at open position and to close at closed position. Here the time curve (Fig. 8) would be reversed for a constant valve lift y. The two surfaces of the cam (upper for opening needle, lower for closing needle) were experimentally determined at the plant, beginning with very small lifts y so as to hold the pressure variations down, and by increasing them (filing down on the cam surface) until the valve lifts y were such as to hold the permissible pressure variations for any needle opening or output of the units.

Practical Formulas for Speed Regulation

The customary flywheel formula [38], or its more general form [51], does not take into consideration the effect of the runaway of a turbine or hydroelectric unit. With no load taken from the turbine the speed attains a value n_r, which is a function of the output N developed under the operating head H. For full-load runaway n_r may vary between 1.7 n and 2.0 n, depending upon the runner characteristics and the windage and friction losses of the revolving parts of the unit. Since n_r is a limited value irrespective of the load change ΔN, a corresponding correction is applied, namely,

$$S' = \frac{S}{1 + \dfrac{S}{\dfrac{n_r}{n} - 1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [55]$$

Usually the value n_r is taken as the full-load value. This will result in too great a reduction of S' and can be corrected by introducing an average value

$$\frac{n_{rx1} + n_{rx2}}{2}$$

the runaway speeds corresponding to the turbine gate openings of the output, N_{x1} and N_{x2}, before and after the load change.

The corrective effect of Equation [55] becomes more pronounced as the speed change increases, for $S = \infty$, $S' = n_r$, and $S - S^* = \infty$.

The above formula does not take into consideration the

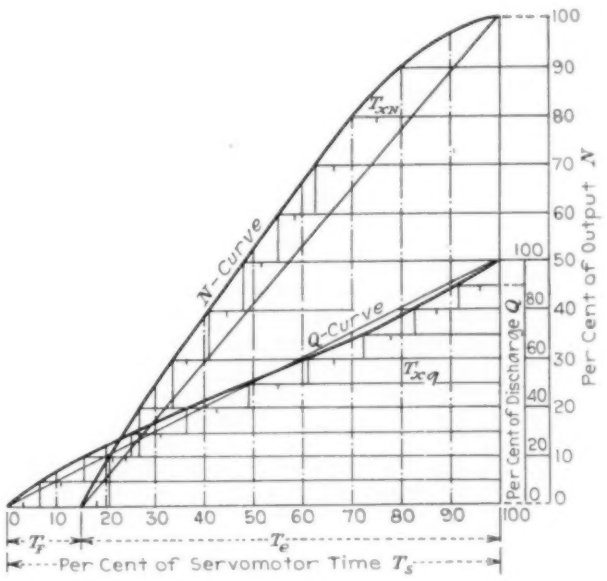


Fig. 14 Power and Discharge Curves N and Q Replotted From Fig. 11 as Functions of Time T_s
(Ordinates taken from dash-dot values of Fig. 11 as obtained from Fig. 13.)

change of efficiency at variable speed and its effect upon N, which, however, for practical purposes of calculation can be neglected.

Pressure Variations and Their Effect on Speed. Any pressure change Δp or change of head ΔH affects the output of a turbine in the ratio of the $^3/_2$ powers of the respective heads, neglecting again the small influence of the variable efficiency if the speed remains constant.

$$\frac{N}{N'} = \left(\frac{H}{H'}\right)^{3/2}$$

If the pressure rise ΔH were instantaneous and remained constant over the entire process of regulation, for practical purposes we could set

$$S^* = S'\left(1 + \frac{\Delta H}{H}\right)^{3/2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [56]$$

The value S^* so obtained exceeds the actual values, if the pressure rise does not prevail over the entire process of regulation. It is therefore safe for practical purposes, although mathematically not sufficiently accurate. With a similar degree of accuracy we can set

$$S_0 = S\left(1 - \frac{\Delta H}{H}\right)^{3/2} \quad \ldots\ldots\ldots\ldots\ldots\ldots [57]$$

as a corrective factor for load increases and resulting pressure decreases. Here the corrective effect due to runaway does not enter into the problem.

Calculations of Pressure Rises in Pipe Lines

The methods of determining $\Delta H/H$ have been liberally discussed in the literature, reference being particularly made here to Mr. S. Logan Kerr's recent paper, "New Aspects of Maximum Pressure Rise in Closed Conduits."[4] This valuable paper, for the first time, points to a critical pressure rise at a point of servomotor position, when a complete closure of the turbine gates is attained by the governor in the time interval $T_r = 2L/a$. In what follows it will be shown that the simple formula cited by Minton Warren in his paper entitled "Penstock and Surge Tank Problems," presented in October, 1914, before the American Society of Civil Engineers, also permits of directly figuring this critical pressure rise.

Warren's general formula is:

$$\frac{\Delta H}{H} = \frac{LV}{gH}\frac{1}{\left(T - \frac{L}{a}\right)} \quad \ldots\ldots\ldots\ldots\ldots\ldots [58]$$

where L = total length of pipe line, ft.
V = full-load velocity of water, in ft. per sec.
g = 32.2 ft. per sec. per sec.
a = velocity of wave, ft. per sec.
H = head, ft.
ΔH = increase in head, ft.
$\frac{\Delta H}{H}$ = relative pressure rise
T_s = servomotor time for full stroke, i.e., full change of velocity V.

For $T = 2L/a$, Warren's formula changes into

$$\frac{\Delta H}{H} = \frac{aV}{gH} \quad \ldots\ldots\ldots\ldots\ldots\ldots [58a]$$

which represents the maximum instantaneous relative pressure rise for full-load rejection in T_s seconds, when $T_s \leqq 2L/a$. This point, however, is not the critical point referred to in Mr. Kerr's paper.

By rewriting Warren's formula, we obtain:

$$\frac{\Delta H_x}{H} = \frac{L V_x}{gH\left(T_x - \frac{L}{a}\right)} \quad \ldots\ldots\ldots\ldots\ldots [58b]$$

as applying to an intermediate gate position at which the water velocity is V_x and the servomotor time T_x.

By setting $V_x = V_m \psi_x$ and $T_x = T_s \varphi_x$ we can write:

$$\frac{\Delta H_x}{H} = \frac{LV_m}{gHT_s}\frac{\psi_x}{\varphi_x}\frac{1}{1 - \frac{L}{a\varphi_x T_s}} \quad \ldots\ldots\ldots\ldots\ldots [59]$$

It will be noted that this value of relative pressure rise $\Delta H_x/H$ is composed of three elements, namely,

(1) $\frac{LV_m}{gHT_s} = A$, a design and operating constant, so to speak. It may be split up further into

$\frac{L}{H}$, ratio of total length of penstock to head,

$\frac{1}{g}$, inversed acceleration,

$\frac{V_m}{T_s}$, which represents the "average" rate of total change of velocity for the total time T_s.

V_m is that average full-load velocity which is obtained as follows:

$V_m = \frac{\Sigma LV}{L}$ and may be termed the average kinetic velocity.

(2) $\frac{\psi_x}{\varphi_x} = \frac{dv}{dt} = \tan\alpha$, the rate of change of velocity of water at the various points of the velocity curve V as a function of the time T_s. Here α represents the angle of the tangent of the velocity-rate curve having φ as abscissa and ψ as ordinate.

(For $\frac{\psi_x}{\varphi_x} = \tan\alpha = 1$ ($\alpha = 45°$), or a straight-line velocity curve, as covered by requirements 1 and 2 outlined at the beginning of this paper and shown also by curve, Fig. 10.)

(3) The value containing the individual time element

$$\frac{1}{1 - \frac{L}{aT_s\varphi_x}} \text{ or } \frac{1}{1 - \frac{B}{\varphi_x}}, \text{ where } B = \frac{L}{aT_s}$$

again a design and operating constant.

It is evident that formula [59] is general and contains as a special case $\psi_x/\varphi_x = 1$, the condition of a straight velocity curve.

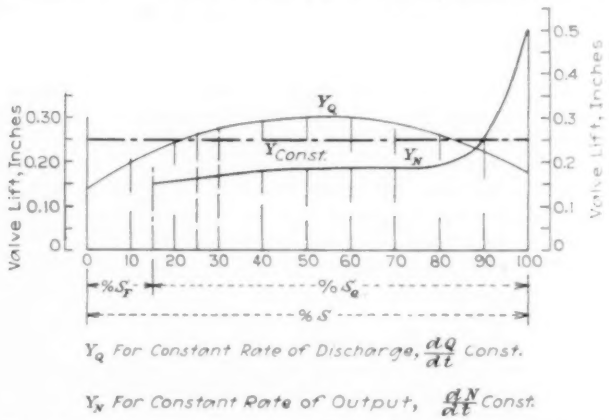


Fig. 15 Valve-Lift Curves as Functions of Per Cent of Servomotor Stroke Figured From Abscissas Taken From Fig. 14

We conclude from the above curve that—

(*a*) The relative pressure rise $\Delta H/H$ is directly proportional to the design and operating constant LV_m/gHT_s.

(*b*) It is directly proportional also to the individual rate of change of velocity $\psi_x/\varphi_x = dv/dt = \tan\alpha$ over the whole range of the curve $V = f(t)$. These rates must average the total value V_m/T_s, therefore an angle $> 45°$ within this range must also cause an angle $< 45°$.

(*c*) It is an inverse function of the individual time element φ_x, viz.:

$$\frac{1}{1 - \frac{B}{\varphi_x}}$$

The critical pressure rise $\Delta H_r/H$ referred to in Mr. Kerr's paper is obtained at once from Warren's general formula by using the author's transformed formula with the restrictions 1 and 2 cited before, i.e., $\psi_x/\varphi_x = 1$; then from [55],

$$\frac{\Delta H}{H} = \frac{LV_m}{gHT_s}\frac{1}{1 - \frac{L}{a\varphi_x T_s}} \quad \ldots\ldots\ldots\ldots [59a]$$

by setting $T_x = \frac{2L}{a}$; or since $T_x = \varphi_x T_s$ or $\varphi_x = \frac{2L}{aT_s}$,

$$\frac{\Delta H_r}{H} = \frac{LV_m}{gHT_s}\frac{1}{1-\frac{LaT_s}{2LaT_s}} = \frac{2LV_m}{gHT_s} \quad \ldots\ldots [60]$$

Fig. 16 shows for a numerical example the percentage relative pressure rises $\Delta H_x/H$ on the basis of a discharge curve Q as shown in Fig. 10 (i.e., $\psi_x/\varphi_x = 1$) as calculated from formulas [59a] and [60] with the following numerical values:

$L = 2000$ ft.
$V_{(max)} = 10$ ft. per sec.
$H = 800$ ft.
$a = 3000$ ft. per sec.

$\frac{2L}{a} = \frac{2 \times 2000}{3000} = 1.33 = T_r$ (critical time)

The values of the percentage pressure rises $\Delta H_x/H$ are plotted for servomotor time T_s of 10, 5, 3, 2, and $1^1/_2$ sec.

For $T_s = 1.333 = 2L/a$ a maximum pressure rise of 116.4 per cent will occur.

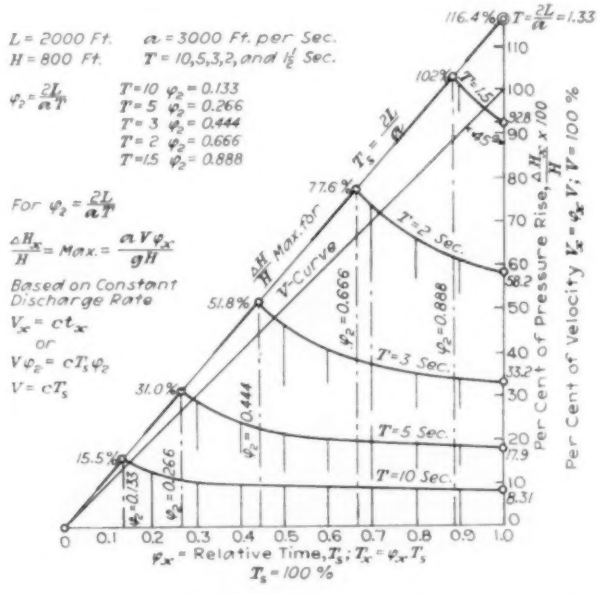


FIG. 16 PRESSURE-RISE CURVES $\Delta H_x/H$ AS FUNCTIONS OF φ_x OR PER CENT OF TIME T_s

For a total time $T_s = 5$ sec., for instance, we obtain, for $\varphi_x = 1$,

$$\frac{\Delta H_x}{H} = \frac{2000 \times 10}{32.2 \times 800 \times 5}\frac{1}{1-\frac{2000}{3000 \times 5 \times 1}} = \mathbf{0.179} \text{ or } 17.9\% \quad \ldots\ldots [60a]$$

The critical pressure rise $\Delta H_r/H$ is obtained by setting $T_x = 2L/a$ or $\varphi_x = 2L/aT_s$, or directly from [60]:

$$\frac{\Delta H_r}{H} = \frac{2LV}{gHT_s} = \frac{2 \times 2000 \times 10}{32.2 \times 800 \times 5} = \frac{10}{32.2} = \mathbf{0.31} \text{ or } 31\% \ldots [60b]$$

If the total governor time T_s were $2L/2$, we should obtain

$$\frac{\Delta H_r}{H} = \frac{2LVa}{gH2L} = \frac{aV}{gH} = \frac{3000 \times 10}{32.2 \times 800} = \mathbf{1.164} \text{ or } 116.4\% \ldots [60c]$$

as before. The maximum pressure rise occurs at $\varphi_x = 1$, $T_s = 2L/a$, in general formula [59].

$$\frac{\Delta H_x}{H} = \frac{LV}{gHT_s}\frac{1}{1-\frac{L}{a\varphi_x T_s}}$$

$$= \frac{2000 \times 10 \times 300}{32.2 \times 800 \times 2 \times 2000}\frac{1}{1-\frac{2000 \times 3000}{3000 \times 2 \times 2000}} \quad \ldots\ldots [59b]$$

which reduces to $\frac{3000 \times 10}{32.2 \times 800}$ as before.

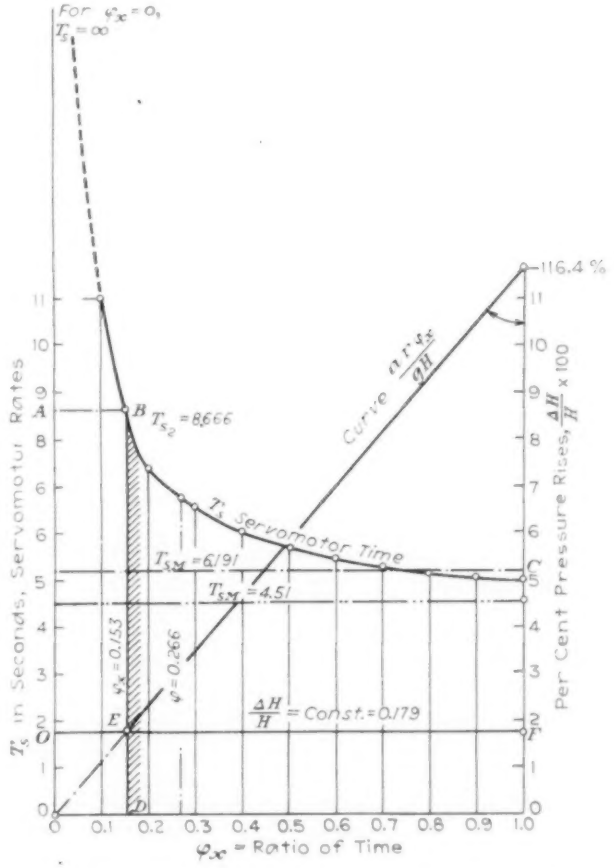


FIG. 17 SERVOMOTOR TIME T_s FOR $\Delta H/H = 0.179$; dv/dt = CONST.

Again we can write from general formula and for $T_s = 2L/a$,

$$\frac{\Delta H_x}{H} = \frac{LV \times a}{gH \times 2L}\frac{\psi_x}{\varphi_x}\frac{1}{1-\frac{1}{2\varphi_x}} = \frac{aV\psi_x}{2gH}\frac{2}{2\varphi_x - 1} \quad \ldots [60d]$$

or for $\varphi_x = 1$,

$$\frac{\Delta H_x}{H} = \frac{aV\psi_x}{gH} \quad \ldots\ldots [61]$$

or in the example,

$$\frac{\Delta H_x}{H}\frac{2000 \times 10 \times \psi_x}{32.2 \times 800} = 1.164\,\psi_x \quad \ldots\ldots [61a]$$

Formula [61] represents in general the critical momentary-pressure-rise curve. It is a straight line as shown in Fig. 16.

Again referring to formula [59a] where $\psi_x/\varphi_x = 1$ (straight-line discharge, Fig. 10),

$$\frac{\Delta H}{H} = \frac{LV}{gHT_s}\frac{1}{1-\frac{L}{a\phi_x T_s}} = \frac{LV}{gH}\frac{1}{\left(T_s - \frac{L}{a\phi_x}\right)} \quad [59c]$$

we obtain the rates of servomotor time T_s for constant $\Delta H/H$:

$$T_s = \frac{LV}{gH\frac{\Delta H}{H}} + \frac{L}{a\phi_x} \quad [62]$$

a curve of the hyperbola class $(y - M)N_x = \text{const.}$ Applied to our numerical example for $\Delta H/H = 0.179$,

$$T_s = \frac{2000 \times 10}{32.2 \times 800 \times 0.179} + \frac{2000}{3000\,\phi_x} = 4.333 + \frac{0.666}{\phi_x} \quad [63]$$

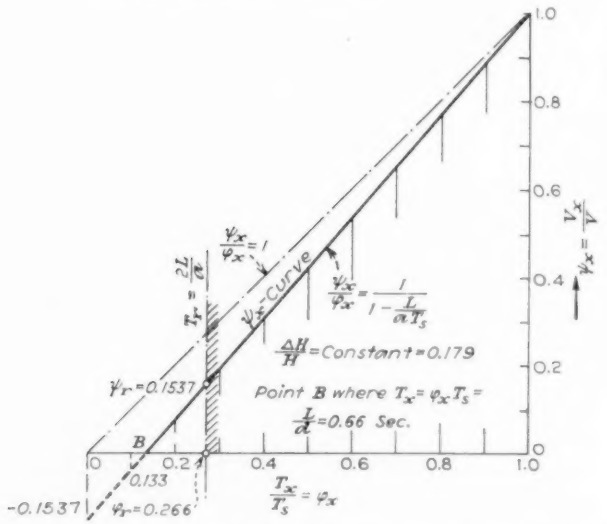


FIG. 18 VELOCITY CURVE ψ_x AS FUNCTION OF TIME ψ_x FOR CONSTANT PRESSURE RISE. $\Delta H/H = 0.179$; $T_s = 5$ SEC.

Here, for $\phi_x = 1$	$T_s = 5$
$= 0.9$	$= 5.066$
$= 0.8$	$= 5.18$
$= 0.7$	$= 5.284$
$= 0.6$	$= 5.44$
$= 0.5$	$= 5.666$
$= 0.4$	$= 6.0$
$= 0.3$	$= 6.555$
$= 0.266$	$= 6.833$
$= 0.2$	$= 7.33$
$= 0.153$	$= 8.666$
$= 0.1$	$= 11.0$
$= 0$	$= \infty$

This T_s-curve is plotted in Fig. 17 for $\phi_x = 0.153$, $T_s = 8.666$, and $T_x = T_s\phi_x = 1.33$ sec. $= 2L/a$, and according to [59b],

$$\frac{\Delta H}{H} = \frac{2000 \times 10}{32.2 \times 800}\frac{1}{8.66 - \frac{0.666}{0.153}} = 0.179 \quad [59d]$$

or at the former critical servomotor time we now obtain the desired 17.9 per cent instead of 31 per cent, by increasing the rate T_s from 5 sec. to 8.66 sec.

The critical time ratio ϕ_x for $\Delta H/H = 0.179$ is obtained from

$$\frac{aV\psi_x}{gH} = \frac{\Delta H}{H} \quad [59e]$$

and since $\psi_x = \varphi_x$,

$$\phi_x = \frac{0.179 \times 800 \times 32.2}{3000 \times 10} = 0.153 \quad [59f]$$

as shown in Figs. 16 and 17, or a servomotor rate

$$T_s = 4.333 + \frac{0.666}{0.153} = 8.666\ (59\,\text{g.})$$

This is also obtained directly from [60]:

$$T_{sr} = \frac{2LV}{gH\frac{\Delta H}{H}} = \frac{2 \times 2000 \times 10}{32.2 \times 800 \times 0.179} = 8.66 \text{ sec.} \quad [60e]$$

The momentary pressure rise of 17.9 per cent is obtained with a servomotor rate of 8.666 sec. and occurs at 0.153 of that total rate. From this point on we can close at a constant rate of $T_s = 8.666$, whereby the pressure rise $\Delta H/H$ drops in a straight line to the zero point, as shown by the dash-dot line in Fig. 17, or we can close suddenly, $T_s = 0$, whereby the pressure rise $\Delta H/H$ remains at its maximum value of 17.9 per cent, i.e., remains constant over the entire range of servomotor motion, as shown in full line in Fig. 17.

By integrating the T_s-curves and dividing by the length of the base line ϕ_x, we obtain the actual average rate T_s, or about 6.191 sec. for rates ABC with $\Delta H/H$ along lines FEO, or 4.51 sec. for rates DBC with $\Delta H/H$ along lines FEG.

From the above it follows that in order to hold the pressure rise constant the servomotor rates T_s must become longer and longer until the critical point B of the curve, Fig. 17, is reached.

The velocity curve V_x or ψ_x, as a function of the time, is obtained from formula [59], where $V_x = V\psi_x$.

$$\psi_x = \frac{\frac{\Delta H_x}{H}gHT_s}{LV}\left(\phi_x - \frac{L}{aT_s}\right) \quad [64]$$

or

$$V_x = V\psi_x = \frac{\frac{\Delta H_x}{H}gHT_s}{L}\left(\phi_x - \frac{L}{aT_s}\right) \quad [64a]$$

Thus, for a desired, constant pressure rise $\Delta H/H$ and for a fixed governor time T_s, we obtain

$$\psi_x = A\,(\phi_x - B) \quad [64b]$$

where

$$A = \frac{\frac{\Delta H_x}{H}gHT_s}{LV} \quad [64c]$$

and

$$B = \frac{L}{aT_s} \quad [64d]$$

This equation represents a straight line:

For

$$\psi_x = 0,\ \phi_x = -B = \frac{-L}{aT_s} \quad [64e]$$

$$\phi_x = 0,\ \psi_x = -AB \equiv \frac{\frac{\Delta Hx}{H}gHT_sL}{LVaT_s} \quad [64f]$$

or

$$\psi \equiv \frac{\frac{\Delta H_x}{H}gH}{Va} \quad [64g]$$

Applied to the numerical example for $\Delta H/H = 0.179$, $T_s = 5$ sec.,

$$\psi_x = \frac{0.179 \times 32.2 \times 800 \times 5}{2000 \times 10}\left(\phi_x - \frac{2000}{3000 \times 5}\right) \quad [64h]$$

$$\psi_x = 1.15276\ (\phi_x - 0.133) \quad [64i]$$

For

$$\left.\begin{aligned} \psi_x &= 0, \quad \phi_x = 0.133 \\ \phi_x &= 0, \quad \psi = -0.1537 \end{aligned}\right\} \text{ as plotted in Fig. 18.}$$

Again referring to formula [59] we can write

$$\frac{\Delta H_x}{H} = A'\left(\frac{\psi_x}{\phi_x}\right)\frac{1}{1 - \frac{B'}{\phi_x}} = \frac{A'\psi_x}{\phi_x - B'} \quad [65]$$

In Fig. 10, the rate dv/dt is constant and equal to 1. The governor reduces equal increments of Q (or v).

For values dv/dt or $\psi_x/\varphi_x < 1$, the pressure rise $\Delta H_x/H$ decreases. This may be better illustrated by reference to Fig. 19:

Curve I represents a constant rate: dv/dt or $\psi_x/\varphi_x = 1$.

Curve III represents also a constant rate but steeper: $\psi_x/\varphi_x > 1$.

Curve II has a variable rate dv/dt starting steep for the first three-tenths of V or Q.

For curve I,

$$\frac{\psi_x}{\phi_x} = \frac{1.0}{1.0} = 1.0 = 45°$$

For curve III,

$$\frac{\psi_x}{\phi_x} = \frac{1.0}{0.5} = 2.0 = 26\tfrac{1}{2}°$$

For curve II,

$$\frac{\psi_x}{\phi_x} = \frac{0.3}{0.15} = 2.0 = 26\tfrac{1}{2}°$$

for the three-tenths of curve under discussion.

For the first three-tenths of reduction of discharge we obtain a time of $T_s' = T_s \times \frac{1.5}{3} = 50$ per cent of the former time T_s.

We thus obtain from [58]

$$\frac{\Delta H_x}{H} = \frac{LV}{gH} \times \frac{1}{\left(T_s - \frac{L}{a}\right)} = \frac{2000 \times 10}{32.2 \times 800}\,\frac{1}{(0.5 \times 5 - 0.666)}$$

$$= 0.423 \quad [58c]$$

or 42.3 per cent as against 17.9 per cent, for constant rate dv/dt of curve I; or figured from the general formula [59],

$$\frac{\Delta H_x}{H} = \frac{LV\psi_x}{gHT_s'\phi_x}\,\frac{1}{1 - \frac{L}{a\phi_x T_s}}$$

we obtain for $\psi_x = 1$ and $\phi_x = 0.5$, as against curve I before:

$$\frac{\Delta H_x}{H} = \frac{2000 \times 10 \times 1.0}{32.2 \times 800 \times 5 \times 0.5}\,\frac{1}{1 - \frac{2}{3 \times 0.5 \times 5}} = 0.423$$

as before, and which is between the 3.5-sec. and 2-sec. curves of Fig. 16.

In the case of curve II the velocity decreases somewhat slower below the 70 per cent ordinate. The rate, however, starts with the same value of ψ_x/φ_x as in curve III, so that the pressure rise remains the same.

Since the pressure rise $\Delta H/H$ is a function of the rate of change of velocity, it is evident that a maximum pressure rise occurs when dv/dt or ψ_x/φ_x assumes the value $\dfrac{1.0}{\phi_x = \dfrac{2L}{aT_s}}$

so that in our numerical example:

$$\frac{\Delta H}{H} = \frac{LV}{gH}\frac{\psi_x}{\phi_x}\frac{1}{T_s}\,\frac{1}{1 - \frac{L}{a\phi_x T_s}} = \frac{2000 \times 10 \times 1 \times 3000 \times 1}{32.2 \times 800 \times 2000 \times 2\,(1 - \frac{1}{2})}$$

$$= 1.164 \text{ or } 116.4 \text{ per cent}$$

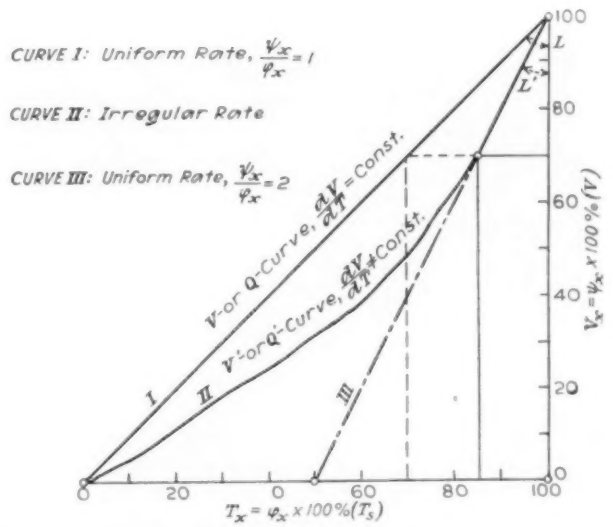


FIG. 19 DISCHARGE OR VELOCITY CURVES ILLUSTRATING CHARACTERISTICS OF RATES OF VELOCITY AS FUNCTIONS OF TIME

The important criteria to be looked out for in practice are the following:

1 The pressure rise for the last increment of complete closing motion, where T_s becomes $2L/a$, so that the momentary relative pressure rise

$$\frac{\Delta H_r}{H} = \frac{aV\psi_x}{gH} \text{ or } \frac{2LV}{gHT_s} \text{ is reached.}$$

2 The pressure rise for that portion of the closing process, where $dv/dt = \psi_x/\varphi_x$ is a maximum, and at which instants T_s' may become equal to $2L/a$

CONCLUSIONS

It can be readily seen that the problem of pressure and speed control of turbines supplied by water through closed conduits is primarily affected by the character of the discharge curve as a function of the governor time, i.e., by the rate of change (dv/dt) of velocity in the pressure supply pipe.

This rate of change of velocity in turn is fixed by the rate of motion of the turbine gates actuated by the servomotor of the governor.

The rate of motion of the servomotor in turn is fixed by the resistance of the turbine gates and by the character and action of the control valve and the speed-controlling means (or flyballs) actuating it.

All of the above elements participating in the process of

control of flow embrace individualities of action, which necessitates, as outlined in the beginning, the application of the numerous assumptions for the purpose of simplifying the mathematical problem.

It has been proved that any desired rate of change of discharge can be obtained with the rate-limiting device which provides means for obtaining that lift of control valve which is necessary to produce the desired rate of gate motion for each individual gate position.

With the possibility of so establishing in advance the character of the discharge curve, the problem becomes a surprisingly simple one, as regards calculation of pressure variations.

The critical part-gate governor time $T_r = \phi_r T_s$ for a desired maximum pressure rise

$$\frac{\Delta H_r}{H} = \frac{2LV}{gHT_s}$$

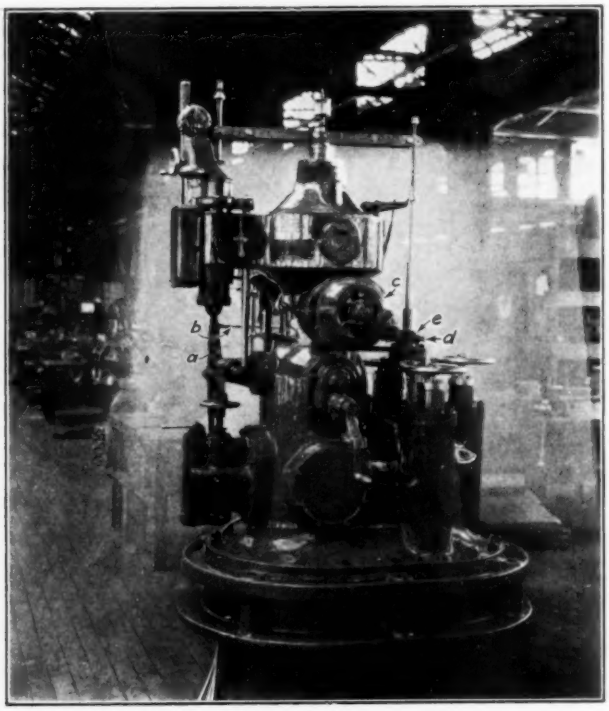


FIG. 20 TURBINE GOVERNOR ACTUATOR EQUIPPED WITH RATE-LIMITING DEVICE

for straight-line discharge, or, more explicitly, $\phi_r = 2L/aT_s$ can be calculated and the corresponding speed rise investigated.

For a limited pressure rise $\Delta H_r/H$ and a fixed head H and length L of pipe line, the governor time T_s or the size of the pipe line (V) must be selected accordingly. If for lack of flywheel effect the speed rise becomes too high for a predetermined size of pipe, then auxiliary means in the form of pressure regulators or surge tanks (to reduce ΔV in the first case and L in the latter) must be provided, or the WR^2 must be increased in order to secure a longer governor time T_s.

With the above values properly selected, it becomes a simple matter of adjusting the rate-limiting device in the field by proper determination of the maximum individual lifts y_x of the control-valve piston as previously explained. This once determined by giving the cam of the "rate-limiting" device the proper shape, assures the simplest and safest solution of the otherwise rather intricate problem. Instead of going through a series of tedious calculations, the results of which, owing to the necessity of making numerous assumptions, cannot be absolute, and only too frequently not even approximate, we can determine the surface of the cam by gradually increasing the valve lifts y_x until the desired limits of $\Delta H_x/H$ at any gate opening are obtained, a procedure which also takes considerably less time in the field than is required for making actual calculations.

The same procedure can also be applied equally effectively for independently producing opening rates satisfactory for commercial operation of the unit. The particular benefit derived from the application of the rate-limiting device is appreciated when one considers that a very slight modification of the time curves *A*, *B*, and *C*, Fig. 11, of Mr. Kerr's paper referred to before, causes differences in pressure rise of 10 per cent and even as high as 100 per cent. An application in practice is shown by Fig. 20, which is a view of an Allis-Chalmers governor actuator equipped with a rate-limiting device. The relay motion from the servomotor is transmitted to lever *a*, which actuates rod *b* and a bell-crank lever *c* with universal joint (not visible on picture because directly behind the lower right-hand part of the electric motor driving the flyballs). The bell crank *c* is link-connected to the cam *d* which can be seen projecting through the front of the control-valve cover *e*. This design is, of course, one only of many different designs which, however, accomplish the same purpose.

Appendix No. 1

EXCESSIVE pressure variations in a conduit must be avoided because of the limited strength of its material. The upper limit or maximum pressure which the conduit can resist without permanent injury is therefore fixed. It is obvious that if a conduit serves a turbine operating under a variable head, the margin between operating pressure and maximum pressure becomes smaller as the operating head approaches its maximum value. Logically, the governor closing time should then be longer in order to hold the pressure rise down so that the peak of pressure does not exceed the permissible value.

$$H + \Delta H = H_{max} \quad [66]$$

$$H + a + (\Delta H - a) = H_{max} \quad [67]$$

In the first case $\Delta H/H$ is the permissible relative pressure rise. In the second case it is

$$\frac{\Delta H - a}{H + a} < \frac{\Delta H}{H} \quad [68]$$

Similarly for a lower operating head,

$$\Delta H - a + (\Delta H + a) = H_{max} \quad [68a]$$

or

$$\frac{\Delta H + a}{\Delta H - a} > \frac{\Delta H}{H}$$

To increase the total (!) servomotor time T_s, the oil supply to the control valve is manually throttled in some installations. This reduces the possible relative pressure rise. However, in the case of a load increase immediately following the rejection, it prevents a relatively rapid opening.

W. P. Creager, vice-president and chief engineer of the Northern New York Utilities, Inc., has suggested (U. S. Patent applied) a manometric device whereby a pressure increase in the penstock throttles the oil supply and augments it accordingly as the pressure drops. Such a device will meet the requirement of Equations [66] and [67] applying to load rejections as such, and also to load increases as such, but it will not prove efficient in the case of a sudden reversal of the load change, where quick

action is desired at the very beginning of the reversal. If the oil supply to the control valve of the governor is built double-ported, with an individual supply to each port chamber, it will be possible by inserting a flat distributor gate or the like, operated by the manometric device, to increase the oil supply to the opening side and at the same time reduce it to the closing side, and vice versa.

The application of the rate-limiting device in addition to the suggested double-ported valve device will preserve the individual character of the change of rate of gate motion as a function of gate position.

The two requirements can be simultaneously met by transmitting the manometric action to the cam of the rate-limiting device in such a manner that an increase of pressure in the penstock will shift the cam surface producing the closing motion so as to reduce the individual valve lifts, and at the same time shift the opening cam surface so as to permit of greater individual valve lifts, the character of each rate curve being retained.

Discussion

Byron E. White.[5] The subjects embraced by the two papers by Mr. Arnold Pfau and Mr. Forrest Nagler merge to such an extent that a discussion of one to a certain degree involves the other. These excellent papers, together with the other papers presented at the Hydraulic Division sessions, indicate that the power industry has now grown up from childhood to manhood. The earlier installations experienced troubles which may be very aptly compared to the growing pains of youth. These papers attack some of the remaining aches and pains of this now adult phase of engineering and offer some very valuable and constructive information.

As between the two diametrically opposed schools of thought—namely, that which considers sensitive governing equipment as a prime essential for every unit and that which advocates the elimination of governors and the substitution of other means of speed control or of frequency control, such as the devices of Kerr and Warren, which school is aptly represented by Mr. Nagler—the writer is inclined to take a middle position. He is already on record to this effect, in a report entitled "Governor Problems" published by the Hydraulic Power Committee of the National Electric Light Association. To him it seems that one criterion is the existence of an isolated load, which may, either by design or by accident, be dependent for speed control upon a particular unit or units. If such is the case, a reasonably good speed-control governor is a prime essential. If, on the other hand, the plant or unit is feeding into an interconnected system and is so connected to it that when its service is interrupted it can by no possibility have an important isolated load directly dependent upon it for its frequency control, then in many instances it has been found entirely satisfactory to have the gate movement controlled by a speed-limiting or a shut-down device, lacking the sensitiveness of the usual type of hydraulic-turbine governor.

The advent of widespread interconnections between regions and plants embracing immense power capacities has made it entirely possible to throw the burden of speed control upon one or a few plants in each system. There are inherent advantages, both in single-plant control and in multiple-plan control. Fortunately, both of these systems may be operated simultaneously, in parallel, without undue voltage or frequency disturbance or undesirable interchange of power.

It is probable that frequency control will tend to take on the characteristic of multiple-plant control, because of the immensity of the systems gradually being interconnected. While each of the unit systems might have its frequency controlled by a single station or unit, when the entire system is viewed as a unit, the overall control of frequency will, in fact, be governed by a multiplicity of plants.

Intimately tied in with the control of frequency is the question of pressure rises in penstocks, which has been admirably treated by Mr. Pfau. The crux of his paper is of course the cam which he has developed, which effectually limits the rate of travel of the governor piston and consequently of the guide vanes so as to limit the pressure rises which may take place in the penstock at any portion of the stroke.

While some of the discussion centered upon the accuracy and application of the Warren formula for pressure rise, utilized by Mr. Pfau, it seems to the writer that this is but a mere side issue, since he has pointed out that at two critical points this formula coincides with that described by Mr. Kerr last year, and in the last analysis the intent of the paper is to describe a new and ingenious mechanical means of limiting such pressure rises the shape of which is determined by experiment and does not depend upon the formula except to demonstrate its action and its possibilities.

Mr. Pfau's device adds another means of accomplishing this desirable result to the others which already have been adopted and has the further advantage that its effect may be made truly variable at different portions of the governor stroke in accordance with the particular requirements which it is designed and desired to fulfil.

E. B. Strowger.[6] With reference to the mechanics of the rate-limit device, its application to the modern hydraulic governor, and the relation of this device to the rate of turbine-gate closure, the paper is to be highly commended. In the treatment accorded the subjects of pressure rise and speed regulation, however, the author has considered approximations which are not rational and which may introduce serious errors in connection with many installations.

If dN/dt equals a constant during the closure of the turbine gates, then there must be no water hammer during the closure, and consequently there must be no physical length of penstock and draft tube for this condition. The author's Equations [38], [39], [40], and [42] therefore cannot be used for accurate results. Equation [51], however, is fundamentally correct. The quantity $\int_{T=x_1}^{T=x_2} \phi dt$ in this equation should be evaluated by taking cognizance of the total excess energy transferred from the penstock to the rotating parts, as already explained in the joint article by Mr. Kerr and the writer on "Speed Changes of Hydraulic Turbines for Sudden Changes of Load."[7] With a physical length of penstock and draft tube it is impossible to justify an evaluation of this quantity by assuming a linear relation between N and T_2, as shown in Fig. 9.

The use of Warren's approximate equation would of course simplify the solution of water-hammer problems somewhat, but unfortunately its use is precluded because it does not generally apply. This has been conclusively shown by N. R. Gibson[8] and by R. S. Quick.[9] The error introduced by the approximation may be as high as 100 per cent, as shown by Quick. In particular reference to Fig. 16 of the paper, the only values

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[7] *Trans. A.S.M.E.*, vol. 48, p. 209.

[8] "Pressures in Penstocks Caused by the Gradual Closing of Turbine Gates," *Trans. A.S.C.E.*, vol. 83.

[9] "Comparison and Limitations of Various Water-Hammer Theories," *Trans. A.S.M.E.*, vol. 49, p. 524.

correctly given are those of the critical pressure rise labeled ΔH_r in the text. These values are plotted on the straight line which is the locus of the limiting values of pressure rise, viz., $\frac{aV}{g}$. Fig. 21 gives the correct pressure-rise curves $\Delta H_z/H$ as functions of φ_z, or per cent of time T_s for the problem given in connection with Fig. 16. As shown from this analysis, the following table gives the error in the author's use of the Warren approximation applied to this particular example, considering

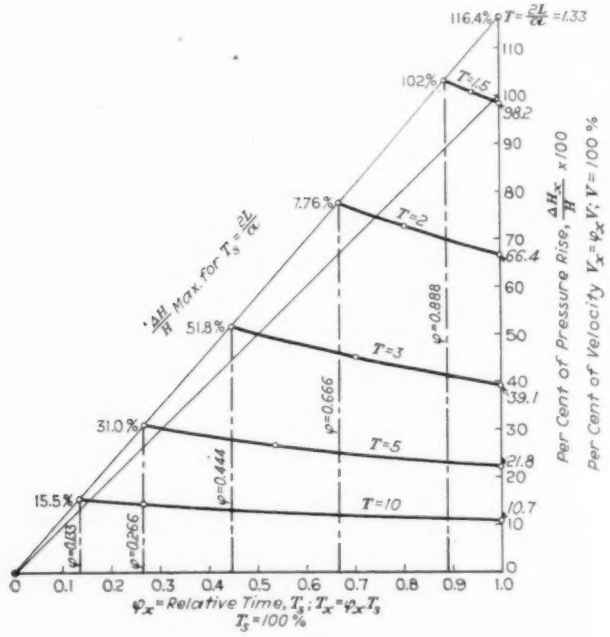


FIG. 21

the closures from $\varphi = 1$, and shows that the errors involved are as high as approximately 20 per cent:

	Pressure Rise $\Delta H_z/H$		
T_s	Pfau, %	Correct value, %	Error, %
10	8.31	10.7	22.4
5	17.9	21.8	17.9
3	33.2	39.1	15.0
2	58.2	66.4	12.3
1.5	92.8	98.2	5.5

The author intimates in his conclusions that numerous assumptions are frequently made which may be "not even approximate" and that tedious calculations are necessary for the refined methods of computing pressure rise. With reference to assumptions made in refined work and their effect upon the problem, reference is made to the article already referred to, by Kerr and Strowger. In this article it was pointed out that the calculations made for the South Falls Plant and for the Queenston plant of the Hydro Electric Power Commission of Ontario checked the actual test results remarkably closely both as regards speed rise and as regards pressure rise. The computations were based upon an analysis of the various elements of the problem which involved no assumptions except perhaps that due to a change in runner efficiency due to the speed characteristic of the turbine.

Regarding the tedious calculations necessary for obtaining pressure rise and speed rise, the writer quite agrees with Mr. Pfau that the arithmetic integration process is somewhat tedious in the case of problems involving a number of intervals of $2L/a$ seconds. However, since publishing the 1926 paper already referred to, on "Speed Rise of Hydraulic Turbines for Sudden Changes of Load," the writer has used the method of computation suggested by Quick and in this connection wishes to present the following description of this method, trusting in so doing that it will prove useful to others who have not as yet taken the time to study it. By changing from the arithmetic-integration method to that of Quick, no change in the fundamental principles is involved. The older method of arithmetic integration may be preferred by some engineers since it may be carried out by an ordinary slide rule, whereas for the other method a calculating machine or a six-place table of logarithms is necessary. However, the latter method eliminates the cut-and-try process.

Although problems in speed regulation of hydroelectric units may not assume in general the importance today which they formerly held when electric systems were more or less isolated, yet there are many special cases which involve a solution. For example, on the Niagara system due to certain tripping operations a large hydro unit may suddenly be isolated from a long-distance load and may hold on a small isolated local load or may be tripped completely from both loads. In either case there is little or no connected Wr^2 to help regulate the speed.

The change in speed in the case of a load change involving the isolation of the unit from the system is a function of the energy changes in the penstock accompanying the load change and the characteristics of the turbine runner. Thus the following factors enter the problem:

(1) Amount of the load change
(2) Head on turbine
(3) Governing time
(4) Penstock velocity
(5) Length of penstock
(6) Velocity of propagation of the pressure wave in the penstock
(7) Wr^2 of rotating parts of unit
(8) ϕ curves of turbine runner.

In the case of a load change where the unit does not separate from the system, then the connected Wr^2 of the system is effective in limiting the speed change.

The calculation consists essentially of three parts: first, the determination of the increase or decrease of head due to the load change; second, the calculation of the speed change, considering the energy changes in the penstock, but neglecting the change in turbine efficiency due to the speed change; and, third, calculating the final value of speed change, considering the runaway characteristic of the turbine. This last part of the problem can usually be omitted as the correction involved is small, as will be shown later.

The calculation of speed change uncorrected for the runaway characteristic involves the use of the formula:

$$\frac{Wr^2}{5870}(N_2^2 - N_1^2) = \Sigma \text{hp.} \times \Delta t \times 550$$

or

$$N_2 = \sqrt{\frac{3{,}229{,}000 \Sigma \Delta t \cdot \text{hp.} + Wr^2 \times N_1^2}{Wr^2}}$$

where

Δt = the time of one interval $2L/a$, sec.
N_1 = the speed at the beginning of the interval, r.p.m.
N_2 = the speed at the end of the interval, r.p.m.
hp. = average horsepower developed by turbine for the interval
W = weight of the rotating mass, lb.
r = radius of gyration, ft.

TABLE 1 COMPUTATION OF RISE IN PRESSURE FOR DETERMINATION OF SPEED REGULATION

Full load thrown off unit, 70,000 hp.
Head (H_o), 213.5 ft.
Governor time, 5 sec.
Uniform motion of regulating piston
Initial penstock velocity (V_o), 10.25 f.p.s.
Penstock length, 690 ft.
Penstock diameter, 21 ft.
Velocity of pressure wave (a), 4680 f.p.s.

$$h_{max.} = \frac{4680 \times 10.25}{32.2} = 1489.75$$

$$K = \frac{h_{max.}}{2H_o} = \frac{1489.75}{2 \times 213.5} = 3.4889$$

N (interval)	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	16.96
t (sec.)	0.0000	0.2949	0.5897	0.8846	1.1795	1.4744	1.7692	2.0641	2.3590	2.6538	2.9487	3.2436	3.5385	3.8333	4.1282	4.4231	4.7179	5.000
ϕ_n	1.0000	0.9660	0.9170	0.8710	0.8210	0.7690	0.7140	0.6560	0.5940	0.5290	0.4630	0.3990	0.3320	0.2670	0.2000	0.1320	0.1065	0.000
2Σ	0.0000	0.0000	0.0186	0.0400	0.0852	0.1354	0.1890	0.2464	0.3078	0.3740	0.4446	0.5178	0.5896	0.6647	0.7373	0.8131	0.8897	
$1-2\alpha$	1.0000	1.0000	0.9814	0.9600	0.9148	0.8646	0.8110	0.7536	0.6922	0.6260	0.5554	0.4822	0.4104	0.3353	0.2627	0.1869	0.1103	
$\phi 2/\pi K$	3.4889	3.2154	2.9338	2.6468	2.3517	2.0632	1.7786	1.5014	1.2310	0.9763	0.7479	0.5554	0.3846	0.2487	0.1396	0.06079	0.01474	
$1-2\Sigma+\phi 2/\pi K$	4.4889	4.2154	3.9152	3.6068	3.2665	2.9278	2.5896	2.2550	1.9232	1.6023	1.3033	1.0376	0.7950	0.5840	0.4023	0.2477	0.1250	
$2K(1-2\Sigma)$	6.9728	6.9778	6.8480	6.6987	6.3833	6.0330	5.6590	5.2585	4.8300	4.3681	3.8755	3.3647	2.8637	2.3397	1.8331	1.3042	0.76965	
$1+\phi 2/\pi K^2$	13.1724	12.2181	11.2357	10.2345	9.2047	8.1983	7.2055	6.2382	5.2949	4.4063	3.6094	2.9379	2.3417	1.8678	1.4869	1.2121	1.05143	
$1+2K(12-2\Sigma)+\phi 2/\pi K^2$	20.1502	19.1959	18.0837	16.9332	15.5880	14.2313	12.8645	11.4967	10.1249	8.7744	7.4849	6.3026	5.2054	4.2075	3.3200	2.5163	1.82108	
$\phi\sqrt{1+2K(1-2\Sigma)+\phi 2/\pi K^2}$	4.4889	4.2091	3.8995	3.5842	3.2414	2.9010	2.5609	2.2243	1.8901	1.5670	1.2667	1.0017	0.7574	0.5477	0.3644	0.2094	0.08771	
P_n	0.0000	0.0093	0.0157	0.0226	0.0251	0.0265	0.0287	0.0307	0.0331	0.0353	0.0366	0.0359	0.0375	0.0363	0.0379	0.0383	0.0373	
$h_n = 1489.75\ P_n$ ft.	0.0000	13.8500	23.3900	33.6700	37.3900	39.9300	42.7600	45.7400	49.3100	52.5900	54.5200	53.4800	55.9100	54.0800	56.4600	57.0400	55.5700	

Table 1 gives the computation for the pressure rise in the penstock due to water hammer for a 5-sec. closure of the gates of a 70,000-hp. unit with penstock length of 690 ft. and an initial penstock velocity of 10.25 ft. per sec., using Quick's method of computation. Uniform gate motion has been used in this particular case, although it is quite a simple procedure to obtain the actual relation between time and servomotor piston stroke for any closure and by means of the relation between guide-vane opening and servomotor-piston stroke to apply the actual discharge-time curve in determining the values of ϕ_n, where ϕ_n is the proportion of initial gate factor ϕ_0 as defined by Quick.

Table 2 gives the calculation of the speed rise with no correction for the runaway characteristic of the turbine. Columns 1, 2, 3, 4, 5, 6, and 7 are obtained from the data of Table 1. Column 8 is obtained from the efficiency curve on Fig. 22 labeled *A*, which curve is based upon an actual field test of the turbine. The rest of the table is computed in the regular way in accordance with the paper already referred to. The answer of 39.0 per cent speed change appearing as the last figure in column 16 is the approximate answer and would be correct if the characteristics of the water wheel did not change during the transient speed change.

To take account of the runaway characteristic of the turbine, Table 3 has been prepared, the first seven columns of which are the same as the corresponding columns in Table 2. The revised efficiency values of column 8 are based upon the efficiency curve labeled *B* in Fig. 22, which curve has been moved from curve *A* by an amount determined from the test of a homologous 36-in. model runner. Although in the case of the example in hand the change in efficiency is of small moment and is rather erratic due to the shape of the runner ϕ curves, as shown in Fig. 23, yet the example will be worked through to a conclusion to serve as an illustration. Here ϕ is the peripheral velocity coefficient of the wheel.

From the results of Table 2 the head values and the approximate speed values for each interval (or, as in Table 4, for every two intervals) are recorded and the corresponding ϕ values computed. Knowing the relation between the gate opening on the model and the main-unit gate opening, revised values of

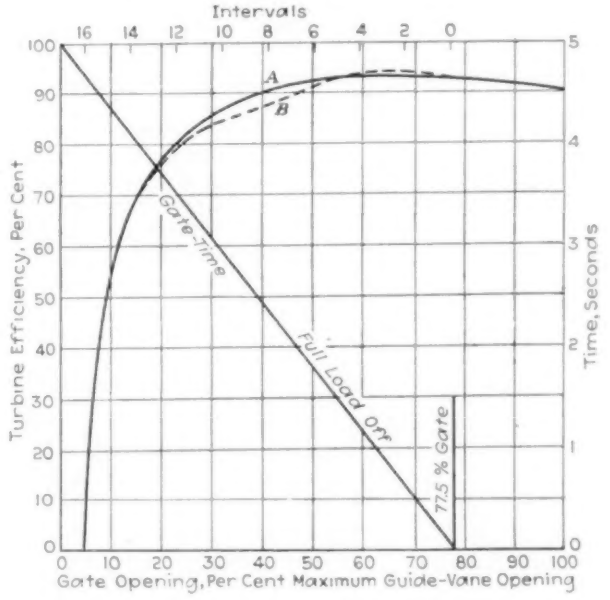


FIG. 22

model efficiency are taken from Fig. 23 for the proper ϕ value and the proper gate as given for the various intervals of Table 4. By comparing these values of efficiency with the model efficiency for the rated condition of H = 213.5 ft. and speed = 107.1 r.p.m., the change in model efficiency is found for runaway conditions existing during the load rejection, which change in efficiency is applied directly to curve *A* of Fig. 22 to obtain curve *B*. The values in column 8 of Table 3 are taken from curve *B*, and the rest of the computation is made similar to that of Table 3 resulting in the value of 38.9 per cent speed change in the place of the former value of 39.0 per cent, which was based upon no correction for the runaway characteristic of the wheel.

In this particular instance the change in speed rise caused by considering the runaway characteristic is not great, but in other cases of design a slightly larger change may result, say 1 or 2 per cent, depending upon the runner characteristics. At any rate the effect of the runaway characteristic will usually be found to be small.

The exact solution given here is not only correct from a theoretical standpoint, but has been tested experimentally by J. J. Traill, of the Hydro Electric Power Commission of Ontario, in the test work conducted at the Queenston and the South Falls Plants, already mentioned, with the result that the calculated results were closely checked.

To show what little work may be involved in a precise pressure

TABLE 2 COMPUTATION OF SPEED RISE (FROM TABLE 1)

(No Correction Made for Runaway Characteristic)

Interval	ϕn	h_n	$H = H_o + h_n \sqrt{\frac{H_o + h_n}{H_o}}$	V_n	$V_n H$	E, per cent	$C = 34.99\,E$	Hp. = $V_n HC$	Average Power for Interval Delivered by turbine	Given to rotating parts	0.01359 × hp.	N_1^2	R.p.m.	Speed change, per cent
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)
0.....	1.000	0.00	213.50	1.000	10.25	2189	92.8	32.48	71,100	73,180	72,930	991	11,449	107.0	0.0
1.....	0.960	13.85	227.35	1.032	10.16	2310	93.1	32.58	75,260	75,900	75,650	1028	12,440	111.6	4.4
2.....	0.917	23.39	236.89	1.054	9.90	2345	93.3	32.64	76,540	77,100	76,850	1044	13,468	116.0	8.5
3.....	0.871	33.67	247.17	1.077	9.61	2375	93.5	32.70	77,660	76,215	75,965	1032	14,512	120.5	12.6
4.....	0.821	37.39	250.89	1.084	9.12	2288	93.4	32.68	74,770	72,795	72,545	986	15,544	124.8	16.7
5.....	0.769	39.93	253.43	1.090	8.59	2177	93.0	32.53	70,820	68,475	68,225	927	16,530	128.7	20.4
6.....	0.714	42.76	256.26	1.096	8.02	2055	92.0	32.18	66,130	63,790	63,540	864	17,457	132.1	23.5
7.....	0.656	45.74	259.24	1.103	7.42	1924	91.3	31.94	61,450	55,975	55,725	757	18,321	135.5	26.7
8.....	0.594	49.31	262.81	1.110	6.08	1598	90.3	31.60	50,500	50,220	49,970	679	19,078	138.2	29.2
9.....	0.529	52.59	266.09	1.117	6.06	1613	88.5	30.96	49,940	46,525	46,275	629	19,757	139.9	30.8
10.....	0.463	54.52	268.02	1.120	5.32	1426	86.4	30.22	43,090	39,405	39,155	532	20,386	142.8	33.5
11.....	0.399	53.48	266.98	1.119	4.58	1223	83.5	29.21	35,720	32,165	31,915	434	20,918	144.7	35.3
12.....	0.332	55.91	269.41	1.125	3.82	1029	79.5	27.80	28,610	24,970	24,720	336	21,352	146.2	36.7
13.....	0.267	54.08	267.58	1.120	3.08	824	74.0	25.89	21,330	17,755	17,505	238	21,688	147.4	37.8
14.....	0.200	56.46	269.96	1.125	2.31	624	65.0	22.73	14,180	10,685	10,435	142	21,926	148.1	38.4
15.....	0.132	57.04	270.54	1.126	1.52	411	50.0	17.50	7,190	4,015	3,765	51	22,068	148.7	39.0
16.....	0.065	55.57	269.07	1.123	0.748	201	12.0	4.20	840	420	170	2	22,119	148.8	39.0
16.96..	0.000	56.50	270.00	1.125	0.000	0	0.0	0.00	0	.....	.....	..	22,121	148.8	39.0

$$V_n = V_o \times \phi_n \sqrt{\frac{H_o + h_n}{H_o}} = V_o \phi_n \sqrt{\frac{H}{H_o}}$$

or $V_n H = V_o \phi_n \sqrt{\frac{H^{3/2}}{H_o}}$

$N_1 = 107$ r.p.m.

$Wr^2 = 70{,}030{,}000$ lb-ft.^{2}

rise computation where the number of intervals is small, the computation for one of the examples suggested by Mr. Pfau worked through by Quick's method is exhibited in Table 5. This problem is for the following conditions:

L	= 2000 ft.	$2L/a$	= 1.333 sec.
V_{max}	= 10 f.p.s.	T_s	= 2 sec.
H	= 800 ft.	N	= 2/1.333 = 1.5
a	= 3000 f.p.s.	h_{max}	= 3000 × 10/32.2 = 0.5823

Approximate methods of computation have been proposed, which methods are accurate enough for some purposes, such as for estimating work. The approximate method published in Creager and Justin's Hydro Electric Handbook is one which is satisfactory for rough work. This method employs a "type characteristic" curve for Francis runners in obtaining the runaway characteristic, instead of treating each runner independently, and also employs the impulse and momentum relation for the pressure rise without correction for the elasticity of the water or of the pipe line.

For purposes of comparison this method has been used for the problem of Tables 1, 2, and 3. The result is a speed rise of 32.2 per cent, a value as based upon the foregoing computation in error by 6.7 per cent of the normal speed change, or 17.2 per cent of the change in speed.

COMPUTATION OF SPEED RISE BY APPROXIMATE METHOD[10]

Conditions. Full load thrown off 70,000-hp. unit in 5 sec. (same condition as in Tables 1 and 3).

For open flume:

$$I = \frac{80{,}000{,}000 \times \text{hp.} \times t}{(\text{r.p.m.})^2 \times Wr^2} = \frac{80{,}000{,}000 \times 70{,}000 \times 5}{(107)^2 \times 70{,}030{,}000}$$

$$= 34.9 \text{ per cent}$$

Corrected for runaway characteristic:

$$I' = \frac{I}{100 + \frac{100 \cdot I}{r - 100}} = \frac{34.9}{100 + \frac{34.9 \times 100}{180 - 100}} = 24.3 \text{ per cent}$$

Corrected for pressure rise:

$$J = \frac{LV}{gtH} = \frac{690 \times 10.25}{32.2 \times 5 \times 213.5} = 0.2055$$

[10] See Hydro Electric Handbook, by Creager and Justin, p. 641.

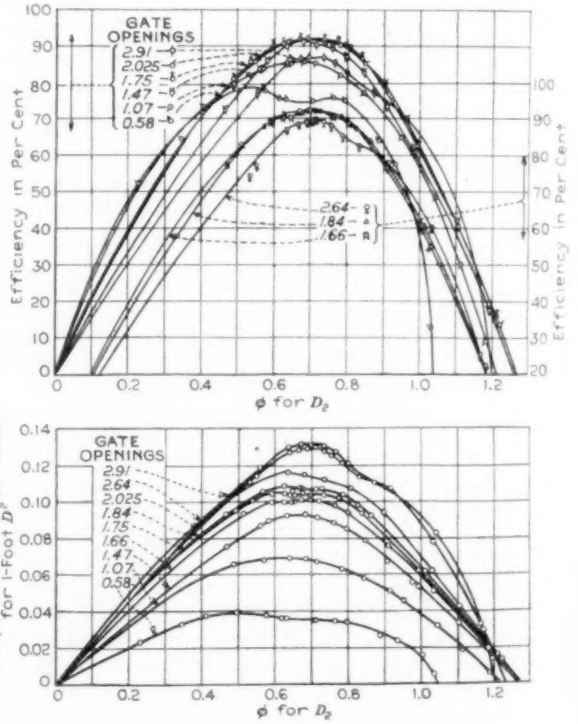


FIG. 23

$$I'' = I'(1 + J)^{1/2} = 24.3(1 + 0.2055)^{1/2} = 32.2 \text{ per cent}$$

It may be mentioned that among the problems where speed-regulation data are sometimes desirable and where accurate work may specially be desired are the following:

(1) Calculation of speed-regulation data for purposes of guarantee in connection with isolated units or units subject to isolation.

(2) Problems involving voltage rise on transmission lines.

(3) Problems involving time settings on protective relays.

(4) For calculating proper value for Wr^2 of unit.

(5) Special problems involving automatic plants and special problems with special load requirements.

The value of the author's paper, concerning the application of the rate-limit device to the hydraulic governor for controlling the rate of gate motion at any part of the servomotor stroke, should not be overlooked in discussing the mathematics of pres-

TABLE 3 COMPUTATION OF SPEED RISE (FROM TABLES 1 AND 2)

(Correction Made From Runaway Characteristic)

Interval	ϕ_n	h_n	$H = \sqrt{\frac{H_o+h_n}{H_o}}$		V_n	V_nH	E, per cent	$C = 34.99/E$	Hp. = V_nHC	Average Power for Interval Delivered by turbine	Given to rotating parts	0.01359 × hp.	N_1^2	R.p.m.	Speed change per cent.
(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)	(9)	(10)	(11)	(12)	(13)	(14)	(15)	(16)
0.....	1.000	0.00	213.50	1.000	10.25	2189	92.8	32.47	71,080	73,200	72,950	991	11,449	107.0	0.0
1.....	0.960	13.85	227.35	1.032	10.16	2310	93.2	32.61	75,330	76,030	75,780	1030	12,440	111.6	4.3
2.....	0.917	23.39	236.89	1.054	9.90	2345	93.5	32.72	76,730	77,390	77,140	1048	13,470	116.1	8.5
3.....	0.871	33.67	247.17	1.077	9.61	2375	93.9	32.86	78,040	76,650	76,400	1038	14,518	120.5	12.6
4.....	0.821	37.39	250.89	1.084	9.12	2288	94.0	32.89	75,250	72,980	72,730	988	15,556	124.8	16.6
5.....	0.769	39.93	253.43	1.090	8.59	2177	92.7	32.44	70,620	67,960	67,710	920	16,544	128.8	20.8
6.....	0.714	42.76	256.26	1.096	8.02	2055	90.8	31.77	65,290	62,780	62,530	850	17,464	132.2	23.5
7.....	0.656	45.74	259.24	1.103	7.42	1924	89.5	31.32	60,260	54,750	54,500	741	18,314	135.4	26.5
8.....	0.594	49.31	262.81	1.110	6.08	1598	87.9	30.76	49,150	48,930	48,680	662	19,055	138.1	29.1
9.....	0.529	52.59	266.09	1.117	6.06	1613	86.3	30.20	48,710	45,410	45,160	614	19,717	140.5	31.3
10.....	0.463	54.52	268.02	1.120	5.32	1426	84.4	29.53	42,110	38,690	38,440	522	20,331	142.6	33.3
11.....	0.399	53.48	266.98	1.119	4.58	1223	82.4	28.83	35,260	31,880	31,630	430	20,853	144.5	35.0
12.....	0.332	55.91	269.41	1.125	3.82	1029	79.2	27.71	28,510	24,910	24,660	335	21,283	146.0	36.4
13.....	0.267	54.08	267.58	1.120	3.08	824	73.9	25.86	21,310	17,750	17,500	238	21,618	147.2	37.6
14.....	0.200	56.46	269.96	1.125	2.31	624	65.0	22.74	14,190	10,690	10,440	142	21,856	147.9	38.2
15.....	0.132	57.04	270.54	1.126	1.52	411	50.0	17.50	7,190	4,020	3,770	51	21,998	148.4	38.7
16.....	0.065	55.57	269.07	1.128	0.748	201	12.0	4.20	840	420	170	2	22,049	148.6	38.9
16.96...	0.000	56.50	270.00	1.125	0.000	0	0.0	0.00	0	....	...	..	22,051	148.6	38.9

$$V_n = V_o \times \phi_n \times \sqrt{\frac{H_o + h_n}{H_o}} = V_o \times \phi_n \sqrt{\frac{H}{H_o}}$$

$$\text{or } V_nH = V_o\phi_n \sqrt{\frac{H^{3/2}}{H_o}}$$

N_1 = 107 r.p.m.

Wr^2 = 70,030,000 lb-ft.^{2}

sure rise and of speed rise. The writer has had personal experience with the rate-limit device as described in connection with making a Gibson test at a hydroelectric plant. The device was utilized in obtaining pressure-time diagrams which were ideal in shape and thus expedited the work to a marked degree.

TABLE 4 DATA USED IN DETERMINING RUNAWAY CHARACTERISTIC

Throat diameter main unit = 13.734 ft.
Throat diameter model = 36 in.
Ratio of diameters $\frac{36}{13.734 \times 12}$ = 0.2185
R.p.m. = 107.14
H = 213.5 ft.
Best ϕ = 0.66
100 per cent guide-vane opening main unit = 2.618-in. model gate opening

$$\phi = \frac{12 \times 13.734 \times \text{r.p.m.}}{1836 \times \sqrt{H}} = 0.08976 \frac{\text{r.p.m.}}{\sqrt{H}}$$

Interval	Time, sec.	Gate opening main unit, Model, in.	% max. guide-vane opening	From Table 3 Head, ft.	Speed, r.p.m.	ϕ	Model efficiency, %	Model efficiency $H = 213.5$ $N = 107.1$ %	Change in model efficiency, %
0....	0	2.060	77.5	213.5	107.1	0.658	91.3	91.3	−0.0
2....	0.59	1.790	68.2	236.9	116.0	0.677	91.7	91.5	+0.2
4....	1.18	1.540	59.0	250.9	124.8	0.707	91.3	90.7	+0.6
6....	1.77	1.310	50.0	256.3	132.1	0.741	87.6	88.8	−1.2
8....	2.36	1.070	41.0	262.8	138.1	0.765	82.9	85.3	−2.4
10....	2.95	0.835	31.9	268.0	142.8	0.783	78.7	80.7	−2.0
12....	3.54	0.592	22.6	269.4	146.2	0.799	74.7	75.0	−0.3
14....	4.13	0.354	13.5	270.0	148.1	0.809			
16....	4.72	0.118	4.5	269.1	148.8	0.814			

TABLE 5 PRESSURE-RISE COMPUTATION

L = 2000 ft. T_s = 2 sec.
V = 10 f.p.s. N = 1.5
H = 800 ft.
A = 3000 f.p.s.

N (interval)	0.0000	0.5000	1.0000	1.5000
t (sec.)	0.0000	0.6667	1.3333	2.0000
ϕ	1.0000	0.6667	0.3333	0.0000
2Σ	0.0000	0.0000	0.0000	0.7474
$1 - 2\Sigma$	1.0000	1.0000	1.0000	0.2526
$\phi_n^2 - K$	0.5823	0.3882	0.0647	0.0000
$1 - 2\Sigma + \phi^2_n - K$	1.5823	1.3882	1.0647	0.2526
$2K(1-2\Sigma)$	1.1646	1.1646	1.1646	0.2942
$1 + \phi^2_n - K^2$	1.3391	1.1507	1.0377	1.0000
$1 + 2K(1-2\Sigma) + \phi^2_n - K^2$	2.5037	2.3153	2.2023	1.2942
$\phi\sqrt{1 + 2K(1-2\Sigma) + \phi^2_n - K^2}$	1.5823	1.0145	0.4946	0.0000
P_n	0.0000	0.3737	0.5701	0.2526
$h_n = (931.677) \times P_n$	0.0000	348.1700	531.1500	235.3400
$\frac{hn}{Ho}$		43.5%	66.4%	29.4%

WILLIAM MONROE WHITE.[11] It seems that the discussion has diverted the attention from the main subject of the paper. Probably the author was too modest to emphasize the importance of the "rate-limiting device" which he invented, and which the writer considers to be one of the most valuable recent contributions toward a practical method of overcoming the many difficulties involved in connection with speed and pressure control of turbines, particularly when operating in connection with pipe lines.

The writer would like to point out the simplicity with which the author's device enables the obtaining of any desired rate of motion of the gate-control mechanism, as outlined in the five paragraphs on the fifth page.

Instead of depending on intricate theoretical calculations, the characteristics for assuring desired field results can be obtained in the field by the erecting engineer. He simply files down the two surfaces of the cam until, by successive tests, he has obtained that rate of governor motion which produces the permissible pressure variations in the pipe line. This procedure takes but very little time, and the adjustment, once properly accomplished, guarantees a permanent protection against pressure variations otherwise apt to be produced irrespective of results calculated in advance with the methods known today to the art.

O. V. KRUSE.[12] The portion of the paper dealing with pressure rise might rightfully be subject to criticism for the reason that the formula of Minton Warren is used as one of the determining factors. During the years 1916–17, the writer was privileged to work with Norman R. Gibson in a general water-hammer investigation, and at this time the limitations of the various theories were realized and used accordingly. As a result of these investigations, there was gained a forcible impression of the necessity of knowing the exact shape of the pressure-rise curve during turbine-gate closure. No formula seemed to satisfy the requirements except the work of Allievi, who took into account, not only the elastic properties of the water, but also the material making up the flow line. This basic consideration of elastic properties, coupled with the variable rate through the turbine gates corresponding to the pressure change, provides a complete background for determining exactly what is transpiring throughout the entire gate closure and permits of an accurate determination of all of the pressure conditions existing for various characteristics of gate closure.

The only grounds on which the subject can be treated fully and completely, in the writer's opinion, are to adhere to this theory of water-hammer determination and not resort to a simpler form of formula, such as Warren's, which is known to be very limited in its scope. Probably a mathematical solution

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[12] I. P. Morris & De La Vergne, Inc., Philadelphia, Pa.

incorporating the full water-hammer theory would be entirely impracticable, but such mathematical work can well be replaced by arithmetic integration. This system of computation is well recognized as being extremely accurate to take care of a multiple of variables. Unquestionably, a determination of the required variation in gate travel from straight-line motion could be made in this way, and if the physical characteristics of the apparatus and flow line are fairly well established, it is not out of reason that the gate motion so determined would give the desired results with little or no field adjustment.

S. LOGAN KERR.[13] The author is to be complimented upon his very detailed analysis of the method for computing pressure differences in the servomotors of a hydraulic-turbine unit due to movements of the governor valve. His analysis of the effect of the variable hydraulic load upon the operating mechanism and on the governor-valve action is of great interest.

Especially to be commended is the definition of governor rating. This value, instead of being defined as the total foot-pounds of energy that must be exerted through a full stroke, is given as the rate at which this energy is employed—namely, in foot-pounds per second. It is this value which is the true indication of a governor size, since the rate of flow through the governor determines the dimensions of the valves and piping.

The writer has used this rating for a number of years and has found it to be far more satisfactory than the overall definition commonly employed.

The rate-limiting device which the author has developed offers a very simple solution to a number of perplexing problems involving the limitation of the rate of movement of the turbine-operating mechanism. It has been our practice, however, to employ a more direct throttling system whereby the maximum rate of movement of the turbine gates is fixed, either by a diaphragm in the piping or by a positive stop on the main or pilot valve which can be securely locked in position for safety. The essential slowing up of the gate movement as it approaches the closed position is accompanied by a cushioning effect in the operating cylinder which is most effective in preventing dangerous water hammer resulting from short strokes from a small gate opening to zero. This is an additional feature of safety in that the cushion and the diaphragm are not readily subject to tampering by the operating staff. A combination of these two factors might offer a still better solution to this involved problem by having the maximum rates of the mechanism fixed by means of diaphragms and the cushion in the cylinders and still have the rate-limiting device on the pilot valve to conform to the immediate requirements for regulation.

The writer would suggest an amendment to the statements on the fifth page under headings 1 and 2, where the author describes the desired effects which can be secured by means of the rate-limiting device. Instead of the "constant rate" of piston motion, the writer would suggest "*maximum* constant rate."

The same amendment should be made under heading 2: "A *maximum* rate which produces a constant rate of change of output." By inserting this additional word in each of these paragraphs, the statement will cover all cases, since the maximum rates will be reached only when the pilot valve has made its full amount of travel.

With respect to the item 3 on this same page and with respect to section entitled, "Calculations of Pressure Rises in Pipe Lines," on the twelfth page, the writer is forced to differ very materially with the author in his assumptions. The writer's paper, mentioned by Mr. Pfau,[14] outlined in detail the method of computing the pressure rise in hydraulic-turbine conduits for all conditions of flow. The paper by Ray S. Quick[15] also is of great interest in connection with the computation of pressure-rise conditions.

In Mr. Quick's paper, the comparison of results secured from the Warren formula (used by Mr. Pfau in his presentation) was found to be markedly in error when compared with the elastic-water-column theory. This latter theory has been successfully proved in the United States by a number of experiments conducted by the Southern California Edison Company, and also through work done by Norman R. Gibson and R. D. Johnson, while in Europe a tremendous amount of work has been already accomplished which confirms beyond a doubt the analysis of water hammer by Lorenzo Allievi[16] and the work of Camichel, Eydoux, and Gariel.[17]

The essential features to point out are (1) that no single formula has as yet been developed which will give the correct result within 1 or 2 per cent of the true maximum pressure rise in a closed conduit; (2) that the maximum pressure rise is a function of the rate of change of velocity with respect to time; and (3) that the pressure rise is dependent upon the initial flow conditions from which the closure starts.

This latter fact cannot be emphasized too strongly, as it is this very feature which complicates the water-hammer studies. As pointed out in the writer's paper presented last year, three zones for water hammer are outlined, and two constants $K = av/2gH$ and $N = aT/2L$ are required.

The limits of the zones are determined by the simple relations as follows:

Zone 1—K being less than 1
N being less than 1
Zone 2—N being greater than 1
K being less than 1
Zone 3—N being greater than 1
K being greater than 1

Critical conditions are found at the common limit to zones 1 and 2—namely, at the point where $N = 1$. In both zones 1 and 2 the water-hammer studies should be carefully made to avoid dangerous pressure-rise conditions. In zone 3 the maximum pressure rise is the same for all conditions of initial flow, provided the rate of change of velocity with respect to time during closure is a constant.

In Table 1 in the writer's paper, previously mentioned, a remarkably close agreement is shown between the formulas of de Sparre-Gariel, the charts of R. S. Quick, and the arithmetic integration of N. R. Gibson. Applying Mr. Pfau's formula to these conditions, there results a very interesting agreement, as shown in Table 6.

This shows a reasonably close agreement throughout the range in flow, since this particular problem falls within the zone outlined by Mr. Quick, in which there is a reasonably close agreement between Mr. Warren's formula and the elastic wave theory.

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[14] See "New Aspects of Maximum Pressure Rise in Closed Conduits," by S. Logan Kerr, *Trans. A.S.M.E.*, vol. 51 (1929), no. 6, paper HYD-51-3.

[15] "Comparison and Limitations of Various Water-Hammer Theories," by Ray S. Quick, presented at Spring Meeting of A.S.M.E., May, 1927.

[16] "Theory of Water Hammer," by Lorenzo Allievi, translated by E. E. Halmos, 1925.

[17] "Étude Theorique et Experimentale des Coups de Belier," Camichel, Eydoux, and Gariel, a publication of the University of Toulouse, France, and under the auspices of La Société Hydrotechnique de France.

TABLE 6

Initial velocity, ft. per sec.	Time of closure intervals	Rise in pressure, in feet: Formulas of de Sparre-Gariel	Charts of R. S. Quick	Arithmetical integration of Gibson	Pfau
10	10	51.8	51.8	51.5	49.0
9	9	51.8	51.8	..	49.4
8	8	51.8	51.0	..	49.7
7	7	51.8	51.0	..	50.2
6	6	51.8	51.2	..	50.8
5	5	51.8	53.8	54.1	51.8
4	4	58.2	59.6	..	53.3
3	3	66.5	67.4	68.0	55.5
2	2	77.7	78.5	..	62.2
1	1	93.3	93.2	93.26	93.2
0	0	0.0	0.0	0.0	0.0

However, by taking the example given on the thirteenth page of the paper, Fig. 16, a very great discrepancy has been found to exist. This is shown in Table 7.

TABLE 7

T = sec.	Rise in pressure, in feet: Allievi-Quick	de Sparre-Gariel	Error, Per cent	Pfau	Error, Per cent
10	83.9	82.2	2.03	66.5	20.7
5	180.0	174.0	3.33	143.0	18.4
3	318.0	313.0	1.57	266.0	16.4
2	523.0	520.0	0.57	466.0	10.9
1.5	783.0	780.0	0.38	742.0	5.0
1.33	933.0	933.0	0.0	933.0	0.0

It will be seen from the figures that the author's formula may be as much as 20 per cent below the true value. This is a very serious error and still further emphasizes the fact that the Warren formula is not reliable for all conditions. The accepted theories of Allievi and the formula or charts developed by Quick, de Sparre, and Gariel and the arithmetic integration method of Gibson should be used.

Author's Closure

The author is deeply indebted to Messrs. Byron E. White and William Monroe White for redirecting the attention to the most important chapter of the paper, that relating to the rate limiting device.

Mr. E. B. Strowger's comment that Equations [38], [39], [40], and [42] cannot be used for accurate results is quite correct and has been fully covered by the word "if" at the beginning of the second paragraph directly above Equation [38]. These equations were cited merely for the purpose of gradually leading to Equation [51], approved by Mr. Strowger and to which no "if"-string was tied.

A "throttling system," as cited by Mr. Kerr, located in the piping, or a positive stop on the main or pilot valve only limits the "maximum point" (the peak point, so to speak) of rate of the turbine-operating mechanism. It does not control independently both the closing and the opening rates of the control mechanism over the entire stroke, as is the case with the rate-limiting device. It is evident, therefore, that only the rate-limiting device permits of obtaining the shortest possible and therefore the most efficient rates within the permissible limits of pressure variations.

A more positive form, as preferred by Mr. Kerr, is also fully available with all the principal advantages of the proposed "rate limiting" device. It is the design by W. M. White in the form of positive displacement or throttling rods integral with the servomotor. This has already been successfully applied with servomotors directly actuating the needles of impulse wheels, where a positive and absolutely foolproof control of rate of flow of water in the high-pressure pipe line is of utmost importance (60,000-hp. impulse wheel at Big Creek 2-A and 40,000-hp. unit at San Francisquito No. 1, both in California). These rods of course also produce the cushioning effect cited by Mr. Kerr and cannot be tampered with by the operating staff. They render superfluous a combination as proposed by Mr. Kerr.

The amendment suggested by Mr. Kerr on the fifth page under headings 1 and 2 seems to be superfluous when it is realized that the name "rate-limiting device" already discloses the fact that it is a "limiting device," and as such becomes fully operative only when the full valve stroke has been attained. It is obvious that only under such conditions is the "maximum" rate obtained, and only under such conditions is the limiting effect of the device desired.

As regards the discussion of the third item, it is conceded that the Warren formula in certain cases does not furnish accurate results. A further analysis of the subject may prove of interest, with particular reference to the three characteristic cases as discussed in Mr. Kerr's paper.

Case 1: $T \leqq \frac{2L}{a}$

The general Warren formula [58]

$$\frac{\Delta H}{H} = \frac{LV}{gH\left(T - \frac{L}{a}\right)}$$

turns into $a \cdot V/g \cdot H$ for total and complete closure.

It cannot exceed the value aV/gH even if $T < 2L/a$. This checks entirely with the theories published by Mr. Kerr.

Special Case: $T_z \leqq 2L/a$ for partial but complete closures.

The transformed Warren formula [59]

$$\frac{\Delta H_z}{H} = \frac{LV_m}{gHT_s}\frac{\psi_z}{\varphi_z}\frac{1}{1 - \frac{L}{a\varphi_z T_s}}$$

with limitation $\psi_z/\varphi_z \equiv 1$ (straight velocity curve) reads [59a]

$$\frac{\Delta H_z}{H} = \frac{LV_m}{gHT_s}\frac{1}{1 - \frac{L}{a\varphi_z T_s}}$$

For $\varphi_z T_s \equiv T_z = 2L/a$, we obtain [60]

$$\frac{\Delta H_r}{H} = \frac{2LV_m}{gHT_s} = \frac{aV_m\psi_z}{gH} \quad \ldots\ldots\ldots\ldots [61]$$

These values are identical with the results obtained with formula [3a] of Mr. Kerr's paper,[4] viz., $h_n = \frac{aV_n}{g}$

$$V_n \equiv V_s = V_m\,\psi_z, \quad \text{or} \quad V_m \equiv V_s$$

$$\Delta H_r \equiv h_n$$

recommended by him, and as evidenced by the two examples cited in his discussion: (1), 93.3 ft.; (2), 933 ft.

Case 2: $\quad T > \frac{2L}{a}; \quad \frac{aV_o}{2g\,H_o} < 1$

Mr. Kerr's formula [4a] for total closure from full is:

$$h = \frac{2LV_o}{gT}\frac{1}{1 + \frac{aV_o}{2g\,H_o}\left(1 - \frac{2L}{aT}\right)}$$

and, with limitations substituted, we obtain for total full closure:

$$h = \frac{2LV_o}{gT}\frac{1}{1 + (<1)\left(1 - \frac{2L}{aT}\right)}$$

$$h = \frac{2LV_o}{gT} \frac{1}{< 2 - < \frac{2L}{aT}}$$

$$= \frac{LV_o}{gT} \frac{1}{< \left[1 - \frac{L}{aT}\right]}$$

or

$$h = \frac{LV_o}{< \left[g\left(T - \frac{L}{a}\right)\right]}$$

It will be noted that the foregoing formula is identical "in structure" with the Warren formula, and it becomes numerically identical in the special case where $\frac{aV_o}{2gH_o} = 1$ or $\frac{V_o}{2gH_o} = \frac{1}{a}$

when $\frac{aV_o}{2gH_o} < 1$, as covered by Case 2 in question, the pressure rise obtained with Warren's formula is too low, as can be seen from foregoing formula $h > \Delta H$
because

$$\frac{1}{< \left[T - \frac{L}{a}\right]} > \frac{1}{T - \frac{L}{a}}$$

For complete closure from partial gate opening Mr. Kerr's formula is:

$$h_n = \frac{2LV_n}{gT_n} \frac{1}{1 + \frac{aV_n}{2gH_o}\left(1 - \frac{2L}{aT_n}\right)}$$

If we substitute for $V_n = V_o\psi_n$, we obtain $T_n = T_s\varphi_n$

$$h_n = \frac{2LV_o}{gT_s} \frac{\psi_n}{\varphi_n} \frac{1}{1 + \frac{aV_o\psi_n}{2gH_o}\left(1 - \frac{2L}{aT_s\varphi_n}\right)}$$

for a straight-line V-curve and a proportionate governor time $\frac{\psi_n}{\varphi_n} \equiv 1$, thus:

$$h_n = \frac{2LV_o}{gT_s} \frac{1}{1 + \frac{aV_o}{2gH_o}\psi_n\left(1 - \frac{2L}{aT_s\varphi_n}\right)}$$

or

$$\equiv A \frac{1}{1 + B\psi_n - B\frac{2L}{aT_s}\frac{\psi_n}{\varphi_n}} = \frac{A}{1 + B\left(\psi_n - \frac{2L}{aT_s}\right)}$$

for the special point where $T_n = \frac{2L}{a}$ we have

$$\varphi_n = \frac{2L}{aT_s} \text{ and since } \varphi_n \equiv \psi_n \text{ because } \frac{\psi_n}{\varphi_n} \equiv 1$$

we obtain

$$h_n \equiv h_r = \frac{A}{1 + B(\psi_n - \varphi_n)} = A = \frac{2LV_o}{gT_s}$$

which again checks with the Warren formula (point b_1, Fig. 4 of Mr. Kerr's paper[4]) for values of $\varphi_n > \frac{2L}{aT_s}$ and since $\varphi_n \equiv \psi_n$

$$\psi_n > \frac{2L}{aT_s}$$

we have

$$h_n = \frac{A}{1 + B\left[> \left(\frac{2L}{aT_s}\right) - \frac{2L}{aT_s}\right]} = \frac{A}{1 + > B}$$

The values h_n are smaller than the value h_r.

For points between $\varphi_2 = 1$ and $\varphi_n = \frac{2L}{aT_s}$

we have:

$$h_n = \frac{2LV_n}{gT_n} \frac{1}{1 + \frac{aV_n}{2gH_o}\left(1 - \frac{2L}{aT_n}\right)}$$

since $\frac{aV_o}{2gH_o} < 1$ and since $V_n < V_o$
we can write:

$$\frac{2LV_n}{gT_n} \frac{1}{1 + \ll \left(1 - \frac{2L}{aT_n}\right)}$$

or

$$h_n = \frac{2LV_n}{gT_n} \frac{1}{1 + \ll 1 - \ll \frac{2L}{aT_n}}$$

$$= \frac{LV_n}{gT_n < \left(1 - \frac{L}{aT_n}\right)} \text{ or } \frac{LV_n}{< \left[g\left(T_n - \frac{L}{a}\right)\right]}$$

This is again the structure of the Warren formula. However, it proves that the values obtained are greater than those obtained by Warren.

$$\Delta H_z < h_n$$

Case 3:

$$T > \frac{2L}{a}; \qquad \frac{aV_o}{2gH_o} > 1$$

Mr. Kerr's Formula [5] for total closure:

$$h = \frac{2LV_o}{gT} \frac{1}{2\left(1 - \frac{LV_o}{2gTH_o}\right)}$$

can be rewritten as follows:

$$h = \frac{LV_o}{g\left(T - \frac{LV_o}{2gH_o}\right)}$$

Here again for the special case of

$$\frac{V_o}{2gH_o} = \frac{1}{a} \text{ instead of } > \frac{1}{a},$$

the foregoing formula goes over into the structure of Warren's formula:

$$h = \frac{LV_o}{g\left(T - \frac{L}{a}\right)}$$

or, with the author's symbols,

$$\Delta H = \frac{LV_m}{g\left(T - \frac{L}{a}\right)}$$

For $\frac{V_o}{2gH_o} > \frac{1}{a}$ we obtain:

$$h = \frac{LV_o}{g\left(T - < \frac{L}{a}\right)}$$

again demonstrating that Warren's formula in this particular case shows too low pressure rises.

Referring to Mr. Kerr's numerical example for Case 3, we have

Warren: $$\frac{LV}{g\left(T - \frac{L}{a}\right)} = \frac{600.10}{32.2\left(4 - \frac{600}{3000} = 0.2\right)}$$

$$= 49.03 \text{ ft.}$$

Kerr: $$\frac{LV}{g\left(T - \frac{LV_o}{2gH_o}\right)} = \frac{600.10}{32.2\left(4 - \frac{600.10}{2 \times 32.2 \times 233}\right)}$$

$$= \frac{6000}{32.2\left(4 - \frac{600}{1500}\right)} = 0.4$$

$$= 51.8 \text{ ft., because}$$

$$\frac{V_o}{2gH_o} = \frac{10}{32.2 \times 2 \times 233} = \frac{1}{1500}, \text{ against } \frac{1}{a} = \frac{1}{3000}$$

For complete partial closure we obtain:

$$h_n = \frac{2LV_n}{gT_n} \frac{1}{2\left(1 - \frac{LV_n}{2gT_nH_o}\right)}$$

or $$h_n = \frac{2LV_o}{gT_s}\frac{\psi_n}{\varphi_n} \frac{1}{2\left(1 - \frac{LV_o}{2gH_oT_s}\right)\frac{\psi_n}{\varphi_n}}$$

or since $$\frac{\psi_n}{\varphi_n} \equiv 1$$

$$= \frac{LV_o}{gT_s} \frac{1}{\left(1 - \frac{LV_o}{2gH_oT_s}\right)} \equiv \text{constant}$$

since $$\frac{aV_o}{2gH_o} > 1$$

it follows that $$\frac{LV_o}{2gH_oT_s} > \frac{L}{aT_s}$$

We can thus write:

$$h_n = \frac{LV_o}{gT_s} \cdot \frac{1}{1 - > \frac{L}{aT_s}} \quad \text{or} \quad \frac{LV_o}{g\left(T_s - > \frac{L}{a}\right)}$$

which again demonstrates the structure of the Warren formula, except that the value is *constant* and is greater than the value obtained by Warren for a full complete closure.

From the foregoing it seems evident that the structure of the Warren formula has maintained itself throughout the most recent publications; however, with modifications, which have brought the calculated results into closer agreement with the

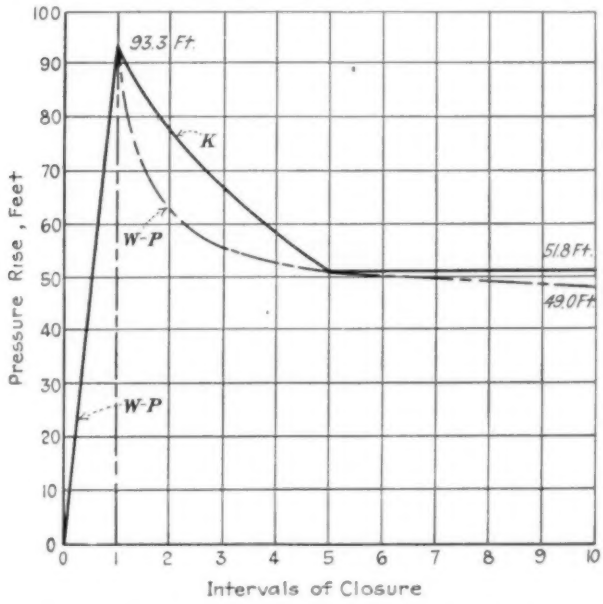


Fig. 24 Pressure-Rise Curve of Partial Closures

actual results under any possible underlying conditions. The two numerical examples tabulated by Mr. Kerr are plotted on Figs. 24 and 25.

The other comments made by Messrs. Strowger, Kruse, and Kerr, with reference to calculations of pressure variations, are fully recognized by the author, who would ask to be permitted to

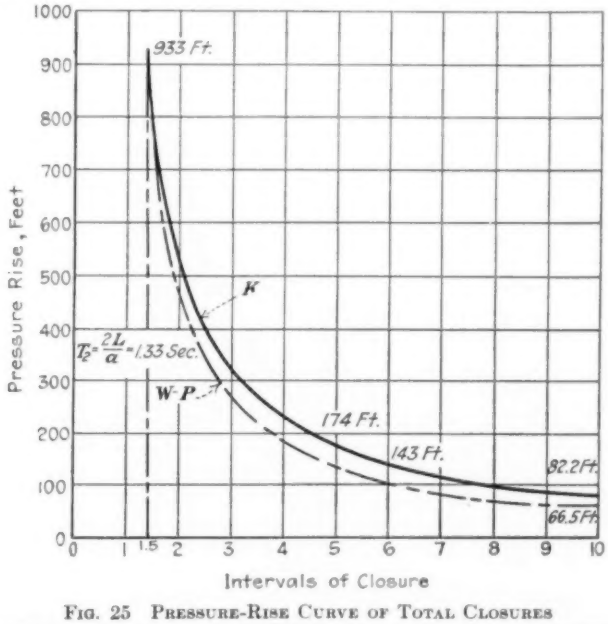


Fig. 25 Pressure-Rise Curve of Total Closures
(Note: *K* curve is according to Kerr; *W-P* curve is according to Warren-Pfau.)

state in closing that the available data relating to a natural phenomena are quite often increased in depth, and consequently in value, by an analysis of discrepancies arising out of results of independent studies.

Report of Committee on Survey of Hydraulic Research, of the Hydraulic Division[1]

THE Committee on Survey of Hydraulic Research of the Hydraulic Division of the Society was organized by the Division with Prof. C. M. Allen[2] as its chairman. A program was mapped out, and work was allocated to the several members. The members of the committee, with their respective assignments, are as follows:

R. E. B. Sharp, I. P. Morris Corporation, Philadelphia, Pa. Research in Europe.

Forrest Nagler, Allis-Chalmers Mfg. Co., Milwaukee, Wis. Water-wheel manufacturers (with R. V. Terry).

R. V. Terry, Newport News Shipbuilding and Drydock Company, Newport News, Va. Water-wheel manufacturers (with Forrest Nagler).

N. R. Gibson, Niagara Falls, N. Y. Water-power companies, exclusive of Pacific Coast (with I. A. Winters).

I. A. Winters, Alabama Power Company, Birmingham, Ala. Water-power companies of the South (with N. R. Gibson).

R. M. Peabody, Southern California Edison Company, Los Angeles, Calif. Water-power companies of the Pacific Coast.

L. F. Harza, Chicago, Ill. Research by consulting engineers (with W. F. Uhl).

W. F. Uhl, Boston, Mass. Research by consulting engineers (with L. F. Harza).

C. M. Allen, Worcester Polytechnic Institute, Worcester, Mass. Research in hydraulic laboratories of engineering colleges.

The material collected in the several fields by members of the committee is summarized in the following tables, preceded and followed by appropriate comment. After the table of research problems suggested by the committee as worthy of experimental study (some of them entirely new problems and others supplementary to experimental work already done), there is a record of research work done or being done.

Hydraulic Problems for Research

The following are hydraulic problems for research, suggested by members of the committee as coming from themselves and from those who submitted answers to questionnaires:

Problem	Suggested by	Special factors	Purpose
Relative turbine efficiency, standard test setting versus designed setting	R. V. Terry	Open flume, metal spirals, concrete spirals, concrete semi-spirals	Relate standard test, with actual operating efficiency
Effect of size on performance of turbines	R. V. Terry	Output, efficiency, best speed	Check law of similitude
Cylindrical gates outside wickets	W. F. Uhl		
Butterfly valves	W. F. Uhl		
Spiral flow (flow in spirals)	F. Nagler		
Draft tubes	L. F. Harza and W. F. Uhl[3]		Add to and correlate present knowledge
Scroll-case design	L. F. Harza	Bends in approach conduit, water passages at exit therefrom	
Mechanics of pitting	F. Nagler and W. F. Uhl[3]		Protection of runners
Lubrication	F. Nagler		Elimination of oil vapors to avoid damage to windings
Stuffing-box packing	W. F. Uhl[3]		
Intake design[4]	W. F. Uhl[3]		Make ice and debris proof. (Use models for study.)
Losses at trash racks[4]	W. F. Uhl[3]		
Friction loss in conduits[4]	W. F. Uhl[3]		
Permissible velocities in concrete water passages	R. V. Terry	Conduits, spiral cases, draft tubes	Avoid erosion
Characteristics of waves from operation of water-power plants[4]	W. F. Uhl[3]		
Movement of flood waves[4]	W. F. Uhl[3]		
Current meters[4]	L. F. Harza and W. F. Uhl[3]	Diagonal flow, disturbed flow	
Spillway discharge coefficients[4]	I. A. Winters and W. F. Uhl[3]	Shape of crest, shape of piers, head on crest, depth and approach to channel alternate and adjacent crest gates open	Increase capacity
Proper design, apron of dam	L. F. Harza	Dimensions, shape water approach, water depth, water exit	Avoid destructive wear
Hydrostatic uplift, structures on previous soils	W. F. Uhl[3]		
Design and curing concrete for dams in cold climates	W. F. Uhl	Aggregates, proportions, water-cement ratio, and consistency for placing, spading next forms on exposed faces, segregation in handling and placing	Control permeability, prevent rapid disintegration
Transportation of sedimentary material[4]	W. F. Uhl[3]		
Keep abreast of foreign developments in hydraulic research	R. E. B. Sharp	Benefit by, avoid duplication	Translation and publication of data as soon as available in the mother tongue
Disk friction	R. E. B. Sharp		

[1] Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of The American Society of Mechanical Engineers.

[2] Professor of Hydraulic Engineering, Worcester Polytechnic Institute, Worcester, Mass. Mem. A.S.M.E.

[3] Problems suggested in answers to questionnaire by W. F. Uhl.

[4] Refer to "Hydraulic Laboratory Practice," by Dr. J. R. Freeman.

Hydraulic Researches in Europe

Researches in Europe are classified under three headings:

1 General hydraulics
2 Hydraulic turbines
3 Centrifugal pumps

Most of the data compiled are under the first two headings.

General Hydraulics

1 Disk Friction—"Resistance of Spheres and Disks at Low Reynolds Coefficients" ("Widerstandsmessungen and Kugel und Scheibe bei kleinen Reynoldsschen Sahlen"), L. Schiller and H. Schmiedel, *Zeit. fuer Flugtechnik und Motorluftschiffahrt* (Munich), vol. 19, no. 21, Nov. 14, 1928, pp. 497–501, 9 figs. Description of apparatus; results of experimental study confirming deductions of Oseen theory. E.I.S. serial no. 28, 45,149. (See comment 1.)

2 "Diverting a Free Liquid Jet on a Flat Surface," by Dr. Ing. Friedrich Reich, *V.D.I. Verlag*, G.M.B.H., Berlin, pp. 77. (See comment 2.)

3 Flow of Fluids, Theory—"Changes in Regimen of Fluids Flowing in Closed Conduits" ("Sui cambiamenti di regime nel moto di fluidi in condotti chiusi"), M. Danti, *Elektitrotechnica* (Milan), vol. 15, no. 14, May 15, 1928, pp. 374–379, 8 figs. Reviews theoretical work of Reynolds, Camerer, Lorenz, Morrow, and others on flow in cylindrical pipes and closed conduits of variable cross-section; bearing on these theories on design of hydraulic turbines. E.I.S. serial no. 28, 19,012.

4 Flow of Water—"Measure of Discharge in a Pressure Conduit" ("Mesure du debit dans une conduite forcee"), A. Schlag, *Revue Universelle des Mines* (Liege), vol. 17, no. 5, Mar. 1, 1928, pp. 215–229, 12 figs. Experimental study of different methods of measurement used at laboratory of machine construction of Liege University; treats of pitot tube and theoretical flow formulas; loss of pressure due to measuring instruments. (Concluded.) E.I.S. serial no. 28, 8116.

5 Flow of Water, Measurement—"Actual State of the Technique of Gaging" ("Etat actuel de la technique des jaugeages"), A. Bijls, *Annales de l'Association des Ingenieurs Sortis des Ecoles Speciales de Gand* (Ghent), vol. 18, series 5, no. 3, 1928, pp. 351–363. Hydrometric mills; new types and recent studies; measurement of small velocities; gaging by weirs, by

chemical means, by pressure pipes, by salt solution, and venturi meter are treated. E.I.S. serial no. 28, 44,368.

6 Hydraulic Laboratories, Europe—"Establishment and Operation of Hydraulic Laboratories" ("Inrichting en werkwijze van hydrodynamische Laboratoria"), J. W. E. C. Proper, *Waterstaats-Ingenieur* (Surabaya, Java), vol. 16, no. 4, Apr., 1928, pp. 104–112, 6 figs., partly on supplementary sheet. Report on author's study trip through Europe; descriptions and general observations of organization and operation of hydraulic laboratories in Germany, Austria, and Czechoslovakia, paying special attention to Karlsruhe laboratory and work of Rehbock; advocates establishment of hydraulic laboratory in Dutch East Indies as economic necessity. E.I.S. serial no. 28, 18,217.

7 Hydraulic Laboratories, Munich University—"Methods of Testing and Equipment at the Hydraulic Institute of Technical University of Munich," B. R. Van Leer, *Purdue Eng. Rev.*, vol. 23, no. 3, Mar., 1928, pp. 3–4, 18, 22, and 26, 1 fig. Description of plant and personnel in hydraulic laboratory of Germany's foremost university; instructions given by institute; laboratory work; administration; commercial testing; location and workshop; switch and pump rooms; physical features of laboratory; experimental instruction; research now under way; future or planned research. E.I.S. serial no. 28, 11,315.

8 Hydraulic Models—"Experimental Research at the Institute of Hydraulics and Hydraulic Construction of the Royal School of Engineering of Pisa" ("Ricerche sperimentale eseguite nell' Instituto di Idraulica e di Costruzioni Idrauliche della R. Scuola d'Ingegneria di Pisa"), G. de Marchi, *Energia Elettrica* (Milan), vol. 5, no. 4, Apr., 1928, pp. 481–489, 18 figs. Plan and equipment of hydraulic laboratory; notes on studies of models on hydraulic-jump structures; flume for study of flow over weirs; tests of models of siphons and siphon spillways. E.I.S. serial no. 28, 15,636.

9 Hydraulic Models, Chutes—"Tests of Models of By-Pass Chute of the Shannon Hydroelectric Power Plant" ("Modellversuche fuer die Schussrinne des Shannon-Kraftwerkes"), Ludin, *Bauingenieur* (Berlin), vol. 9, no. 11, Mar. 16, 1928, pp. 179–184, 9 figs. Laboratory tests of 1 to 50 models of sluice-regulated chute of about 11-m. fall, by-passing flow about turbines when plant is not operating; study of surface curves and stilling basin effect; final design determined on basis of these tests. E.I.S. serial no. 28, 10,988.

10 Hydraulic Models, Groines—"Hydraulic Experiments With Models of the Effect of Currents and Surf Breakers Upon a Sandy Sea Beach, Etc." ("Modellversuche ueber die Wirkungen der Stroemungen und Brandungswellen auf einen sandigen Meeresstrand, etc."), B. Kressner, *Bautechnik* (Berlin), vol. 6, no. 25, June 12, 1928, pp. 374–386, 33 figs. Report from hydraulic laboratory of Danzig Institute of Technology on elaborate experimental study of beach formation, design of groins and seashore protection; description of apparatus for production of wave motion, design of models of beach and groins; mechanics of surf effect, detailed topography of artificial beach. E.I.S. serial no. 28, 22,104. (See comment 3.)

11 Laboratory Research—"Application of Laboratory Research to the Study of Hydraulic Problems," by Dr. Geo. H. de Thierry, published by Massachusetts Institute of Technology, serial no. 10, Jan., 1928.

12 Orifices, Discharge—"Discharge Coefficients of I. G. Measuring Orifices for Water, Oil, Steam, and Gas" "Durchflussbeiwerte der I.G.-Messmuendungen fuer Wasser, Oel, Dampf, und Gas"), R. Witte, *V.D.I. Zeit.* (Berlin), vol. 72, no. 40, Oct. 20, 1928, pp. 1493–1503, 51 figs. Report from physics laboratory of I.-G. Farbeinindustrie Aktiengesellschaft on results of Lappe's calibration of rounded orfices and sharp-edged diaphragm orifices of pipes, from 50 to 500 mm. in diameter; comparative calibration data of V.D.I. and I.G. orifices and nozzles. E.I.S. serial no. 28, 39,063.

13 Research, Hydraulics—"The Research Institute for Hydraulic and Hydroelectric Structures," B. R. Van Leer, *Mech. Eng.*, vol. 50, no. 8, Aug., 1928, pp. 607–610, 8 figs. Account of plans and activities of Forschungsinstitut fuer Wasserbau und Wasserkraft, and experimental field of this German institution; institute proposes to be connecting link between scientific laboratory studies upon models and actual original structures in field service; it has prepared plans and drawings of buildings, channels, and experimental stations and has conducted experiments to find discharge coefficients and downstream erosion on overflow spillways and weirs. E.I.S. serial no. 28, 25,156.

14 Study of Siphons—"The Performance of Automatic Siphons" ("Sul rendimento dei sifoni autolivellatori"), E. Scimemi, *Elettrotechnica* (Milan), vol. 15, no. 13, May 5, 1928, pp. 345–351, 13 figs. Theory, discharge formulas, and discharge coefficients of siphons and siphon spillways; recent determination of coefficients of discharge of American, German, and Italian siphons; detailed report on tests of models of five Italian canal siphons, checking Froude principle of similitude and proving comparative inefficiency of siphons of sharply curved or discontinuous profiles. E.I.S. serial no. 28, 16,359. (See comment 3.)

15 Surges, Hydraulic—"Surges in Canals and Pipes" ("Om svallningar i kanaler ochtuber"), H. O. Dahl, *Ingenioers Velenskaps Akademien-Handlingar* (Stockholm), no. 63, 18 pp., 4 figs. Elementary mathematical theory of surges in open channels and pressure conduits; effect of elasticity of liquid, short and long periods of valve closure; problem of surges in open channels still unsolved. E.I.S. serial no. 28, 7510. (See comment 3.)

16 Surge Tanks—"Criterion of Stability for Differential Surge Tanks at Load Disturbances of Finite Amplitudes" ("Das Stabilitaetskriterium fuer gedsempfte Wasserschloesser bei Belastungsstoerungen mit endlichen Schwingungsweiten"), J. Schueller, *Wasserkraft und Wasserwirtschaft* (Munich), vol. 23, no. 22, Nov. 15, 1928, pp. 347–349. Theoretical mathematical discussion based on work of E. Braun, R. D. Johnson, D. Thoma, and others. E.I.S. serial no. 28, 44,940.

17 "Water Levels in Open Channels, Including Discussion of Standing Wave Phenomenon" ("Berechnung der Wasserspeigellage"), by Dr. Ing. Paul Boss, *V.D.I. Verlag*, G.M.B.H., Berlin, pp. 96. (See comment 3.)

18 Gibson Method—"The Degree of Accuracy of the Gibson Water-Meter Researches," by D. Thoma, Transactions of the Hydraulic Laboratory of the Technical University at Munich, 1926, publication no. 1. (See comment 4.)

19 Venturi Meters—"Influence of Elbows Ahead of Venturi Meters Upon the Meter Readings," Hydraulic Institute at Munich. (See comment 5.)

20 Trash Racks—"Investigations of the Loss Through Trash Racks Inclined Obliquely to the Stream Flow," Hydraulic Institute at Munich. (See comment 5.)

21 Losses in Pipe Bends—"Experiments to Determine the Loss in Right-Angle Pipe Tees, Pipe Bends, and Various Forms of Elbows," Hydraulic Institute at Munich. (See comment 5.)

22 Trash Racks—"Loss of Head in Protecting Racks at Hydraulic Power Plants," Hydraulic Laboratory at Stockholm. (See comment 5.)

23 Flow of Liquids—"Alternate Eddies Due to Obstacles of Knife-Blade Form" ("Sur les tourbillons alternes dus a des obstacles en lames de couteau"), H. Benard, *Academie des Sciences Comptes Rendus* (Paris), vol. 187, no. 23, Dec. 3, 1928, pp. 1028–1030, 1 fig. Discussion of eddies formed by thin plate submerged in liquid stream and comparison with cylindrical-body eddies. E.I.S. serial no. 28, 46,610.

24 Flow of Water, Pipes—"Tests on Loss Through Sudden Enlargements in a Water Pipe" ("Essais sur la perte par elargissement brusque dans une conduite d'eau"), L. Bergeron, *Genie Civil* (Paris), vol. 93, no. 20, Nov. 17, 1928, pp. 475–477, 4 figs. Borda theory of sudden enlargement loss conformed by tests made at Hydraulic Institute of Polytechnic School of Munich; author discusses results which concern Boda-Carnot theory. E.I.S. serial no. 28, 43,445.

Comments on General Hydraulics Articles: 1 This would appear to be a subject worthy of study and further research in this country. 2 A very complete treatise which should be available in English. 3 Should probably come more under Civil Engineering heading; see "Hydraulic Laboratory Practice." 4 The translation of this article would be desirable. 5 Are available in "Hydraulic Laboratory Practice" and will be worthy of study.

Hydraulic Turbines

25 Corrosion—"Cavitation and Corrosion in Hydraulic Problems," by Fottinger, *V.D.I. Verlag*, Berlin, 1926. (See comments 6 and 7.)

26 Corrosion—"Destruction of Turbine Blades by Wear" ("Oedeleggelse av Turbinskovler ved Taering"), H. Thoresen, *Teknisk Ukeblad* (Oslo), vol. 75, no. 21, May 25, 1928, pp. 209–214, 7 figs. Various theories as to cause of cavitation and results of laboratory experiments; stage has been reached when Francis turbines may be constructed without risk of cavitation, with reasonable specific number of revolutions for falls of average height; methods of repairing wheels damaged by cavitation. (Concluded.) E.I.S. serial no. 28, 21,110.

27 Corrosion, Discussion of Phenomenon—"Corrosion and Cavitation in Hydraulic Turbines" ("Corrosions et cavatations dans les turbomachines hydrauliques"), A. Metral, *Revue Generale de l'Electricite* (Paris), vol. 23, no. 26, June 30, 1928, pp. 403H–416H, 5 figs. Article treats of hypotheses for explaining corrosion and also phenomenon of cavitation in hydraulic turbines; reports some tests on impact action of water drops at high speeds on moving metal walls at Brown-Boveri testing laboratory at Baden, Switzerland. E.I.S. serial no. 28, 25,190. (See comment 7.)

28 Corrosion, in Hydraulic Turbines—"Cavitation Phenomena in Hydraulic Turbines," J. Ackeret. *Bul. Technique de la Suisse Romande* (Vevey), vol. 54, no. 9, May 5, 1928, pp. 99–102, 6 figs. Shows causes of low pressures occurring and localizes critical points in turbine; author considers low-head turbine having draft tube. Abstracted from *Escher-Wyss Mitteilungen*, no. 2. See English abstract, *Mech. Eng.*, vol. 50, no. 7, July, 1928, pp. 549–550. E.I.S. serial no. 28, 21,848. (See comment 7.)

29 Corrosion, Pelton Water Wheels—"Cavitation in Pelton Wheels" ("Fenomeni di corrosione nelle Turbine Pelton"), G. R. Mosca, *Energia Elettrica* (Milan), vol. 5, no. 11, Nov., 1928, pp. 1316–1326, 31 figs. Analysis of phenomenon; experience of Norwegian and Italian hydroelectric power plants; cavitation of needle valves; review of studies by Honegger, Thoma, and Camerer. E.I.S. serial no. 28, 44,292. (See comment 7.)

30 Corrosion—"Turbine and the Cavitation Problem," by Elov Englesson, *Canadian Engineer*, April, 1928. (See comment 7.)

31 Corrosion, Turbine Runners—"Destruction of Turbine Blades by Wear" ("Odeleggelse av Turbinskovler ved Taering"), H. Thoresen, *Teknisk Ukeblad* (Oslo), vol. 75, no. 20, May 18, 1928, pp. 197–201, 6 figs. Studies made to ascertain cause of wear of certain parts of turbines, especially hollows of blades; effect of vacuum and other causes; methods of laboratory established for purpose in Kristinehamn, Sweden. (To be continued.) E.I.S. serial no. 28, 17,637. (See comment 7.)

32 Design Charts for Calculating Turbines—"Alignment Charts for Calculating Turbines" ("Abaques pour le calcul des turbo-machines"), A. Schlag, *Revue Universelle des Mines* (Liege), vol. 71, June 15, 1928, pp. 280–285, 3 figs. Hydraulic turbines, pumps, and ventilators are treated, divided into six types; alignment charts are given for calculating wheels of turbines. E.I.S. serial no. 28, 23,597. (See comment 8.)

33 Design of Hydraulic Turbines (Theoretical discussion of)—"Diffuser Rings of Full-Admission Hydraulic Turbines" ("Ueber das Leitrad der Vollstrahl-Wasserturbinen"), H. Baudisch, *Schweizerische Bauzeitung* (Zurich), vol. 92, no. 3, July 21, 1928, p. 29. Theoretical discussion of principles of design; differentiates between dynamic and static (Kaplan) method of energy transmission in turbines and combination of two principles. E.I.S. serial no. 28, 27,756. (See comment 8.)

34 Dimensions—"A Method of Obtaining the Leading Dimensions and of Setting Out the Blade Forms for Hydraulic Turbines of the Francis Mixed-Flow and Propeller Types," by Herbert Mawson, Selected Paper No. 48 of the Institution of Civil Engineers (London). (See comment 8.)

35 Draft Head, Maximum Allowable on Turbine Runners—"The Highest Permissible Suction Head in Hydraulic Turbines" ("Das hoechstzulaessige Sauggefaelle von Wasserturbinen"), J. Ackeret, *Schweizerische Bauzeitung* (Zurich), vol. 91, no. 11, Mar. 17, 1928, pp. 135–136, 2 figs. Theoretical discussion calling attention to so-called dynamic suction head, which is important factor in design of Kaplan and propeller turbines and in prevention of cavitation. E.I.S. serial no. 28, 10,568.

36 Draft Tubes—"Critical Considerations of the Method of Computation and Structural Development of Draft Tubes," by Dipl. Ing. Joseph Bronner, *Wasserkraft Jahrbuch*, 1927–28.

37 Flow in Casing (Study of)—"Flow of Water in Casing of an Axial Flow Turbine," K. Kaneshige, *Tokyo Imperial Univ. Jl.* (Japan), vol. 18, no. 8, Apr., 1928, pp. 153–182, 25 figs. Flow treated is same as in Lorenz's works, but special reference is made to size of rotating core and quantity of discharge; vibration of turbine draft tube and devices for its remedies; apparatus used by writer is explained, together with experimental data from which present deductions were obtained. E.I.S. serial no. 28, 16,246. (See comment 8.)

38 Flow of Water in Turbine Runners—"Principal and Secondary Flow in Turbine Runners," by Dipl. Ing. Schilhanse Munchen, *Wasserkraft Jahrbuch*, 1925–26. (See comment 8.)

39 "Progress in Construction of Water Turbines" ("Fortschritte im Bau von Wasserturbinen"), by Prof. Dr. Ing. Alsterlen, Hannover, *V.D.I. Zeit.*, Dec. 1 and 15, 1928. (See comment 9.)

40 Kaplan—"The Employment of Propeller Turbines and Kaplan Turbines," by Dipl. Ing. G. von Troelitsch, J. M. Voith, Heidenheim.

41 "Kaplan and Propeller Turbines Compared," by Elov Englesson, A. B. Karlstadt, *Mekaniska Verkstad*, Verkstaden, Kristenhamn, Sweden.

42 "Neue Wasserturbine," by Donat Banki, *Druck des Franklin Vereins*, 1917.

43 Pressure on Runner Blades—"Investigations Regarding the Pressure Distribution on a Stepped Row of Blades," translation from "Ergebuisse

des Aerodypamischen Versuchsanstalt Zu Gottingen III Lieferung," *Druck und Verlag*, by R. Oldenbourg, Munchen und Berlin, 1927. (See comment 10.)

44 Propeller Turbines—"Air Foil Theory of Propeller Turbines and Propeller Pumps, With Special Reference to the Effects of Blade Interference Upon the Lift and the Cavitation," F. Numachi. *Soc. Mech. Engrs. Jl.* (Tokyo), vol. 31, no. 136, Aug., 1928, pp. 530–583, 4 figs. Writer has deduced the theory of these machines in light of theory of airfoil; preliminary study of theory of blade interference; theory of propeller turbines and of propeller pumps. (In English.) E.I.S. serial no. 28, 45,265. (See comment 11.)

45 Propeller Turbines—"Theory and Design of a Propeller Turbine," O. Miyagi, *Soc. Mech. Engrs. Jl.*, vol. 31, no. 136, Aug., 1928, pp. 607–615, 4 figs. Forms of casing and hub are first given, and streamline forms in clearance space are determined by making use of cylindrical coordinates; inflow velocities into runner are then calculated from streamline, and entrance and exit forms of runner vanes, to satisfy condition of minimum hydraulic losses, are determined; expressions for developing moment and power are deduced; graphical solution is introduced. (In English.) E.I.S. serial no. 28, 45,002.

46 Propeller Turbine—"The Hydro-Mechanical Fundamentals of the Axial Propeller Turbine," by Prof. Dr. Alfred Lochner and Ing. Robert Zimmermann, *Wasserkraft Jahrbuch*, 1927–28.

47 Propeller Type Turbines (Mathematical Dimensions of)—"The Propeller Type Turbine" ("La turbina helice"), J. de Ugalde Agundez, *Revista de Obras Publicas* (Madrid), vol. 76, no. 11, June 1, 1928, pp. 198–200, 8 figs. Graphs and mathematical discussion on efficiency of newest designs of turbine blades for utilization of waterfalls of slight drop. E.I.S. serial no. 28, 25,636.

48 Research—"Résumé of Interesting European Research," by B. R. Van Leer, presented at Annual Meeting, A.S.M.E., Dec., 1928.

49 Testing, Sweden—"Arrangement for Testing of Turbines at the Machine Works in Kristinehamn" ("Anordningar for provning av turbiner bid Verkstaden Kristinehamn"), H. Lind., *Tekniska Foreningens I Finland Forhandlingar* (Helsingfors), vol. 48, no. 4, April, 1928, pp. 72–81, 7 figs. Describes construction and methods of testing station for water turbines from 50 to over 1000 r.p.m., output from 20 to 90 hp.; illustrations of laboratories and accessories. (To be continued.) E.I.S. serial no. 28, 17,512. (See comment 12.)

50 Testing, Sweden (Continuation of foregoing article)—"Arrangements for Testing of Turbines at the Machine Works in Kristinehamn" ("Anordningar for provning av turbiner vid Verkstaden Kristinehamn"), H. Lind, *Tekniska Foreningens I Finland Forhandlingar* (Helsingfors), vol. 48, no. 5, May, 1928, pp. 103–108, 4 figs. Causes of cavitation; description of cavitation laboratory completed in 1924 and its testing methods; description of high-pressure laboratory and its operation. E.I.S. serial no. 28, 25,814. (See comment 12.)

51 Research—"Experimental Research in the Field of Water Power," by D. Thoma, paper no. 100 of the First World Power Conference, London.

52 Cavitation in Water Turbines—"Hydraulic Problems," by D. Thoma, published by Julius Springer, Berlin, 1926. (See comments 6 and 7.)

53 Model Turbine Tests—Description of three laboratories built by Verkstaden, Kristinehamn, Sweden, for testing large model turbines, small model turbines, for investigation of cavitation on small model turbines, and for testing impulse wheels. See appendix 4, "Hydraulic Laboratory Practice," A.S.M.E. (See comment 12.)

Comments on Hydraulic-Turbine Articles: 6 Refers to "Hydraulische Probleme" ("Hydraulic Problems.") 7 Worthy of study in connection with problem of corrosion. 8 May be of assistance to designers of hydraulic turbines, but should be further reviewed. 9 Is a most complete article dealing with foreign ideas and practice in hydraulic-turbine design. Some of the headings in this article are as follows: Improvement of Bucket Shapes; Omitting the Runner Band; Venting the Draft Tube; Shaping of Guide Vanes; Recalculation From Model Tests; High-Head Francis Turbines; Kaplan and Propeller Turbines; Efficiency With Adjustable Runner Blades; Head Limits for Reaction Turbines; Allowed Elevation Above Tail-Race Level. 10 Is a subject on which German engineers have been conducting a great deal of research. Their investigations were originally made in connection with the design of airplane wings. This article touches on the connection between tube runner vanes (primarily of the Kaplan and propeller type) and pressure distribution on airplane wings. A study of this article will show the possibilities of research on this subject. 11 Is on the same subject and should be investigated. 12 Descriptive of European hydraulic-turbine-testing laboratories.

Centrifugal Pumps

54 "Analytical and Experimental Investigations Relating to Centrifugal Pumps," by Herbert Mawson, *Proceedings of the Institution of Civil Engineers* (London). (See comment 13.)

55 "Efficiency of Large Centrifugal Pumps for Low Lifts," T. Ono. *Soc. Mech. Engrs. Jl.* (Tokyo), vol. 31, no. 136, Aug., 1928, pp. 584–606, 18 figs. Investigation of effects on efficiency of large centrifugal pumps for low lifts; angle of inlets and exit of vane; dimensions of casing; length of vane. (In Japanese.) E.I.S. serial no. 28, 44,996. (See comment 14.)

Comments on Centrifugal-Pump Articles: 13 Is available in English. 14 Would appear to be of interest to centrifugal-pump designers.

Some of the more important sources from which these data were obtained and to which attention is especially directed are:

1 "Hydraulic Laboratory Practice," by John R. Freeman, published by A.S.M.E.

2 Engineering Index Service, a current publications information service conducted by A.S.M.E.

3 "Hydraulische Probleme" ("Hydraulic Problems"), published by the Scientific Committee of the Society of German Engineers.

4 "Vorträge aus dem Gebiete der Hydro-und-Aerodynamik" ("Lectures From the Field of Hydrodynamics and Aerodynamics"), published by Th. v. Karman and T. Levi-Civita.

The Engineering Index Service has been a fruitful source of information.

The two German books are considered worthy of further careful study. To show the nature and variety of hydraulic problems dealt with, a translation of the respective tables of contents has been prepared, and is as follows:

"Hydraulic Problems"

A scientific survey; lectures given at the Hydraulic Convention in Goettingen on May 5 and 6, 1925; published by the Scientific Committee of the Society of German Engineers.

Introduction, iii.
Preface, v.
Prandtl: Report on the later researches on turbulence, 1.
Foettinger: Examinations of cavitation and corrosion in turbines, centrifugal pumps, and propellers, 14. Elementary characteristics of the cavitation process, 15. Rules for locating places of diminished pressure, 18. Physical processes, 19. Minor physical-chemical effects, 23. Injurious effects of cavitation, 25. Theory of quick corrosion and fractures, 32. Later experiments in cavitation, 34. Results of the experiments, 44. Hydrodynamic examinations, 45. Model laws and rules of similarity of cavitation, 59. Means of avoiding cavitation, 62. Final observation, 64.
D. Thoma: The cavitation in water turbines, 65. Discussion on preceding lectures, 75.
Oesterlen: The development of turbine draft tubes, 111. Discussion on the lecture by Osterlen, 122.
Fluegel: About the approximate understanding of flow losses and the bend problem, 133. Supplement, 155.
Bets: Occurrences on vane ends of Kaplan turbines, 161.
Spannhake: Application of the conformed picture in computations of flaws in turbines, 180. The problem, 180. Edge conditions for the flow, 183. Equation of moments of the turbine, 185. The determination of the whirl flow around the sero-point with circulation around the vane ends, 189. The determination of displacement flow, 190. Layer formation of flow particles, 193. Comparison with infinite number of vanes, 195. Computation of flow pictures, 198. Final observation, 199.
Pfleiderer: The final number of vanes and the impossible operating field in turbines, 201.
Thoma: Abnormal flow in measuring nozzles, 213. Discussion of lecture by Thoma, 215. Index of literature, 216.

"Lectures From the Field of Hydrodynamics and Aerodynamics"

Published by Th. v. Karman and T. Levi-Civita, Innsbruck, 1922.

U. Cisotti: About Italy's share in the progress of classical hydrodynamics in the last 15 years, 1.
L. Prandtl: About the origin of eddies in the ideal fluid, with application to the theory of carrying wings and other problems, 18.
E. Trefetz: Prandtl's theory of carrying planes and propellers, 34.
C. Wieselsberger: The most important results on the theory of carrying wings and tests of the same in experiments, 47.
V. Bjerknes: Computation of forces acting on carrying wings, 59.
E. Pistolesi: New Statements and deductions on the theory of air propellers, 67.
T. Levi-Civita: About the transport velocity in a stationary undulatory motion, 85.
W. W. Ekman: Dynamic laws of ocean currents, 97.
V. Th. Thysse: Computation of tidal waves with considerable friction, 116.
C. W. Oseen: The analytic theory of the equations of motion of an incompressible viscous liquid, 123.
Th. v. Karman: Remark on the question of the form of flow around obstacles with large values of Reynolds formula, 136.
W. Heisenberg: Non-parallel solutions of differential equations for rubbing fluids, 139.
J. M. Burgers: An experiment for estimating turbulent flow resistance, 143.
Th. v. Karman: About the surface friction of fluids, 146.
G. Kempf: The frictional resistance of rotating disks, 168.
M. Panetti: Experimental determination of the distribution of pressure on rotating level planes, 171.
B. Caldonazzo: About the uniform rotation of a solid body in an unlimited fluid, 175.
M. Panetti: Laboratory measurements of the damping moment of an airplane in light vibrations, 186.
L. Hopf: About the dynamics of airplanes, 192.
R. Verduzio: Determination of diagrams of the rise of hydroplanes on the basis of hydrodynamic experiments, 211.
A. G. v. Baumhauer: Contribution to the question of helicopters, 220.
G. Zerkowitz: About flow at velocities beyond sound velocity, 228.
D. Thoma: New perceptions of the hydrodynamics of water turbines, 240.
E. Witoszynski: About the expansion and deflection of jets, 248.

Recommendations by the Committee

The great value of up-to-date information on what is being done in other countries is quite apparent. To the end that the latest published data on research in hydraulics in those countries may be made available to American engineers at the earliest possible moment, it is recommended that—

(1) the Society assume the initiative in making a study of the means and agencies available for translation of the complete text of current foreign publications of hydraulic research, and that

(2) the Society take the necessary steps to develop and coordinate the various agencies to that end.

Research by Water-Wheel Manufacturers

Letters addressed to the various manufacturers asking about research work done by them were invariably replied to in a cooperative spirit. Extensive research work has been done and is still being carried on with the idea of improving the product to meet competition. Under the circumstances it is considered unwise to make public any information gained by individual manufacturers at great expense that could be used by competitors. It was intimated that this field of research should be reserved for the manufacturers themselves because of the large investment of time and money already made, on which there would be no adequate return in the event investigations duplicating work done or to be done by the manufacturers should be carried out and results made public. There is much to be said on both sides of such a question.

Water-Power Companies

A considerable amount of hydraulic investigation has been and is still being carried on by the power companies of the country. From this source a most generous response was received to the requests for a report on research endeavor and accomplishment. This was to be expected, since little commercial benefit could be anticipated from secrecy. On the contrary, aside from the benefit from the application and use of the new knowledge, there would be profit from publicity and good will. A summary of the returns from these letters to the power companies is as follows:

Letters of inquiry sent out	136
Replies received	74
Reporting no research	38
Reporting research	28
Reports promised, but not received	8

Research studies reported are tabulated (in a necessarily abbreviated form), as follows:

Company	Subject	Comment
Cumberland County P. & L. Co.	West Buxton, H. E. Plant	Construction features of 5000-kva. addition
Hydro-Electric Power Commission of Ontario	Engineering research	Regulation, hydroelectric units; governor action, pressure rise, speed rise, velocity of pressure wave in wood-stave pipes; hydraulic losses in large canals and power-plant intakes
Idaho Power Co.	Lifting of rotating element	14,500-hp. element lifted 25 to 32 in. on rejection of load; remedy
	Test of material for turbine runners	"Water bronze" shown preferable to manganese bronze
Marseilles Land and Water Power Co.	Current-meter measurements and discharge capacity of dam	Interrelation, gage heights, slopes and discharge; results not in form for publication
Interstate Public Service Co.	Energy dissipators at toe of dams	Results from use of tumble bay and baffle piers on dams with sand foundations
R. D. Johnson, hydraulic engineer	Johnson - Wahlman draft distributors	Theory and operation of submerged gathering tube for intakes
Kaministiquia Power Co., Ltd.	Factors affecting stream and load changes	Plan of operation of plant
Public Service Co. of New Hampshire	Stream gagings	Gaging stations installed and maintained in their territory
Manitoba Power Co., Ltd.	Development of high-speed propellor turbines	Tests to be made on units
	Flow in 90-mile aqueduct	Tests by Prof. Finlayson of Manitoba University
New England Power Construction Co.	Model tests	Spillways, intakes, valves, discharge channels, flow meters, etc.
Northern New York Utilities	Dams	Tabulation of temperature and seepage through dam
Pennsylvania P. & L. Co.	40,000-kw. hydroelectric development	Construction features of the Wallenpaupack development
Pennsylvania Water and Power Co.	Backwater computations, plant operation, and maintenance	Graphic method for computing backwater curve; experience with various materials for turbine parts; hydraulic experiment and experience data
Rochester Gas and Electric Corp.	Sector gate dams	Construction and operation of sector gate dams
St. Lawrence Valley Power Corp.	Hydraulic losses of the Allen Falls development	Tests on steel pipe line, penstock, racks, etc.
Alabama Power Co.	Increase of head at times of flood, by models, 1 to 24 and 1 to 10 full size	Head increase from 70 to 75 ft. up to floods of 200,000 c.f.s. Trans. A.S.M.E., p. 361, 1922
	Draft tube best for backwater suppressor; on 12-in. homologous runner with various types of tubes	Tube developed to use long horizontal diffuser. Trans. A.S.C.E., p. 893, 1924
	Spillway apron to prevent river-bed erosion; models of 150-ft. dam to scale 1 to 36 and 1 to 100	Developed stilling pool to absorb energy (1925–26, not published)
	Backwater curve for power house location; 1 to 100 model of complete river topography	High-water elevations established locating power house 15 ft. lower than planned (1927–28, not published)
	Design of scroll cases and draft tubes	Designs developed netting from $\frac{3}{4}$ to $1\frac{1}{2}$ per cent increase in efficiency and 5 to 10 per cent saving in cost (1927–28, not published)
Arizona Power Co.	Removal of $MgCO_3$ and other lime products from surface of flume; tests on flume itself	Development of frames with wire brushes and springs shaped to section; towed by current buckets and returned by light cars
U. S. Dept. of Int., Bureau of Reclamation, Denver, Col.	Model studies of concrete dams tested under mercury load; Stevenson Creek test dam	Verified feasibility of design from study of models. Proc. A.S.C.E., May, 1928, part III, vol. 1, of Arch Dam report; 1927 to date
	Determination of data for use in design of dam; installation of pressure pipes and measurement of pressure in dams	Data on uplift pressures under dams on various kinds of foundation. Proc. A.S.C.E., March, 1928, p. 685; tests 1916 to date
	Discharge coefficients for use in design; various heads and openings	Not compiled, tests 1925 to date
	Channel and spillway design; experiments on model and on spillway of Arrowrock Dam	Established design method based on laws of momentum. Trans. A.S.C.E., vol. 89, p. 881; tests 1923–24
	Determining of data for use in determining evaporation losses from reservoir surfaces	Water losses by evaporation more accurately determined. Trans. A.S.C.E., vol. 90, p. 266; tests 1905 to date
	Data for use in design of new irrigation projects by measurement of water use on existing projects	Produced a compilation of water-use data. To be published in Proc. A.S.C.E.; tests 1905 to date
	Valves, reduction in size, and amount of metal, and amount of housing space	Metal in valves reduced over 30 per cent; dimensions of of valves and structures reduced. Washington Specifications No. 480, balanced needle valves for Echo, Gibson, and Coolidge Dams: Studies 1926–28
	More efficient and economical spillway device for Owyhee Dam	Hydraulically controlled floating ring around a central discharge orifice. *Eng. News-Record*, Apr. 28, 1928, p. 663
Byllesby Eng. and Man. Corp., Kentucky	Coefficient of friction, earthfill cofferdam on smooth limestone ledge river bottom; 1 to 10 model, water rising slowly on one side	Sliding resistance as great as overturning; tests 1925
	Comparison, permeability of well-made concrete with concrete given coat of liquid waterproofing; by immersion and increase in weight	No special merit in the waterproofing; tests 1925
	Method of building cofferdam in constricted width with heavy current	Tight forms, 2-ft. concrete wall, braced, held 6 ft. of rapidly moving water; studies 1926
Tennessee Electric Power Co.	Stop leaks through seams in limestone rock under dam	Hot asphalt through electrically heated pipes effective in stopping portion of leakage; tests 1920–24.

	Eng. News Record, May 20, 1926
Removal of sand from water to reduce wear in runner clearances	Water from penstock through centrifugal sand separator admitted to annular space around clearance rings displacing sand bearing water; eliminated sand wear of clearances
To dissipate the energy of spillway discharge to prevent river bed erosion; by models	Concrete tumble-bay designed successfully to destroy kinetic energy
Performance of runner under 35-ft. and 70-ft. head; water flow by salt velocity	Runner designed for 35-ft. head adapted to efficient operation under 70-ft. head; studies 1927

Research by Consulting Engineers

There is an apparent lack of interest among consulting hydraulic engineers to whom questionnaires were addressed, as evidenced by the small number of replies received. Only four answered, and only one of them had anything to report. It is possible that these engineers lack the facilities, the organization, and the time that the large companies and engineering colleges have, and that they have to depend upon the published information from these latter sources. The researches reported by the one engineer referred to are:

Albert C. Arend, consulting engineer, Omaha, Neb.	Study of deposition and erosion for tailwater conditions during seven years at a small hydroelectric plant on the Cedar River at Fullerton, Neb.
	Deposition of rock and silt bars from flood flows and pondage back of dam on North Branch of the Big Blue River at Milford, Neb.; continuity of flow and effect of same on tailwater elevations at the upper or Milford Dam
	Water-surface gradients between Holmesville and Blue Springs Dams, 8 miles apart in the Big Blue River, Neb.; to settle a controversy on tailwater interference.

Research by Hydraulic Laboratories

Replies to questionnaires were received from twenty-nine colleges, two commercial laboratories, and one consulting engineer. The reports from these, as summarized, show thirteen of the colleges carrying on no research whatever and many of the rest only a limited amount of work. The commercial laboratories carry on research in only a limited field, and the results are not made public. A digest of the replies to questionnaires is as follows:

Armour Institute of Technology, Chicago, Ill., Prof. G. F. Genhardt	Nothing because of limitations of equipment
Associated Factory Mutual Fire Insurance Companies, Boston, Mass., H. B. Stewart, Asst. Engr.	Hydraulic work done by these laboratories is for the most part tests on valves and other equipment used in fire-protective equipment
The Polytechnic Institute, Brooklyn, New York, Prof. H. P. Hammond	Not made any researches in hydraulics
Brown University, Providence, R. I., Prof. W. H. Kenerson	Commercial tests on venturi meters and one or two syphon spillways, as well as discharge from special model weirs
California Institute of Technology, Pasadena, Calif., Prof. R. L. Daugherty	Some work on siphon spillways; flow of water from street gutters through openings in the curb into the storm drains below
Case School of Applied Science, Cleveland, Ohio, Prof. R. H. Danforth	Laboratory not planned for research work; none done
Cochrane Corporation, Philadelphia, Pa., Percy S. Lyon	No research information
Colorado College, Colorado Springs, Col., Prof. F. M. Okey	No research work
Columbia University, New York City, Prof. R. T. Livingston	No real non-professional research. Commercial tests and investigations: (1) Rotary-pump development tests for J. B. Russell & Assoc., New York City; (2) calibration of large-ratio pipe line orifices for fluid meters, for Leeds & Northrup Co., Philadelphia
Harvard University, Cambridge, Mass., Dean H. J. Hughes	(1) Effect of corrosion on the discharge of small pipes of different materials together with an investigation of the roughness of pipes; (2) the hydraulic properties of filtering materials
Robert E. Horton, consulting hydraulic engineer, Voorheesville, N. Y.	Experimental study of flood waves; experiments on the permeability of soils
University of Illinois, Urbana, Ill., Prof. M. L. Enger	Comparative tests of friction, losses in cement lines, and tar-coated pipe; the variation of pressure in a manifold pipe; experiments on relief valves; carrying capacity of vertical pipes; piezometer in pipe lines
State University of Iowa, Iowa City, Iowa, Prof. Floyd A. Nagler	Flow of water through culverts; investigations on the accuracy of various types of current meters; experiments pertaining to the discharge of the Keokuk spillways; flow of water around bends, flow of water around bridge piers; weir investigations; tumble-bay and hydraulic-jump experiments; flow conditions at the free outlet of conduits; river water-surface elevations; hydraulic-jump experiments; backwater suppression and head-increaser experiments; flood-wave experiments; hydraulic tests on flap gates; tile-alignment tests; effect of diagonal channel obstructions
Johns Hopkins University, Baltimore, Md., Prof. J. H. Gregory	Work not done on sufficiently large scale to be of importance
Lafayette College, Easton, Pa., Prof. Lynn Perry	Flow of water and air in pipes and through standard connections such as valves, published in *Engineering News-Record*, May 29, 1924, and Feb. 12, 1925; loss of head in standard valves, open and partly open and standard pipe of various sizes; this work has been done for loss of head in the flow of water and the flow of air; this latter will be published shortly; also some research work in connection as well as the structural strength of various types of materials of which open channels are usually constructed
Lehigh University, Bethlehem, Pa., Prof. R. J. Fogg	Experiment run to determine the evaporation of water under varying conditions
Leland Stanford University, Stanford University, Calif., Prof. Charles Moser	The relation between the coefficient of friction f and the turbine factor Vdw/u for fluids flowing in pipes of annular cross-section; stream flow in impulse water-wheel buckets; an experimental investigation of hydraulic air compression; investigation of losses at changes of cross-sections in open flumes
Ohio State University, Columbus, Ohio, Prof. Horace Judd	Study of water flow through concentric, eccentric, and segmental sharp-edged orifices and rounded-approach flow nozzles, located in 10-in. and 15-in. pipe lines, for drops across the orifices and nozzles of from 2 to 9 ft.
University of Pennsylvania, Philadelphia, Pa., Prof. W. S. Pardo	Tests of a large number of venturi meters from $^{6}/_{8}$ in. by $^{7}/_{16}$ in. to 30 in. by 18 in., under varying conditions, published in *Engineering News-Record* and in paper by J. W. Ledoux, April, 1927. Tests of a plain pitot tube in 4-in. and 7-in. pipes; of Simplex round pitot tube in 3-, 4-, 6-, 10-, and 12-in. pipes; of Simplex flat pitot tube in 3-, 4-, 6-, 10-, and 12-in. pipes; of collins pitot tube in 6- and 10-in. pipes; of 5- and 2-ft. suppressed weirs; of 1-, 2-, and 3-ft. contracted weirs; of 60-, 90- and 120-deg. V-notch weirs; of a number of compound weirs; of 2-ft. suppressed low weir; of standard circular orifice from 0.05 to 0.6 ft. diameter, head up to 100 ft.; of 3-in. by 24-in. rectangular orifice; of a 4-in. by 6-in. rectangular orifice; of 12-in. by 8.825-in. orifice pipe meter; of 6-in. by 3.75-in. orifice pipe meter; of 12-in. by 4.75-in. orifice pipe meter; of 10-in. by 4-in. orifice pipe meter; of 12-in. by 8.874-in. orifice pipe meter; of 10-in. by 7.249-in. orifice pipe meter; loss of head in $1^{1}/_{2}$-in. brass pipe at low velocities to determine the value of the critical velocity; loss of head in 4-in. cast-iron pipe; loss of head in 6-in. cement-asbestos pipe; coefficients of conical and trumpet adjutages for 3 to 21 deg.; tests of models of penstocks of turbines and suction pipes of centrifugal pumps, to determine coefficients of full-sized plant; also test of pumps, turbines, service meters, V-notch meters, Simplex meters, and other hydraulic machinery
Pennsylvania State College, State College, Pa., Prof. E. D. Walker	No research on account of lack of facilities
Princeton University, Princeton, N. J.	No research at Princeton
Purdue University, Lafayette, Ind., Prof. F. W. Grove	Calibration of circular orifices under heads of less than 1 ft.; calibration of triangular weirs; calibration of semi-circular weirs; determination of the discharge from horizontal pipes by the coordinate method
Rensselaer Polytechnic Institute, Troy, N. Y., Prof. G. K. Pals-	A study of the effect of the end shape on the towing resistance of a barge

grove	model; experimental investigation of a weir having a crest shaped as a portion of a nozzle
Stevens Institute of Technology, Hoboken, N. J., Prof. R. M. Anderson	No research
Syracuse University, Syracuse, N. Y., Dean Louis Mitchell	No research
University of Texas, Austin, Tex., Prof. S. P. Finch	No research
University of Toronto, Toronto, Ont., Canada, Prof. Robert W. Angus	Extensive hydraulic laboratory; excellent testing flume for work on model water turbines up to 15 in. under 9-ft. head; installing Kaplan turbine for testing work; tests on standard short tube, the effect of air leakage on characteristics of centrifugal pumps, studies of surge tanks, tests on intakes for power plants, on venturi meters under unusual conditions and settings, friction losses in hose, etc.
U. S. Naval Academy, Annapolis, Md., Capt. H. C. Dinger	Tests of high-speed single-stage boiler-feed pumps, made to determine performance, characteristics, and general suitability for use in the naval service; results are generally confidential
Washington University, St. Louis, Mo., Prof. J. L. Van Ornum	No research
Yale University, New Haven, Conn., Prof. R. H. Suttie	No research
R. L. Parshall, irrigation engineer, U. S. Dept. of Agriculture, Bureau of Public Roads, Washton, D. C.	Pamphlets: "Flow Through Submerged Rectangular Orifices, With Modified Contractions," by V. M. Cone; "Frictional Resistance in Artificial Waterways," by V. M. Cone; "The Colorado Statute Inch and Some Miner's-Inch Measuring Devices," by V. M. Cone; "Divisors," by V. M. Cone; "The Venturi Flume," by Ralph L. Parshall and Carl Rohwer; "The Improved Venturi Flume," by Ralph L. Parshall; "Water-Stage Recording Instruments," by R. L. Parshall; "Construction and Use of Farm Weirs," by V. M. Cone; "A New Irrigation Weir," by V. M. Cone; "The Farmer's Short-Box Measuring Flume," by Carl Rohwer, also photos.
University of Wisconsin, Madison, Wis., Prof. L. H. Kessler	Pamphlets: "Experiments on Loss of Head In Valves and Pipes of One-Half to Twelve Inches in Diameter," investigators, C. I. Corp and R. O. Ruble; "Discharge Over Weirs in Side of Channel," investigator, C. I. Corp; "Study of the Variations of Stream Flow and Its Effect on Utilization," investigator, D. W. Mead; "Study of Air-Lift Pumps with Special Reference to Pipe Losses," investigators, C. N. Ward and L. H. Kessler; "Loss in Bends, U's, and Reverse Curves," investigators, C. I. Corp and T. M. Niles; "Hydraulic Ram Tests," investigator, George J. Davis, Jr.; "Loss of Head Due to Sudden and Gradual Changes of Pipe Section," investigator, C. I. Corp; "Discharge Through Orifices on End of Pipe Line," investigator, C. I. Corp.; "Experimental Study of Hydraulic Jump," investigator, C. I. Corp; "Capacity and Loss of Head in Fire Hydrants," investigator, C. N. Ward; "Pitot Tube Studies," investigators, C. N. Ward and L. H. Kessler; "Study of Submerged Weirs," investigators, C. I. Corp and Glen N. Cox; "Study of Wrought-Iron-Pipe Friction," investigator, L. H. Kessler; "Study of Flow Over Triangular or V-Notch Weirs," investigators, C. I. Corp and R. J. Piltz; "Resistance to Flow Through Locomotive Water Hydrants and Relief From Water-Hammer Pressure Due to the Closure of Cylindrical Valves," investigator, L. H. Kessler; "Experiments on Loss of Head in U-Pipe, S-Pipe, and Twisted-S-Pipe Bends," by C. I. Corp and H. T. Hartwell.
Alden Hydraulic Laboratory, Worcester Polytechnic Institute, Worcester, Mass.	"Salt velocity method of water measurement," developed here, and research work on it more or less continuous since; description of this method appears in *Trans. A.S.M.E.*, vol. 45, paper 1902, p. 285; comparative draft-tube tests made for the Alabama Power Company, appearing in the *A.S.C.E. Trans.*, vol. 77, p. 893. The following research work has been done, but there is no published account: Extensive experiments on piezometers, a paper to be written; tests on model dams, spillways, intakes, diversion channels, tail races, river beds, etc., have been made for several power companies; analysis of forces on butterfly valves; tests on a vertical intake shaft for the Massachusetts Metropolitan Water Supply; a large amount of research work on venturi meters, both standard in the laboratory and venturi-meter sections in the field; work on pitometers and pitot tubes
Edward J. Hoff, U. S. Dept. of Agriculture, Bureau of Public Roads, Berkeley, Calif.	Several pages of notes concerning current meters

Power—Steam or Hydro, or Both

By WM. W. TEFFT,[1] JACKSON, MICH.

Emphasis is placed on the necessity of including all pertinent factors in a comparison of steam with hydro power, drawing attention to the fact that any conclusions based upon a part of these factors will be erroneous, misleading, and valueless. In considering the various factors that go to make up the final conclusion attention is called to the superiority of steam power where need for power is immediate, where the more profitable hydro sites have already been developed, and where the actual load factor is very high. It is also made clear that hydro power is superior in reliability and as reserve capacity, and can operate to advantage at a much lower load factor than can steam. It might be said that the paper deals largely with load factor. Curves based upon a large number of developments in each case indicate that usually for 60 per cent load factor or less the hydro plants are superior, the statement being made that no steam plant can now be built for 25 per cent load factor that will produce current for 1 cent per kw-hr., but there are many water-power plants, most of them in fact, that can do this.

It is concluded that there is ample proof that a power system combining both steam and hydro plants can generate power cheaper than either kind of prime mover.

SO MUCH has been written in the last decade, and particularly recently, about the relative merits of steam and of hydro power and their cost that it would seem as though the various subdivisions of this subject had been presented adequately. If this paper succeeds in setting up a simple and accurate way of determining when to build steam or hydro or both, and how much of each, its purpose will have been accomplished. The apparently widespread propaganda in the semi-technical and popular press in favor of steam power, with its benefits to miners, mine owners, railroads, and manufacturers of steam equipment, as well as politically, is not a part of this discussion, but only the overall cost of power delivered to the customers' meter, which is where the revenue is collected.

Whether steam has been coming into its own is practically settled by the facts that hydro power last year was just over 40 per cent of the public-utility total, and that it increased 20 times over the previous 25 years, whereas steam increased but 14 times over the same period. (See Bibliography, 38.) Whether steam or hydro or a combination of both will be developed in the future should depend, it would seem, on which of these three will deliver the cheapest reliable power to the customers' meter. In the interest of simplicity the problem also will be limited to electric power for public-utility systems, since it represents some 55 per cent of the power of the country, which is as much as all the rest of the world (35) and is typical of most all other power development. Also electric power systems make possible the most advantageous use of both kinds of prime movers. The problem resolves itself into two factors: 1 Economics of a combined system. 2 Determination of proportion of steam and hydro in such a combined system.

[1] Consulting Engineer, Vice-President, Fargo Engineering Co. Mem. A.S.M.E.

Contributed by the Hydraulic Division and presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

ECONOMICS OF A COMBINED SYSTEM

In general it has been conclusively shown and described in the technical press that the cost of power in a combined system is less than it would be if the total amount were supplied by either (10, 16, 23, 27, 33, 36, 38, 44); this is evidenced by the existence of many combined systems representing as they do the conclusions of separate groups of engineers. This is only evidence, as the solutions of yesterday will not do today or for tomorrow. The trend, however, is indicated. The total cost of steam power, including all fixed and operating charges, has not lessened greatly in the last few years, the gain in efficiency being just about offset by the fixed charges on its increasing cost and the increasing cost of fuel (44, p. 34). Unit hydro construction costs have increased somewhat, and a high efficiency of over 92 per cent has obtained for some time. Just here it should be noted, though, that the rapidly changing design of steam plants introduces a very high obsolescence charge to offset in some large proportion their gain in efficiency. There is scarcely a steam plant in the country ten years old but is obsolete. Hydros have permanence. Few, if any, well-designed hydroelectric plants ten years old but are turning out power cheaper than new steam plants in their neighborhood. Of course the cheaper dam sites were developed first.

DETERMINATION OF PROPORTION OF STEAM AND HYDRO

Now to attempt here a definite determination of the proper ratio of steam and hydro in a system, or even to illustrate with a specific instance, would be misleading and valueless. A parallel checking list of pertinent items is intended to set up a weighted outline that may be applied to specific cases to arrive at the proper proportion of steam and hydro. It cannot include everything, but is based upon intimate and extensive experience and avoids prejudice and undue emphasis on isolated factors. Usually the most economical ratio is obtained when some spilling of power water is necessary during high-river stages.

COMPARISON OF STEAM AND HYDRO

This checking list is as follows:

CONSTRUCTION COST

Steam.—Steam usually less than hydro per kilowatt of machine capacity. Should include transmission investment necessary to deliver current to place in system comparable to hydro (34, p. 1207; 30).

Hydro.—Hydro usually more than steam per kilowatt of machine capacity. Should include transmission investment necessary to deliver current to place in system comparable to steam. One of the most promising opportunities for the reduction of construction cost lies in reducing interest during construction.

TIME REQUIRED FOR DESIGN

Steam.—Should be designed as construction proceeds to avoid obsolescence amounting to close to 10 per cent per year. This increases construction cost, but not as much as the 10 per cent.

Hydro.—May have ample time, as efficiency does not change much from year to year.

FINANCING

Hydro.—Hydro securities are more easily sold than for steam. The amount of earnings allowed by utilities commissions is based upon the physical value of the property, say 8 per cent of it. Then as hydro costs more per kilowatt than steam its owner is automatically allowed to earn more than on a kilowatt of steam, perhaps three times as much. This on the premise that the kilowatt-hour cost is the same as or less than with steam.

TIME REQUIRED FOR CONSTRUCTION

Steam.—About half as long as hydro.

Hydro.—About twice as long as steam.

POPULARITY

Steam.—Smoke and cinders and ash are unpopular. Also, the warming of streams is not desirable. Manufacturers sell more equipment per kilowatt for steam than for hydro. Coal miners, mine owners and railroads receive greater benefits.

Hydro.—The hydro is usually popular because: (*a*) Its cost is largely spent in its neighborhood. (*b*) Fishing is usually bettered. (*c*) The shoreline

becomes available for summer homes. (*d*) Taxes are more than for steam. (*e*) Securities seem to sell easier.

Fixed Charges

For purposes of comparison to determine kilowatt-hour costs, fixed charges should be combined with operating charges and load factor.

Steam.—Interest, 6%; taxes, 2%; depreciation and obsolescence, 10% (the present economical life of a steam turbine is 40,000 hours or 5.5 working years). (See Bibliography, 50.) Total, 18%.

Hydro.—Interest, 6%; taxes, 2% or less; depreciation and obsolescence, 2%. Total, 10%.

Operating Charges

For the purpose of comparison to determine kilowatt-hour costs, operating charges should be combined with fixed charges and load factor.

Total Charges

Only total charges are logically comparable. For purposes of comparison to determine kilowatt-hour costs, total fixed and operating charges should be combined with load factor.

Reliability

Steam.—Steam plants are not so reliable as hydro, thus requiring more reserve capacity on this score.

Hydro.—Hydro plants are mechanically two and one-half to six times as reliable as steam plants at their respective switchboards, and from two and two-tenths to five times as reliable if the interruptions of 100 miles of modern transmission line are added to the hydro plant. (Wm. W. Tefft Report, January, 1924; 7, N.E.L.A. Report, June, 1925; 12, N.E.L.A. Report, May 17–21, 1926; 20, N.E.L.A. Report National, February, 1927; 22, Hydroelectric Handbook, Creager and Justin, p. 874; 40, N.E.L.A. Report, May, 1929.) Hydro plants need reserve prime movers for low-water periods. These may be steam or hydro pumping plants.

Reserve Capacity

In determining the kind of reserve capacity hydro pumping plants are more reliable and usually cheaper than steam.

Steam.—Requires reserve capacity on account of inferior reliability of steam. Say, add 20 per cent to plant capacity for reserve; this is irrespective of load factor as a steam plant has no storage as does a hydro.

Hydro.—Requires reserve capacity on account of low water. Load factor affects the amount to be determined for comparative purposes. For a plant operating at 50 per cent load factor, the reserve capacity will need to be but one-half as much as for 100 per cent load factor, assuming pond storage for 12 hours.

For comparative purposes, hydro usually requires approximately 30 per cent of its capacity, net as reserve over steam.

Pumped Storage Hydro Plants

Steam.—Pumped storage plants, where economically feasible, make ideal reserve for steam plants as they are immediately available in emergency and improve the load factor on the steam plants.

Hydro.—(31, 39, 45, 49)—Where topographic conditions permit, hydro pumped storage plants can be built for from $60 to $80 per kilowatt and having an overall efficiency of 60 per cent. They often are economical for hydro reserve as well as for steam.

Load Factor

Steam.—The cost of steam power is greatly affected by change of load factor. A plant producing power at a total cost of 0.75 cent per kilowatt-hour at 50 per cent load factor will not be able to better 1.15 cents per kilowatt-hour at 25 per cent load factor, an increase of 53 per cent in cost for a change of but 25 per cent in load factor.

Hydro.—(Fig. 3; 29, p. 475; 36)—Hydro is very much superior to steam as variation of load factor has but a negligible effect on the cost of power per kilowatt-hour. Because of this, the cost of power from interconnected system plants is greatly reduced by the hydro plants.

The foregoing list of items to be considered in any competent study of the relative merits and value of steam and of hydro power, together with the abbreviated comments on each, is intended to emphasize the error in trying to arrive at a conclusion by considering only part of these items. No conclusion of value can be reached by considering that fixed charges on hydro are greater than steam or by considering only that operating charges on steam are greater than for hydro power. Nothing less than a summary of all of the factors involved can be of any great value in determining whether steam or hydro power is better. Construction costs, financing, time required for construction, popularity of securities, fixed charges, total charges, reliability, reserve capacity required, pumped storage hydroelectric plants, and last, but by no means least, load factor, all are vital parts of such a determination.

The construction costs, time required for design, financing, and time required for construction, all have a bearing on the first cost. While the first cost of steam per kilowatt is usually less than for hydro, the operating charges tend to equalize them, so that the summation when considered on the basis of total annual kilowatt-hour may show little difference. During the last few years the gain in steam efficiency has been very marked and rather uniform, while the gain in hydro efficiency has been very little, as there is not much more to gain. This gaining of steam efficiency has had the effect of increasing the percentage of fixed charges applicable to steam plants due to obsolescence. It is not a part of this paper to go extensively into the matter of depreciation and obsolescence, as it is a subject by itself, but there are few if any steam plants in this country ten years old but are obsolete although perhaps in perfect mechanical operating condition. The present economic life of a steam turbine is but 5.5 working years (50), so it would seem that the economics of any new power installation on a system, steam as well as hydro, should be based upon the system load factor rather than upon any assumption that the new turbine on account of its high efficiency will carry the base load for a term of years. The next new plant would take its place. Thus the depreciation and obsolescence item for steam would be at least 10 per cent, whereas for permanent high efficiency hydro plants 2 per cent is ample,

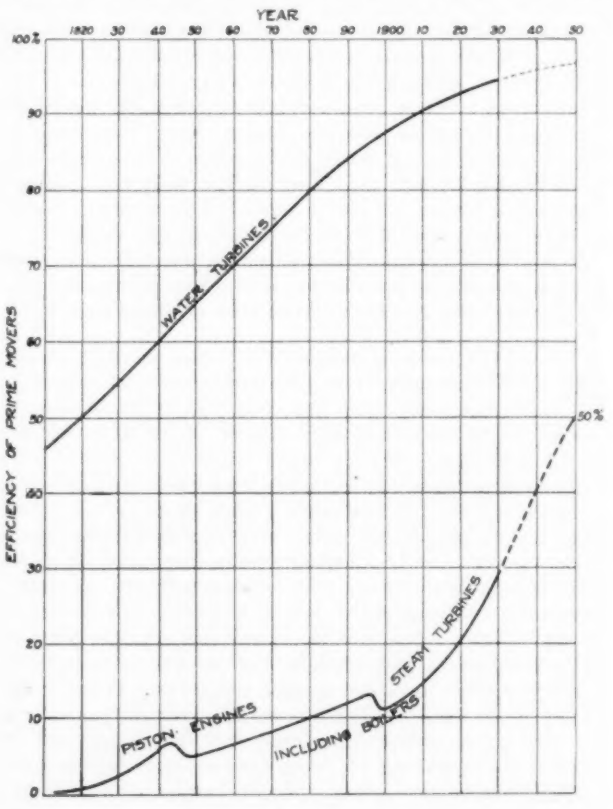


Fig. 1

making the total fixed charges for hydro 10 per cent and for steam 18 per cent.

Relative reliability is of the utmost importance. According to experience and the figures representing composite data for the entire country, steam stations may be expected to have an outage of 8.2 per cent of the time when their service is really needed as compared with a hydro plant and its modern 100-mile transmission line to load center, having a similar outage of 1.33 per cent of the time, which gives a ratio of 6.16 (W. W. Tefft Investigation, 1923–24; 7, 12, 20, 22, 40).

Even if the figure for the steam unit only is used, without any penalty for all the rest of the steam station, there is a ratio of 3.34 to 1.33 or 2.5. On the basis of these figures it seems conclusively proved that a hydro plant with its 100-mile transmission line and accessory apparatus will have only one-fourth or one-fifth of the outage of necessity as a steam plant at the load center. Also, it should be borne in mind that with the ever-

increasing size of steam units the practice of carrying one unit in reserve is a much greater financial penalty than holding a hydro unit in reserve, since the latter is usually of much smaller capacity with proportionately smaller fixed charges.

The inferiority of reliability of the steam plant as compared with the hydroelectric plant and its transmission line is, the author believes, quite typical of the two methods of generating power. A steam plant, with its multiplicity of machines operating at high speeds, high pressures, high temperatures, high metal stresses, and severe and repeated temperature changes, hardly could be expected to be as reliable as the extremely simple hydroelectric units operating at low speed.

A hydroelectric plant of three units can be put into operation in an emergency from no load (stationary) to full capacity in less than four minutes, while a steam plant, even with the boilers banked, would require thirty minutes or more, usually one hour.

The matter of superior hydroelectric reliability extends beyond the hydroelectric plants themselves, and renders the steam plants and also the transmission lines more reliable, inasmuch as an interruption to service in a steam plant can be made up for ten hours or more by the hydro pondage, and likewise an interruption of one transmission line can be made up for by drawing on the stored water and by the increased loading of another transmission line.

Both steam and hydro plants require reserve capacity, but for different reasons: the hydro on account of decrease in stream flow, and the steam on account of inferior reliability. The reserve capacity required for hydro plants on account of lack of reliability is negligible, inasmuch as they have been shown to be 98.7 per cent reliable when needed. If in the comparison of this item the hydros could be charged with all of the reserve required on account of diminishing stream flow, then the steam should be charged with all of the reserve required for any purpose, such as annual shutdown for inspection and cleaning of boilers, and all plant apparatus affecting the kilowatt output; 20 per cent reserve for an all-steam-generating system would appear reasonable. Reserve for hydro plants depends upon the load factor at which they use the available water, which, at low water time under economic operation, would not be more than 50 per cent. The reserve capacity required to keep the needed output up to the plant capacity would be the difference between plant capacity and the power in kilowatts that low water could develop at 50 per cent load factor. For comparative purposes hydro in the Middle West requires approximately 30 per cent of its capacity as a reserve over what a steam plant would require for the same capacity.

The reserve capacity may take the form of steam plants designed cheaply for that kind of service only; the utilization of steam plants obsolete for base load purposes; additional hydro plants or pumped storage hydroelectric plants. Of these the pumped storage hydroelectric plants, where topographic conditions permit, appear to have merit.

"There has recently developed an intensified interest in the economic possibilities of peak-load hydroelectric plants, making use of water which has been wholly or in part pumped into storage above such a plant during off-peak periods. This has come about through the recent completion of the Rocky River plant of the Connecticut Light & Power Company near New Milford, Conn., which has at present one 24,000-kw. unit and provision for a second unit of equal size. While it was well known that some 45 plants of this kind had been constructed abroad, it seems that it required the evidence of a successful venture in this country to stimulate power engineers to a detailed study of possible locations for similar peak-load plants in their own territories, only three in this country having come to attention—namely, the Rocky River plant, the Mill Creek plant in Utah, and the Tule River plant in California. A number of large hydro pump plants are being designed in the United States, and construction will probably be undertaken in some instances in the near future." (49, H. S. Hunt.)

They cost usually from $60 to $80 per kilowatt and make ideal reserve for steam plants as they are immediately available in emergency, and improve the load factor on the steam plants.

This storing of energy by means of pumping water makes feasible the development of tidal power at costs below five mills per kilowatt-hour and at either low or high load factors.

Load factor is not by any means the only advantage accruing to hydro development, but it is one of the most important. The determination of the relative merits of hydro or steam developments rests on the cost of power per kilowatt-hour delivered into the power system. On a purely annual kilowatt-hour basis there may not be much difference in the Middle West, where, of course, the cheap sites were developed first. This, however, is not quite the way to analyze their relative merits, as under these conditions with the proper analysis hydro will be the cheaper. A hydro plant properly designed can be operated on a very low load factor, particularly at the low-water periods when its power is most valuable. This one characteristic, and there are several others that will not be mentioned here, is of very great importance. By operating the hydro plants on an interconnected system at a low load factor the load factor on the steam plants is greatly increased and consequently bettered, so that one of the greatest advantages of hydro development, and it is an important advantage justifying very careful and thorough study in every instance, lies in reducing the cost of steam power rather than the actual generation of cheaper hydro power.

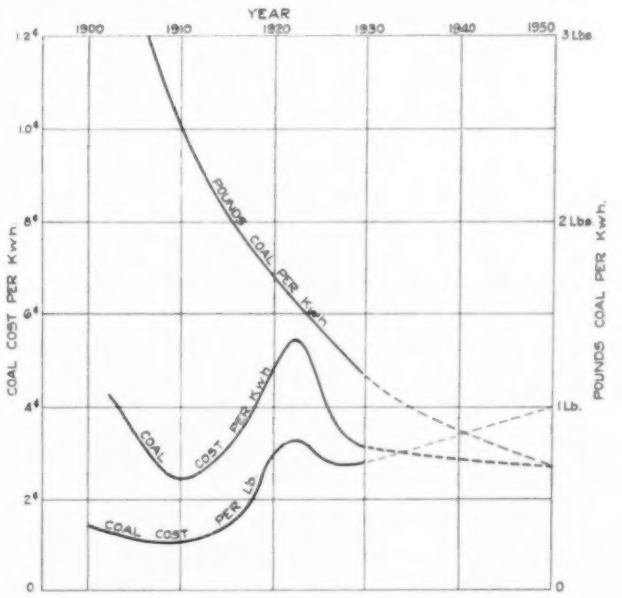


FIG. 2

The two summary curves on Fig. 3, one showing the total steam cost and the other the total hydro cost, plotted against load factor, have a background of some 58 operating hydro plants and over 30 steam plants. From these two curves it will be seen that the cost of steam and hydro power are about equal at 65 per cent load factor on the basis of load factor only, and without giving to hydro any other advantage, such as reliability. For a load factor of around 49 per cent, which is close to the usual system load factor, the direct differential in cost shown

by the curves is some 7 per cent in favor of hydro, without taking any advantage of the bettering of the steam load factor by the hydro.

For instance, if the cost per kilowatt-hour at the proper location on a system, including all operating and fixed charges, is 0.7 cent for both steam and hydro when each is supplying, say, one-half of the required power, the average cost, of course, would be 0.7 cent. If, however, a hydro plant is made to operate at a low load factor, thus bettering the load factor on the steam plant, the cost of the hydro power would still remain at approximately 0.7 cent, but the cost of the steam power might drop to 0.54 cent, thus making an average cost to the system of 0.62 cent, or a reduction of total cost to the system of over 11 per cent.

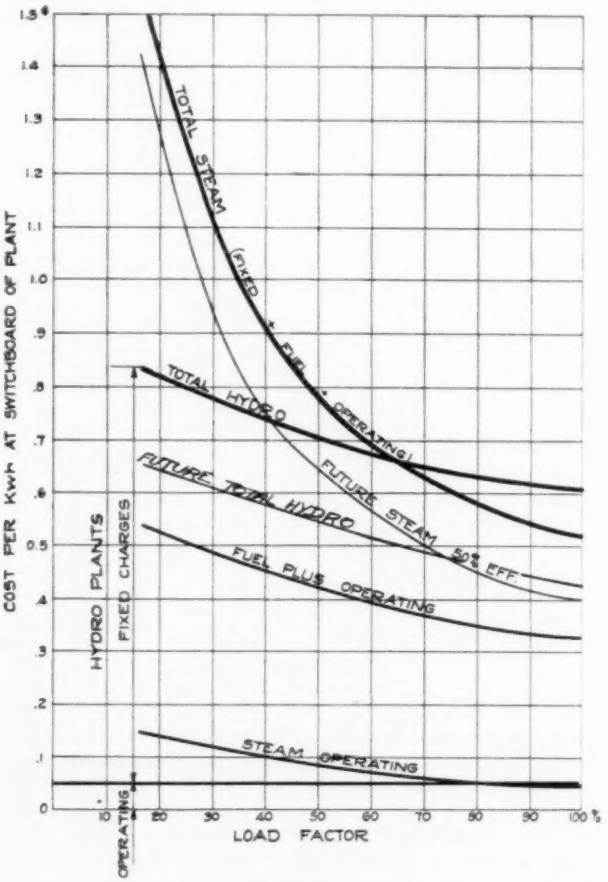


FIG. 3

The cost of power in the future from steam will probably remain about constant, on the assumption that steam-plant efficiencies will gradually increase until they reach 50 per cent in 1950. This assumes that efficiencies will increase somewhat faster in the future than has been the case in the past, although there does not now seem to be much evidence to indicate that the increase will be any faster than it has been. On this assumption, on Fig. 3 there is shown a curve indicating the approximate future cost of steam power for various load factors. Another curve on the same figure indicates the future hydro cost based only slightly upon decrease in construction cost, but mainly upon expected increase of stream flow during the next quarter of a century. In further comparing steam and water power a condition inherent to water power should be emphasized which will materially enhance the value of water power during the next 25 years.

Fuel can be fed to steam stations according to requirement, but in most hydraulic stations water is supplied according to the existing sequence in the natural supply. This natural supply is variable, often within wide limits. The output of hydraulic stations, for instance in Michigan during 1928, was 1,143,397,000 kw-hr. Had the flow of rivers been as in 1925, the output would have been some 400,000,000 kw-hr. less. The return on the investment in a hydraulic station is dependent on the average annual power output and its load factor. Should this average shift to higher levels, the total cost of hydraulic power as denoted by the curve (Fig. 3) will move downward and eventually below the cost of steam power at all values of load factor.

The expectation for the next 25 years is exactly that. The data and considerations involved are too voluminous to be presented here in full, but may be indicated briefly. Gradually it has been realized that trends and oscillations are visible in hydraulic data which follow definite sequences not traceable in weather data. Meteorology deals chiefly with momentary values, while hydraulic data are cumulative. Water is collected on large drainage areas and partially runs off into storage basins or lakes, where it is again accumulated. While weeks of dry weather follow short periods of rain, river flow continues uninterrupted, following only the cumulative values of rainfall. Lakes exhibit important changes of level during long periods, though the faint climate cause is not felt in the weather.

It appears that these small climatic changes, negligible in meteorology but highly important for water-power engineering, can be correlated and used to estimate the future trend. (See U.S.G.S. Water Supply Paper No. 30, by Lane and Horton 1899, p. 28, which contains an estimate of an exactly verified minimum three years in advance.) Based on the researches of Prof. Edward Bruckner (see "Climatic Changes Since 1700," Ed Holzel, Vienna, 1890), they have been extended and utilized for practical purposes (see A. Streiff: "On the Investigation of Cycles and the Relation of the Bruckner and Solar Cycle;" "Notes on Estimating Runoff;" "The Practical Importance of Climatic Cycles in Engineering;" *Monthly Weather Review*, Washington, D. C., July, 1926; March, 1928, July, 1929).

The long cycle appears to have attained its minimum from 1895 to 1915, which minimum extended over the whole North American Continent; during which period the most vigorous development of hydroelectric power occurred. Hydroelectric power is now facing a period of increased flow, which increase for some localities as southwestern Texas and southern Michigan is estimated to average some 40 per cent during the next 25 years. This is due to the amplifying effect of stream flow over rainfall (see Fig. 5). The examples shown indicate that an increase in rainfall of 30 per cent may double the runoff, the latter increasing three times faster than the rainfall.

A 33 per cent increase in runoff would immediately bring the curve of cost of hydraulic power below the cost of steam power, at 100 per cent load factor and twice the present efficiency of steam plants. A 16 per cent increase would reduce the cost of hydraulic power below that of steam plants at all load factors.

It appears, therefore, that the future of hydroelectric power, viewed from this angle and at least for the next 25 years, is bright.

CONCLUSION

There is ample proof that a power system combining both steam and hydro plants can generate power cheaper than either kind of prime mover for the same conditions, thus indicating the mutual advantage of one to the other. The percentage of each must be solved for in each individual case, the emphasis of the solution being upon including all of the pertinent factors

with their proper values. Throughout the country steam power is being developed perhaps three times as fast as hydro power, partly because the better hydro sites were developed first and so the remaining ones are not so profitable, but mainly because this appears to be somewhere near the proper ratio of steam and hydro. Hydro power generated in 1928 is 40 per cent of the public utility total, being an increase of about 8 per cent over the preceding year. In the future steam plant efficiencies will increase; the cost of obtaining them will increase also, and it would seem that the prospect of hydro continuing to maintain its percentage of development is still bright. A study of the prospect of increased flow of rivers, with the magnified effect that this would have upon the hydro-power output and the development of cheap tidal power at almost any desired load factor through the aid of hydro pump storage, indicates rather clearly that it is to be expected.

Many of the hydro developments that appeared at the moment to be most profitable have been developed. As conditions changed, however, some that appeared less profitable then have become more profitable now, such as Conowingo on the Susquehanna River near Philadelphia.

If the progress of the art should eliminate a part or all of the cost of transmission, most of the benefit would accrue to the hydro.

There can be no force to the statement or propaganda that water power is declining when 40 per cent of the power of the nation is hydraulic, and when the largest concentrations of power in this country are hydraulic, and when the future looks fully as bright for hydraulic as for steam. Neither can there be any force in the statement that steam will be superseded when there are localities in which no hydroelectric is possible, nor in the statement that steam is cheaper than hydro when it is more economical in many places on this continent to develop steam with hydro power than with coal. In fact, nothing less than a complete analysis of all of the cold facts will arrive at the proper conclusion. Hydro will continue to be profitable as long as the present conditions of load factor exist. Few, if any, power systems have load factors higher than 50 per cent, and many of them much less. No steam plant can now be built for 25 per cent load factor that will produce current for 1 cent per kw-hr., but there are many water power plants, most of them in fact, that can do this.

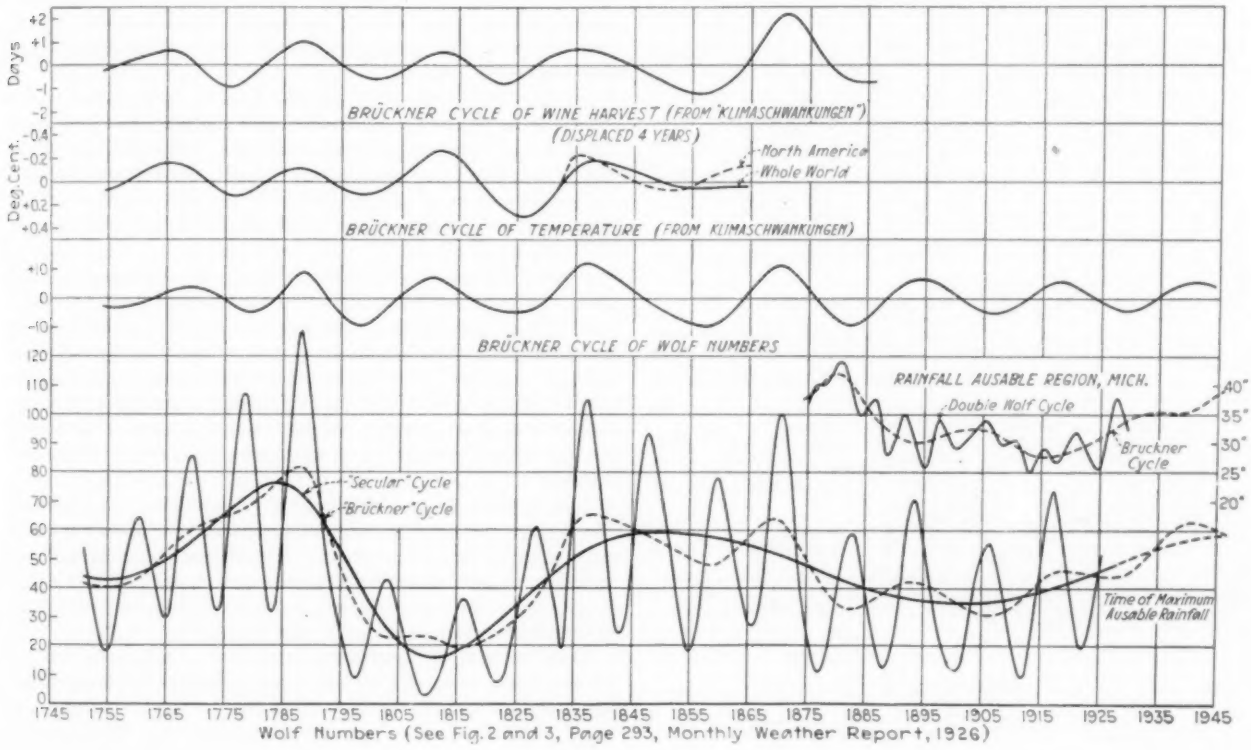


FIG. 4

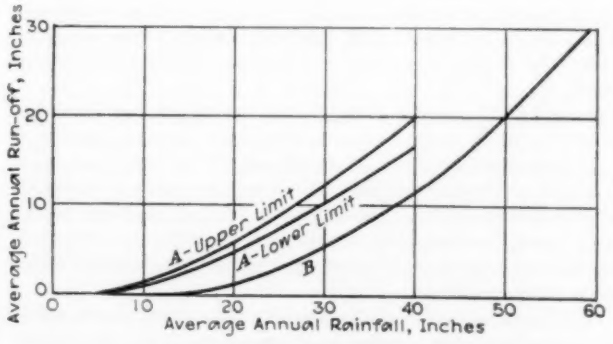


FIG. 5 AMPLIFYING EFFECT OF RUNOFF

(*A*, Ausable River, Michigan, after Streiff; *B*, Devil Canyon Watershed, California, after Sonderegger.)

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Discussion

ARNOLD PFAU.[2] The author has touched upon the subject of "pumped storage." Our country should pay a little more attention to these possibilities, particularly because it will surely prove to be a valuable contribution toward an increase in economy in connection with large, combined power systems.

In Switzerland, the writer understands, they are now introducing the use of a so-called "national bus bar," to which all electric energy can be delivered which otherwise would have to be wasted by private utilities as well as by municipal or government hydro plants.

Owing to old established rights, the benchmarks of the rivers are closely fixed, so that water-power owners must be careful to avoid interference with their neighbors. They must not back up into the tailrace of the party above, and they cannot hold back water which might lower the headwater level of the party below. Thus, many low-head plants operating at night at a very low load factor have to open the waste gates in order to hold the water levels within permissible limits.

With the adoption of the national bus bar, these plants, without any appreciable increase in operating expenses, can deliver the full stream-flow output, thereby not only increasing their own revenue but enabling plants located elsewhere and operated from reservoirs to shut down completely during off-peak hours. Not only can the prime movers of some of these plants be shut down, but surplus energy is available to operate pumps and thus replenish the storage, so that a larger peak-load capacity becomes available.

As our trunk transmission-line system develops, it will also become possible to build pumping storage plants in locations best suited for most economic development, so that the river-flow plants can dispose of their surplus off-peak output.

BLAKE R. VAN LEER.[3] The author has presented in a thorough manner a subject of vital concern not only to the engineering profession, but to our country at large. There are other aspects of this subject which were purposely omitted from his paper because they are intangible and incapable of measurement. The writer refers to the political and social aspects of steam power as compared with hydro.

To one who is familiar with public opinion of the nation as reflected by the actions and discussions taking place in our national Congress, there can be no doubt that there exists in America today a decidedly large and growing determination upon the part of the people to preserve and conserve our natural resources. This attitude is frequently advanced and advocated even though the cost may be exorbitant.

There is not the slightest difference of opinion on the point that every time a ton of coal is burned under a steam plant some small part of our natural resources has gone forever. When power available in our rivers is not used in the generation of hydroelectric energy, every hour that such use is not made means the loss of a portion of our natural resources. In other words, if we do not use our coal today, we may yet use it tomorrow, but if we do not use our hydro energy today, that particular energy has gone forever.

This argument seems to carry great weight with the people of the nation, and they are definitely determined to see that our natural resources are conserved. This means that they will advocate the development of hydroelectric energy even though it does not show a financial return in comparison to coal. This is one of the political aspects of the question.

There is another equally important phase of this question; it is the social aspect. This brings about the development of communities and industries in very interesting and peculiar ways. Most hydraulic engineers are familiar with the situation as it exists in California. There the large cities and popular centers are found along the seacoast, and the hydroelectric energy is located in the mountains several hundred miles to the east. Between the mountain range and the seacoast lies a series of valleys, which, were it not for irrigation made possible from wells and rivers flowing through the territory, would be little more than desert. However, because there is little coal in California and oil is a comparatively recent discovery in that section of the country, hydroelectric plants were built in the mountains and threaded through transmission lines, crossing what were once unproductive valleys, in order to reach the populous centers on the coast which use the principal amount of the energy generated in the mountains.

This made it possible for farmers and many small consumers to have in these arid valleys a cheap source of electrical energy running almost through their fields. The amount of energy which these people at first used was so small that it would have been uneconomical to have built a power line to them, but since the power line must cross their fields in order to get to the cities, it of necessity became available.

Thus we find that a large, useful, and productive civilization has grown up in what were once arid desert valleys in California, and this growth has been produced in no small measure by the development of California's hydroelectric energy. Had steam power exclusively been developed in the large cities in California it would have been many years before the interior valleys would have reached the growth and prosperity which they enjoy today.

[2] Consulting Engineer, Allis-Chalmers Manufacturing Company, Milwaukee, Wis. Mem. A.S.M.E.

[3] Assistant Secretary, American Engineering Council, Washington, D. C. Mem. A.S.M.E.

Thus it is seen that in some special cases the development and use of hydroelectric energy may have a very vital and significant influence upon the social development of a state.

ABRAHAM STREIFF.[4] Never before in the history of hydroelectric power has Europe witnessed such a large-scale expansion of water-power development as at the present time. While in the United States it is being debated whether steam or hydro installations are more economical, in Europe water power is experiencing an unparalleled boom.

This expansion of water power is occurring in the face of the ceaseless refinements in steam power, which now is entering upon an era of higher pressures and temperatures with greater economies, most of which are in advance of American practice. We may mention the high-pressure steam generators of Loeffler, Schmidt, Brown-Boveri, Benson; superheat up to 850 deg. and pressures up to 3200 lb. per sq. in.; the steam accumulator of Ruth; the steady development of the Diesel engine. The turbine locomotive of Ljungstrom is there in daily use; the Swiss Federal Railways now operate locomotives having 850 lb. boiler pressure.

In spite of this intensive development of steam, European hydro is expanding on an unprecedented scale; for the steam plant in its present form has its natural limitations. Every improvement is an added complication to a system already filled with the most varied machinery and piping. Coal consumption decreases, but complication and cost increase. Sixty per cent of the heat equivalent of coal is still wasted in the 450 tons of cooling water that must be pumped for each ton of fuel burned. The removal of soot and cinders assumes costly proportions. With fuel supply subject to strikes and wars and to ever-increasing cost, it is small wonder that hydro power in Europe is at present at the pinnacle of its development.

That situation presages the future of the United States, where now the natural resources are wasted at the most reckless rate ever seen. Coal and oil may be plentiful and comparatively cheap at present, but just as surely as the forests are vanishing, fuel deposits are disappearing, fuel costs increasing, while water power remains undiminished and conserves the land by equalizing and storing floods. These considerations are in Europe already of such importance as to give rise to the construction of an increasing number of huge water-power stations. A few of the latest ones may be named:

Ryburg-Schworstadt on the Rhine; 112,000 hp.; annual output, 500,000,000 kw-h.

Oberhasle, Switzerland; 244,000 hp.; annual output, 538,000,000 kw-h.

Kembs, Alsace; 120,000 hp.

Shannon River, Ireland, 228,000 hp.

Kardaun, South Tyrol; 285,000 hp.; annual output, 500,000,000 kw-h. This plant has a modern settling basin, system Dufour, for a discharge of 3150 c.f.s.

Ill Works, Austria; 300,000 hp.; to be transmitted to the Westphalian industrial region at 220,000 volts. The maximum head of this system is 4400 ft.; the main reservoir also receives water from an auxiliary stream pumped against 1300 ft. head. The transmission line, the first European line of 220,000 volts, will serve to interchange power made from white or black coal as needed. Here we find the author's conclusion of a combined system as the best solution exemplified in a new large development.

The basic causes of such hydro expansion as now occurs in Europe are less pronounced here, but in this country too a boom in water power is in sight. It is caused by an unexpected, old, but newly recognized factor, the climatic cycle.

Long discounted by the meteorologists as unimportant for purposes of weather service, it assumes important dimensions for the future of hydroelectric power supply. The cumulative nature of run-off and the amplifying effect of a stream flow on rainfall changes vary the mean value of stream flow and hence the mean value of power derived therefrom. The great development of hydro in the last 25 years occurred at the lowest point of the long, so-called "secular" cycle, and the expectation for the next 25 years is that stream flow will pass through the high point of that cycle, as it did in the 70's and 80's, and before that.

That future stream flow can be estimated many years ahead has been demonstrated. The late Professor Bruckner, geologist of the University of Vienna, wrote his book, "Climatic Changes Since 1700," in 1887–90. In that work he discusses the future of the climate in the Great Basin, Utah (p. 286), and states: "Undoubtedly the next minimum will occur toward the end of the century (1900). These regions face a time of crop failures which, according to press reports from the United States, has already commenced."

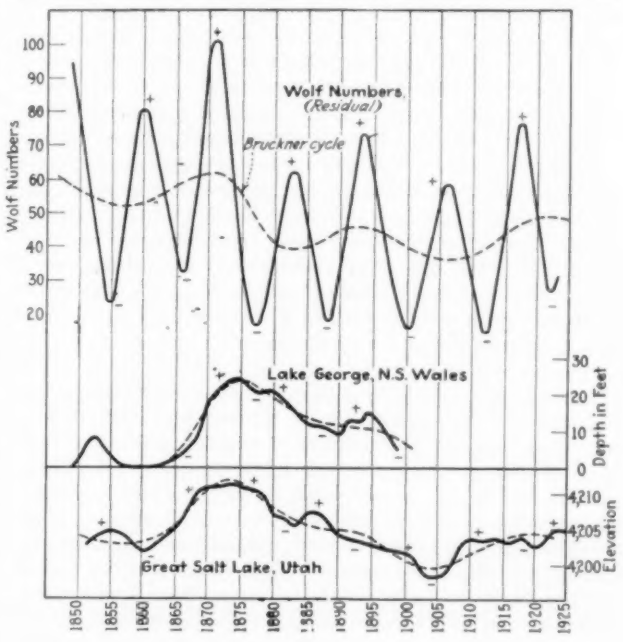


FIG. 6

The graph of the Great Salt Lake (see Fig. 6) shows that the estimate was exactly verified. The lake fell sharpest around 1900 and has risen ever since 1905.

But also in American hydraulic literature do we find examples of such prognostication. In Water-supply paper No. 30, 1899, p. 28, Mr. Robert E. Horton states of rainfall in the Kalamazoo region, Michigan: "A fifth minimum may be expected around 1901." This expectation pronounced three years ahead was also exactly verified.

The increase in stream flow is expected to culminate around 1945. This does not mean that the flow of streams will be uniformly at high levels, but the mean values will be higher than at present and with these the output of water power. In lower Michigan the run-off, around 1945, is expected to average double of that around 1910.

It is of interest to compare the history of water and steam power in this region. In 1912 a dam was built on the Ausable River. This dam is now, after 17 years, in perfect running order with its original machinery (maximum 85 per cent efficiency) in place. It was constructed for about $800,000; it would cost more than twice that today. This dam has produced uninterruptedly an

[4] Stevens & Wood, Inc., New York, N. Y.

average of 35,000,000 kw-h. during this time. During the next 25 years that average is expected to be greatly increased, without one dollar of additional expense, through the agency of the climatic cycle.

A steam plant on the same system was built in the same year, 1912. This steam station has now been completely dismantled and replaced by a modern installation. The original investment is practically without returns. The power cost of the new installation will in the next 25 years rise with the price of coal and the increase in wages.

Nevertheless, as the author emphasizes, the best combination was reached by the use of both and has furnished uninterrupted power supply for 17 years.

A system fed by steam power alone cannot have a load factor greater than that of the daily load curve. Hence the increase of steam power while decreasing hydro will thwart the possibility of operating steam at higher load factors and eliminate the only condition under which steam can be more economical than hydro—namely, at high load factor. Without auxiliary hydro, steam power is doomed to operate at uneconomical load factors below 50 per cent, which is the usual case of the American power system.

While this applies to a large section of the country, comparative merits of steam and hydro are necessarily regional. The abundance of natural gas and crude oil in California greatly favors the use of steam power, while no steam power can possibly compete with the rates at which firm hydroelectric power is obtainable in Eastern Canada, namely, $15 per hp. per year or 2.3 mills per kw-hr., and where in consequence steam is made by hydroelectric power for industrial use.

Edwin H. Brown.[5] The author has marshaled his statements in a brief and concise style that is commendable, but tends to imply that the paper is a summary of accepted facts. It is true, as the author says, that generally a ten-year-old steam plant is obsolete, but in judging the plans for future power generation the reason for obsolescence should be thoroughly sought. Thus the 1920 steam plant is not obsolete, because coal costs more now than it then did, as the author's Fig. 2 shows that the cost of coal per ton is the same now as ten years ago. A new steam plant has been built because the load has increased and the old plant has become a standby. The new plant is more efficient because the steam engineer has always available an increased head. This could be best shown if the author had included in his Fig. 1 a head curve for the steam plant. Such a curve would have shown that in the year 1800 the steam plant rendered available to its prime mover a head of about 130,000 ft. and that in 1930 this head would be over 475,000 ft.

It would appear that for a system with an increasing load the comparison of steam- and hydro-plant installations should weigh carefully the investment of capital in a hydro plant against which only 2 per cent for depreciation and obsolescence is charged. The increase in steam pressure (or head) in a steam plant is accompanied by an increase in capacity for given physical dimensions of apparatus and may result in no increased cost for the gain in efficiency that is made.

The increasing load that has been mentioned as being primarily responsible for the new plant that renders the old one obsolete has also tended, in American practice, to make modernizing extensions to old steam plants obsolete. If the load increased at a less rate than it has done in the past, it is entirely conceivable that more work would have been done in enlarging and improving older plants in some of the ways that have been described in the technical press and in papers on the subject. For instance, in an older steam plant, the pressure could be increased and a turbine be put between the new boilers and the lower-pressure turbine. This would modernize and enlarge the plant, as steam-turbine efficiency has been high for over ten years, and no great loss would result from using the older turbine as the lower-pressure stages of the modernized plant. Such work, if done, would reduce the apparent obsolescence of the steam plant that is now ten years old. It has, in all probability, not been done extensively because the rate of increase of load, together with load factor, has made it undesirable.

It seems that the author could well have touched on the influence of increasing load on capital expenditures.

[5] Assistant Manager, Steam Turbine Department, Allis-Chalmers Manufacturing Co., Milwaukee, Wis. Mem. A.S.M.E.

Joel D. Justin.[6] The paper is of great interest at this time when there is so much in the public press about the utilization of our undeveloped water powers. The writer is particularly interested, as he has spent much of his time in the past several years in connection with problems involving the coordination of these two principal sources of power.

Steam and Hydro Complementary, Not Competitive

As a result of many investigations, he is convinced that in any section of the country where suitable hydro sites exist the load curve of a large power company can be more economically supplied by a suitable combination of hydro and steam plants than by steam plants alone. It is unfortunate that these two sources of power are usually considered as competitive. This is so only in a very limited sense. Their relationship can be better expressed by saying that they are complementary.

Steam Plants for Base Load, Hydro for Peak

Usually, steam plants should be installed primarily for the base load and hydro plants to take the peaks. During times of high water, the position is temporarily reversed, and the hydros supply the base and the steam the peak. This makes it highly desirable that the steam base load should be sufficiently large to permit full absorption of the available hydro energy at times of high water. The increment cost of energy for a modern steam plant is low, and in most cases new peak-load steam plants are seldom justifiable. This is because at small additional capital expenditure a highly efficient steam plant can be built and assigned to the base of the load, resulting in a high annual capacity factor. The old steam plants are then raised up on the load curve or held in reserve and generate a smaller number of kilowatt-hours per year. There are a few hydro plants, like Niagara, which are suitable for constant base-load operation, but such plants are rare. The usual hydro plant with pondage must be conceived of as primarily suited for taking care of the upper portion of the load curve, which may have a load factor of 10 to 40 per cent.

Basic Reason Why Hydros Are Suitable for Peaks

There is a basic economic reason why hydro plants with pondage are particularly suited to take care of the upper portion of the load curve which has a relatively low load factor. The greater portion of the investment of hydro plants is in items independent of the amount of the capacity installed, such as the dams and the lands required. Thus, at a given site, on a given river, it may cost $200 per kw. to provide an installation on a 60 per cent load-factor basis during the low flow week, but additional capacity may often be installed at a cost of $40 to $70 per kilowatt-hour. If the capacity which it is proposed to provide will do the same work on the load curve as alternative steam capacity, it merely remains to determine whether the proposed hydro installation can serve this portion of the load curve more cheaply than additional steam capacity.

[6] Hydro-Electric Engineer, United Gas Improvement Company, Philadelphia, Pa.

Extensive Economic Study Required to Determine Advisability of Hydro

Any load can successfully be served exclusively by steam capacity. Consequently, when new capacity is required, the obvious thing to do is to provide more steam capacity. If, however, it is proposed to build a hydro plant to serve a portion of the load, a rather extensive economic study is required to determine whether the proposed hydro project would be more economical than to provide additional steam.

Firm Capacity of Hydro Plant

For this reason it is desirable to set up the proposed hydro project on a steam alternative basis. The first question to determine is what portion of the hydro capacity that it is proposed to instal can be relied on to perform the same function on the load curve as alternative steam capacity. This portion of the installed capacity is usually referred to as firm capacity. It may be determined for any given year by a study of the system-load curves at times of maximum demand, pondage conditions, and available stream flow.

The accompanying diagram (Fig. 7), which is the load curve of a large power system for the maximum demand week of December, shows clearly what is meant by firm capacity of a hydro plant. The system for which the load curve is presented had a peak load in 1926 of 1,150,000 kw. In the system there are hydro plants, with ample pondage, having an installation of 250,000 kw. During a minimum-flow December week, there is available at these hydro plants 4,050,000 kw-hr. The shaded area shows the load that could have been carried by this hydro with the energy available. It indicates that the firm capacity of these plants in that week was 241,500 kw., which is within 8500 kw. of the installed capacity. It will be noted that although the hydro plant during such a week of maximum demand and minimum available energy would have been operating on a load factor of only 10 per cent, the hydro was supplying 21 per cent of the required capacity. An analysis of other weeks showed that the week of maximum demand provided (as is usually the case) the most severe criterion for determining the load which the hydro plants could be relied on to carry.

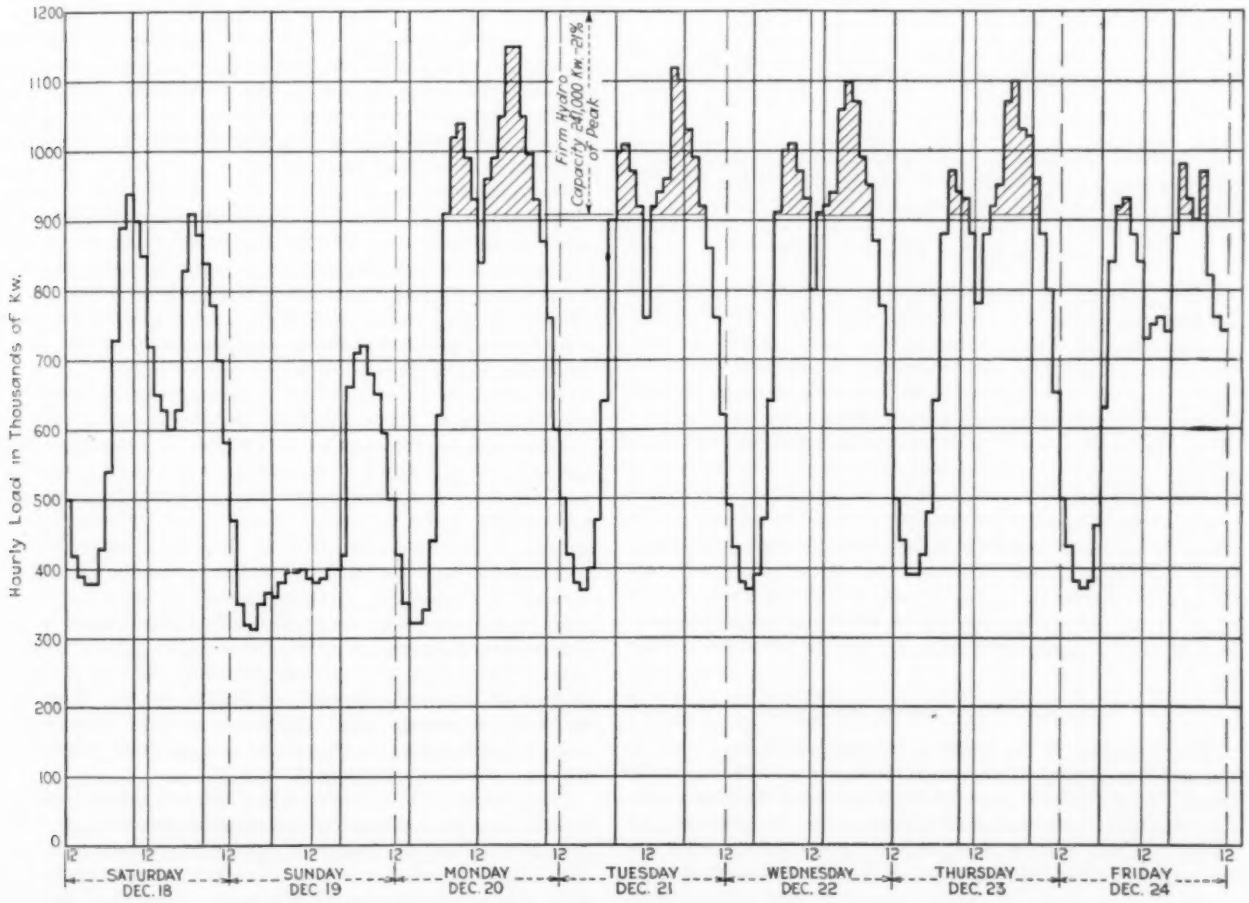


Fig. 7 Load Curve for the Peak Week of 1926 of a Large Power System
(Kw-hr. for week, 122,000,000; hydro kw-hr. available, 4,050,000.)

Steam Plant Costs as a Yardstick for Hydro

In order to measure the relative values of a hydro plant for serving any part of the load curve, it is necessary to use steam as a yardstick. Grossly erroneous conclusions may be drawn by using either average or increment costs. The total annual cost of a steam plant may be divided into the following factors:

(1) Fixed charges in dollars per kilowatt per year.

(2) Fixed component of operating in dollars per kilowatt per year. This is a fixed annual item for any given plant regardless of peak carried or energy turned out.

(3) Peak prepared-for cost, dollars per kilowatt per month. This cost varies with the peak prepared for and carried for each month.

(4) Increment cost of energy per kilowatt-hour. This is a constant unit cost regardless of quantity of energy.

For preliminary studies, items (1), (2), and (3) may be computed on an annual basis and called the annual unit capacity cost of steam. For a 400-lb. steam plant having a capital cost of $100 per kilowatt, fixed charges of 12 per cent, and a $4 coal cost, the annual unit capacity cost may be $16 per kilowatt per year, made up of $12 fixed charges and $4 for the sum of items (2) and (3), and increment cost of energy 2.4 mills per kilowatt-hour.

Thus, assuming, for simplicity, no transmission differential against hydro, if a 100,000-kw. hydro project is being considered with a firm capacity of 80,000 kw. and a utilizable energy production of 360,000,000 kw-hr. per average year, the comparative annual value would be:

(From firm capacity) 80,000 × $16	$1,280,000
(From energy) 360,000,000 kw-hr. at 2.4 mills	864,000
Hydro comparative annual value	$2,144,000

If the total annual cost of hydro were 10 per cent, this would justify a maximum investment of $21,440,000 for the 100,000-kw. hydro plant having a firm capacity of 80,000 kw.

Importance of Firm Capacity

From the tentative set-up given, the very great importance of the firm capacity of the hydro plant is evident. Thus, it is at once evident that if the determined firm capacity had been only 40,000 kw., the justifiable expenditure would have been much less. Firm capacity is dependent on stream flow, pondage, and the load curve. If a hydro plant has ample pondage, the firm capacity will increase with the growth of the system-load curve and gradually approach the installed capacity. Although this factor is of the utmost importance in connection with considering the economic advisability of hydro projects, it does not receive the attention which it requires, for certainly the hydro is only fully useful in so far as it can do the work which a steam plant might do. There are of course collateral advantages in having some hydro power in any power company system, which the author has mentioned, but it is often better not to attempt to give these a dollar value.

Proportion of Steam and Hydro

The proportion of steam and hydro capacity in any given system which is economically advisable is dependent on so many different factors that each particular case requires a thorough analysis. Based on a number of investigations, it may, however, be stated that in a number of systems in the eastern United States it is economically advantageous to serve from 16 to 40 per cent of the total load of the system from peak-load hydro plants, the remainder of the load being served by steam plants and base-load hydro plants like Niagara. The variation in the percentages is caused by such factors as shape of load curve, capital cost of hydro projects, stream flow available, cost of storage, availability and extent of pondage, additional transmission required for hydro, etc.

Depreciation and Obsolescence

One statement made by the author should most certainly be challenged. Not many engineers will agree with the statement that 10 per cent per year is a proper obsolescence and depreciation charge against steam plants. It is only relatively true to say that a steam plant ten years old is obsolete. Such a plant still has earning capacity, but its function is altered. Instead of operating on the base of the load, as it did when first built, it is now either held in reserve or operated over short-time peaks. In this function, however, it obviates the necessity of additional capital expenditures for another new plant at the rate of perhaps $100 per kilowatt and hence cannot be considered obsolete until such time as it would pay to scrap it. This is a much debated subject, but to the writer it does not seem logical to include obsolescence as such in economic set-ups of this nature. We really do not have to worry about obsolescence at all—it takes care of itself. This is for the reason that, if we are going to build a new plant to fully supersede an old plant, the net earnings for the new plant must be great enough to carry not only the investment in the new plant, but also the unamortized investment in the old plant. In other words, to build a new steam plant to supersede an old one, the annual fixed charges plus operation and maintenance costs on the new plant should be less than mere operation and maintenance costs on the old plant.

It is believed that 3.5 to 5 per cent is ample allowance for depreciation on steam plants. Depreciation on hydro plants will vary with the percentage of the total investment in dams and reservoirs from 0.5 per cent to 1.5 per cent. In general, there will be a differential in favor of hydro plants on the score of depreciation of from 2 to 3 per cent.

Nelson E. Funk.[7] The writer is a firm believer in the author's principal conclusion—namely, that where economically practicable hydro-plant sites are available, a system can generate power cheaper by combining steam and hydro plants than by utilizing either one to the exclusion of the other. The recent construction of the Conowingo Plant is sufficient evidence of the official policy of our company in the same direction.

However, it is felt that the author's comparison of steam and hydro stations, in many of its details, should not be allowed to stand without criticism as an example of economic study. Among the minor points of comparison which seem unsound is that regarding financing. Certainly, fundamental economics would dictate that if the overall kilowatt-hour cost was the same with either of two proposed systems, the one requiring the least investment of capital would be preferred. Likewise, the author's criticism of the reliability of steam plants is not well taken. He states that hydro plants are mechanically $2^1/_2$ to 6 times as reliable as steam plants, when in reality this is the ratio of the outage times. The true comparison of the reliability of steam and hydro plants is in the service demand availability factors, taken from the N.E.L.A. reports which the author quotes; these factors being for hydro units 98.40 per cent and for steam units 95.16 per cent. The comparison is obvious.

The portions of the author's paper which it is believed are most open to question are those in which the statements are made, without proof and as if they were generally accepted facts, that the present economic life of a steam turbine is approximately 4.6 years, that annual charges for retirement should be of the order of 10 per cent on steam-plant equipment, and that few steam plants 10 years old are not obsolete.

In the Richmond Station of the Philadelphia Electric Company, two turbines installed in 1925, and hence four years old at the present date, are the first-line units of the company's system. In Chester and Delaware Stations, which are of economy slightly less than that of Richmond Station, the units have been installed at various times from five to eleven years ago; while Schuylkill Station, with units installed from twelve to fifteen years ago, still remains in operation and has resisted every effort to justify its retirement. This record indicates the fallacy of the author's statement of the short economic life of steam turbines. As a matter of fact, the very rapid early development of steam-station efficiency to which the author refers as

[7] Vice-President in Charge of Engineering, Philadelphia Electric Company, Philadelphia, Pa. Mem. A.S.M.E.

proof of the short economic life of steam units is really evidence that the units installed within the last fifteen years are not likely to be retired from the standpoint of obsolescence alone for many years. Hence, the physical life of the units becomes the controlling factor in determining the suitable rate of annual retirement charge, and this indicates that 3.5 per cent is a much more logical rate than 10 per cent, although even the latter figure would be insufficient if the author's term of useful life were correct.

Regardless of the accounting methods which may be used in regard to the annual retirement charge, and whether or not the existing investment has been written off, the actual comparison which must be made when the retirement of existing capacity is being considered is that the existing capacity represents money which has already been invested and which must be thrown away if the equipment is retired, while the new investment represents additional money which must be found and upon which fixed charges must be paid.

The truly economic utilization of existing capacity is to retain it in service during its physical life, assigning the peak of the load to the plant of highest cost and the base to that of lowest cost, unless new capacity can be installed of a total cost (fixed plus operating) per kilowatt-hour less than the operating expense alone of the existing capacity.

This method of utilization results in pushing any given plant each year higher into the peak of the load so that its capacity is fully utilized with a smaller and smaller output of kilowatt-hours. Hence, the total cost of the plant (fixed plus operating) likewise decreases from year to year, and its effect upon the total cost of the growing system load becomes smaller and smaller. It will be found that this method of utilization of existing capacity with reasonable annual retirement charges will result in a materially lower cost of generation per kilowatt-hour than premature retirement of existing capacity, simply because new capacity can be installed to operate more cheaply.

Author's Closure

It is pleasing to note the amount and the competence of the discussion that has followed this paper. As indicated in the paper itself, it could only be an outline, as evidenced by the several thousand pages of current literature on the subject that were read during the time of its preparation. No paper of reasonable length could summarize all this without appearing to be a summary of accepted facts. It could only be a summary representing the author's interpretation of existing data in the light of his own experience. The author believes the criticism on this point to be well taken.

The influence of increasing load on capital expenditure has a marked effect on the cost of power, particularly on the factor of obsolescence and depreciation. It has had the effect of minimizing the importance of obsolescence or depreciation. This particular question of obsolescence seems to have drawn more discussion than any other. This is a factor that can be judged somewhat from experience, but at present can only be approximately determined. Certainly no percentage can be arrived at that will apply to all conditions. It is also evident that the percentage range is rather wide, as indicated by the discussion.

It does seem plain to the author, however, that the percentage usually used for obsolescence of steam plants is too low considering the rapid advance of the art and the vital effect of load factor on cost of steam power. Several outstanding steam engineers seem to recognize this. The author has at hand the statement of one of the steam-turbine manufacturers in this country stating that "the present economic life of a steam turbine is about 40,000 hours" (about 5.6 working years). Also he would quote from the paper, "Factors Affecting Power Cost," by C. F. Hirshfeld, appearing in *Mechanical Engineering* in the issue of November, 1929 (this was after the author's paper had been submitted, but before it was published):

> It is not so long ago that power-plant equipment was assumed to have a life of at least 20 years and power-plant buildings a life of 25 or 30 years. Such values are conservative when one considers only the operating possibility. However, when one considers the operating probability, far different figures frequently present themselves. It is not that the apparatus is worn out, but that it is less desirable and useful than before because of size, thermal economy, personnel requirements, location, flexibility, or what not. The graphs of Figs. 2 and 3, which give the histories of outputs of different units in two typical stations, show very well what happens when plants are built serially or units are installed serially. Each unit, in general, reaches a peak of production and then drops off in favor of more modern units. It is obvious that while 20 or more years of use may be assumed as feasible, about *five to seven years* of heavy-duty life represents a good average if these graphs are typical. And they are typical of a great many actual cases. It is such uncertainties as are indicated by these graphs that make many executives "play safe" when it comes to estimating probable useful as against usable life.

There are to be sure, as indicated by Mr. Nelson E. Funk, important instances where steam turbines operate against the base of the system load for longer periods. The addition of hydro power to a system on account of its allocation to the peak of the load has the effect of prolonging the economic life of existing steam turbine plants on that system.

It would not seem that the rapid development of fuel-burning plant efficiencies in the past would argue against the continuance of their rapid development in the future, inasmuch as the overall efficiency now is not much over 26 per cent.

The relative reliability, or, as the author has taken it, the relative *unreliability* of steam and hydro plants, affects the reserve capacity required. This reserve capacity is on account of generating units being off the line; so a comparison of outages was chosen as emphasizing more clearly the relative effect upon required reserve capacity than would operating percentages. In other words, the shortcomings of the two methods of generation were purposely chosen since attention to them represents progress. The discussion states, "Certainly fundamental economics would indicate that if the overall kilowatt-hour cost was the same with either of two proposed systems, the one requiring the least investment of capital would be preferred." This may be taken as a fundamental principle of economics, but in the particular case pointed out in the paper, making the larger investment does not in any way penalize the consumer or the public, but does have the effect of rewarding good management and enterprise by increasing this part of the rate base.

HYD-52-7a

The Flow of Fluids Through Orifices in Six-Inch Pipes

By S. R. BEITLER[1] and PAUL BUCHER,[2] COLUMBUS, OHIO

In this paper the authors tell of work done in the calibration of orifices and a nozzle in a six-inch pipe line using water and steam as the fluids to be metered. Two series of orifices were tested. One series included concentric orifices with from 15 per cent to 84 per cent diameter ratio. The second series included concentric orifices with from 64 per cent to 87 per cent diameter ratio. The nozzle tested was a one-flow type with 58 per cent diameter ratio.

WITH the improvement in efficiency of power plants and the necessity for a more accurate determination of the various losses, there has come a demand for increased accuracy in the meters used. A few years ago if a meter was accurate to about 3 per cent, it was considered very good, but today the users of meters are demanding accuracies of 1 per cent or less, and so an increased knowledge of the things affecting the accuracy of the meter is of great importance.

Most of the meters used for this type of work are what are known as the inferential-head type which depend upon measuring the rate of flow by determining the pressure drop across some "primary element" placed in the pipe, through which the fluid is flowing. The "primary elements" most commonly used are the venturi tube, flow nozzle, and thin-plate orifice. The theory upon which these primary elements operate should be familiar to all engineers and will not be discussed here.[3] The "coefficient of discharge" of these elements is of great importance, and so far as is known now can only be determined experimentally.

While a great deal of work has been done in determining the coefficients of discharge of the various primary elements for different fluids, there are very few published results of accurate determinations of these coefficients, especially for steam as the fluid is measured.

Because of this fact, it was suggested to the council of the Engineering Experiment Station of the Ohio State University that the equipment of the mechanical-engineering laboratory could well be used in attempting to determine some facts about the flow of steam, water, and possibly air through orifices and nozzles in pipe lines

Inasmuch as the Bailey Meter Company of Cleveland, Ohio, was intensely interested in this subject, the matter was taken up with its engineers, and they agreed to assist with the project. They have supplied all of the material, such as orifices, nozzles, special sections of pipe, and other apparatus; have granted money for the payment of observers; have given the benefit of their advice and experience whenever needed; and have loaned to the experiment station the services of R. L. Galley, one of their junior engineers, to assist in the work on the project. They have also supplied the data taken for the calibration of these orifices and nozzles with water, which had been previously determined in connection with their private research.

It is desired especially to express appreciation of the help and advice of R. E. Sprenkle and J. C. Albright, of the Bailey Meter Company, and of Professors F. W. Marquis and Horace Judd, of the Department of Mechanical Engineering of the Ohio State University, who assisted in solving some of the difficult problems which arose in the course of the work.

Scope of Investigation

This report covers the work done in the calibration of orifices, and a nozzle in a 6-in. pipe line using water and steam as the fluid

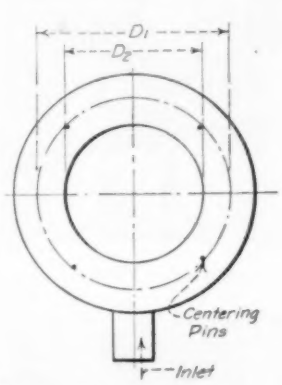


Fig. 1 Concentric Orifice

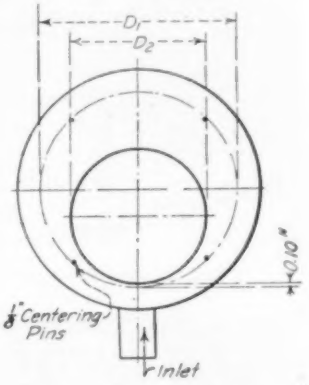


Fig. 2 Eccentric Orifice

to be metered. It is thought that the results which have been achieved so far warrant their publication although more data are continually being gathered on this subject, so that this may be considered as a progress report.

Two series of orifices have been tested, one a series of concentric orifices running from 15 per cent to 84 per cent diameter ratio, and another series of eccentric orifices varying from 64 per cent to 87 per cent diameter ratio, and a third set of one flow nozzle of 58 per cent diameter ratio. Most of the orifices were calibrated using both water and steam, but the nozzle was calibrated using steam only.

The original scope of the investigation called for the use of both water and steam with all orifices and with four flow nozzles, but it was found that the size of condensing and superheating equipment limited the capacity for steam so that it was not possible to calibrate the larger size orifices and flow nozzles using steam.

The continuation of the work is being carried on using a 3-in. line, which will make possible the use of a wide variety of orifices and nozzles, up to the largest practical diameter ratio.

The orifices and nozzle were calibrated using both water and steam. For steam the upstream pressure was 5 lb. per sq. in. gage pressure, 50 lb. per sq. in. gage pressure, and 100 lb. per sq. in. gage pressure, and from 80–245 deg. fahr. of superheat. The pipe-line velocities for water varied from 9.3 to 1042 ft. per min., and for steam at 65 lb. abs. from 346 to 7430 ft. per min. In all there were 101 tests using water, 26 tests using steam at 115 lb.

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[1] Assistant Professor of Mechanical Engineering, Ohio State University. Jun. Mem. A.S.M.E.,

[2] Associate Professor of Steam Engineering, Ohio State University, Assoc-Mem. A.S.M.E.

[3] For a complete discussion of this, see Part 1 of the report of the A.S.M.E. Special Research Committee on Fluid Meters, 1927.

Contributed by the Special Research Committee on Fluid Meters and presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of The American Society of Mechanical Engineers. For discussion and authors' closure, see pp. 123–135.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

abs. pressure, 149 tests at 65 lb. per sq. in. abs. pressure, 20 tests at 20 lb. per sq. in. abs. pressure on orifices, and 6 tests at 65 lb. per sq. in. abs. pressure using a flow nozzle, making a total of 302 tests. The results of some of these tests, however, are not considered as accurate for reasons which will be discussed later.

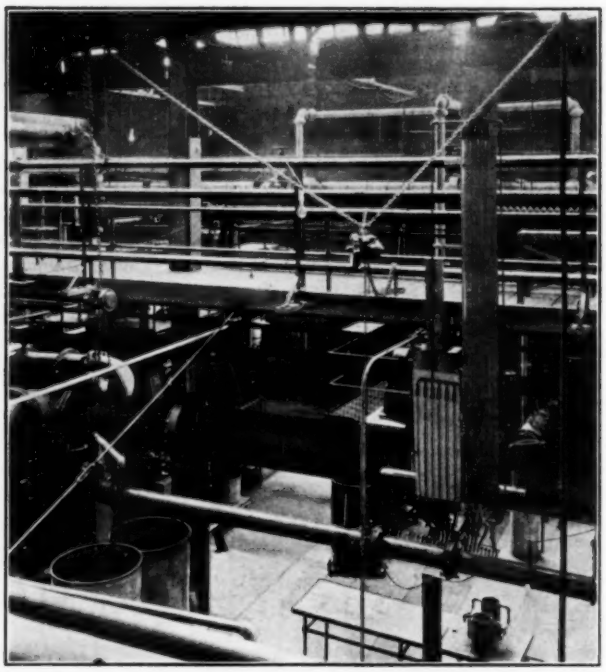


FIG. 3 PIPE LINE AND MANOMETER FOR WATER

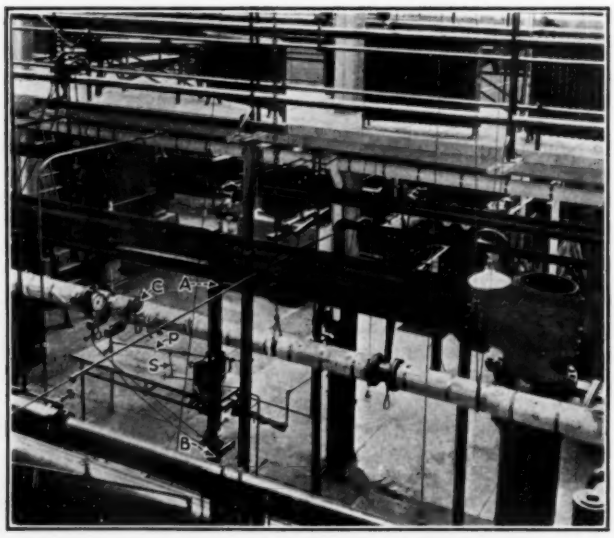


FIG. 4 PIPE LINE AND MANOMETER FOR STEAM

APPARATUS

The general shapes of the orifices used are shown in Figs. 1 and 2. These were made of Monel metal plate, and were carefully bored to the required diameter. Most of these plates were 0.039 in. thick except one plate of 0.124 in. thickness which was used for the smaller diameter ratio orifices for steam. This orifice was originally approximately 15 per cent diameter ratio, but was rebored as needed until the final ratio was 45 per cent. Also, one 50 per cent orifice for steam was cut from plate 0.061 in. thick. These thick plates were beveled on the downstream side to $^1/_{32}$ in. thickness at the orifice, which gave a square edge on the upstream side.

The orifice plates were located in the pipe by means of centering pins riveted to the plate and projecting from the upstream side of the plate. These pins were located so as to place the concentric orifices in the center of the upstream section of pipe, while the eccentric orifices were located with the edge of the orifice $^1/_{10}$

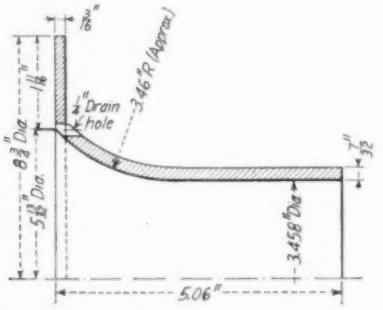


FIG. 5 HALF SECTION OF FLOW NOZZLE

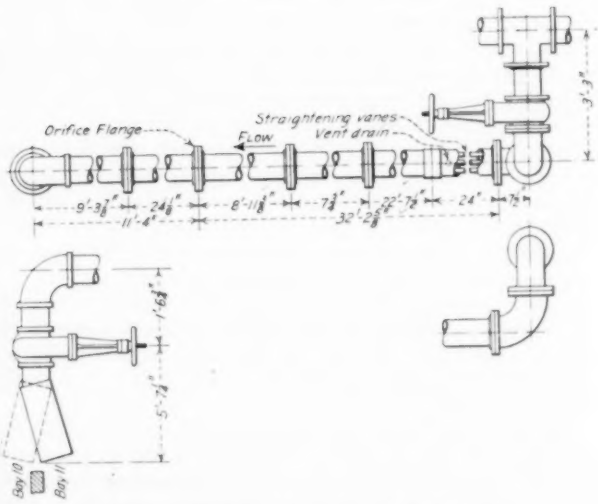


FIG. 6 LAYOUT OF TEST LINE FOR WATER

in. from the side of the pipe as shown in Fig. 2. The dimensions of all of the orifices used are given in Table 7.

Fig. 5 shows the flow nozzle tested. This is a standard flow nozzle as manufactured by the Bailey Meter Company. It was made of steel, cadmium plated, and was accurately bored to diameter before plating. This boring is done after the flanges have been turned up true, and the nozzle is reversed and mounted on a face plate. This produces a nozzle which is cylindrical on the outlet end with the flange straight and round.

The orifices and nozzles were mounted between the flanges of the two special sections of pipe shown in the illustrations Figs. 3, 4, and 6. The upstream section consisted of a piece of 6-in. standard-weight pipe 8 ft. $11^3/_8$ in. long. Pressure connections were made in this section at the points shown on Figs. 6, 7, and 8 by drilling and tapping for nipples of $^1/_8$-in. brass pipe. After the nipples were screwed into the pipe, the inside ends were carefully filed and scraped until they coincided with the inside wall of the pipe, and the edges of the holes were very slightly rounded.

The downstream section was made also of 6-in. standard pipe with pressure taps of the same type. The location of these is

also shown in Figs. 6, 7, and 8, and it will be seen that these connections were slightly staggered along the pipe. This was necessary in order to secure pressure connections at each $^1/_{10}$ diameter so that the location of the *vena contracta* could be accurately determined.

Vents and drains were provided as shown on Figs. 6 and 8. The vents were used only in starting, in order to get the air out of the line, while the drains were used to drain the pipe in order

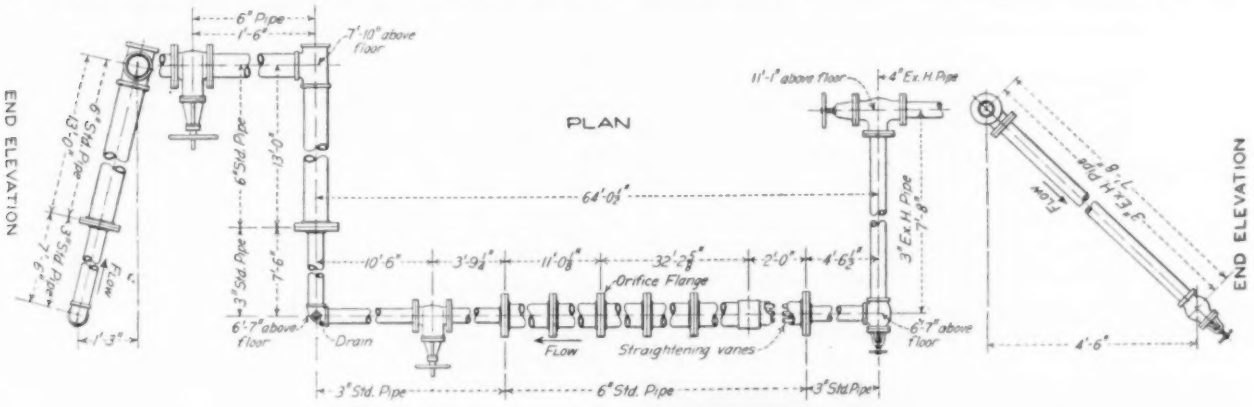


FIG. 7 LAYOUT OF TEST LINE FOR STEAM

to change orifices. A brass thermometer well was screwed into the pipe as shown. The temperature of the water was secured from an ordinary calibrated mercury thermometer placed in this well.

For steam the thermometer wells and drains were arranged as shown in Figs. 7 and 9. The thermometer wells were of brass of the finned wall type as recommended by the A.S.M.E. Code on

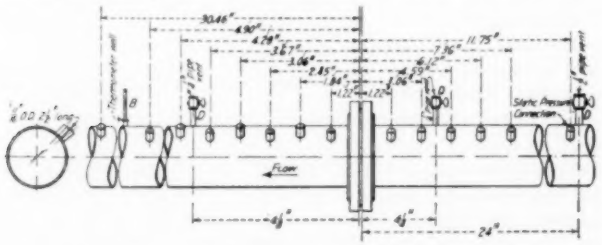


FIG. 8 DETAILS OF PIPE CONNECTIONS FOR WATER

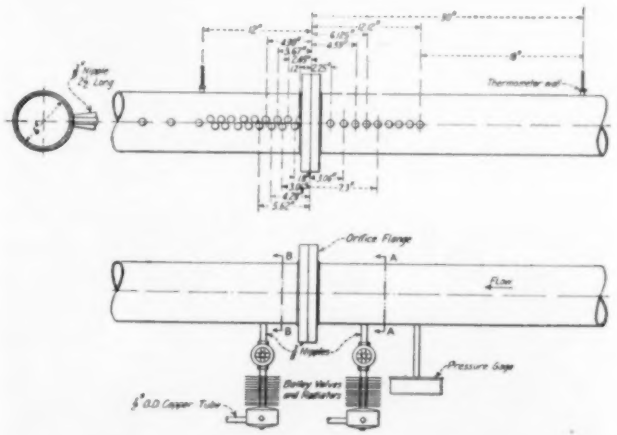


FIG. 9 DETAILS OF PIPE CONNECTIONS FOR STEAM

Instruments and Apparatus. While the thermometer placed at five diameters upstream was used for most of the runs, for some of the latter runs the one at 30 diameters was used, and for a few runs both of these were used in order to determine the temperature drop between the two positions. The drain was placed in the pipe to remove the condensation which collected in the pipe when warming up the line.

The location and general arrangement of these sections are shown in Figs. 6 and 8. It will be noted that for the 6-in. pipe, the pipe was used which gave the same arrangement for pressure connections for both steam and water. The upstream section was of such length as to give about 70 diameters of straight pipe upstream from the orifice, and the straightening vanes as shown on the drawings were put in to insure straight-line flow conditions. It was felt that this arrangement would give as nearly as possible parallel and smooth flow at the orifice section.

For water the inlet was connected to the laboratory service line by means of the valves and fittings as shown in Fig. 6. Water for calibration purposes was supplied from a 6-in. centrifugal pump having a capacity of about 1250 gal. per min., and from a 6-in. duplex pump of about 1500 gal. per min. capacity. The duplex pump was never used alone, but was used at times when the amount of water needed was beyond the capacity of the centrifugal pump. The centrifugal pump was steam-turbine driven, and was operated on a speed governor which kept a relatively constant pressure for any given capacity, while the duplex pump was operated on a pressure governor which gave a constant pressure condition so that for any one test run, the flow was fairly constant. The quantity of water used was controlled by the pump discharge valves.

When steam was used the inlet of the line was connected to the laboratory superheated steam header. The steam was generated in the University power house, and was conveyed to the laboratory through pipes located in tunnels. The steam conditions at the power house are about 185 lb. gage pressure and 130 deg. of superheat, but the pressure is reduced before it reaches the laboratory to about 140 lb. gage, while the temperature or quality of steam varies a great deal depending upon the quantity of steam being used. The steam was piped through a separate gas-fired superheater, and then into the superheat header. With this arrangement it was possible to supply steam at a fairly constant pressure and temperature to the orifice line for any particular run, so that conditions would not change appreciably during any run. The quantity of steam flowing was controlled by the 3-in. angle valve shown in Fig. 7.

The entire pipe line, except for a few fittings, and the section just before entering the condenser, was lagged by covering with 85 per cent magnesia pipe covering of standard thickness.

The quantity of water was measured either by weighing or by volumetric measurement. The arrangement for weighing the water is shown in Fig. 3. A piece of 6-in. pipe about 8 ft. 9 in. long was attached to the downstream orifice section, and a 6-in. elbow was placed on the end of this with the outlet turned up. The outlet of the elbow was bushed down to 2 in. size and a 2-in. valve connected to the bushing by a nipple. This valve also served to control the pressure in the pipe. The piping from the valve was arranged as shown in Fig. 3 so that it was possible to shift the discharge into either of the two weighing tanks shown

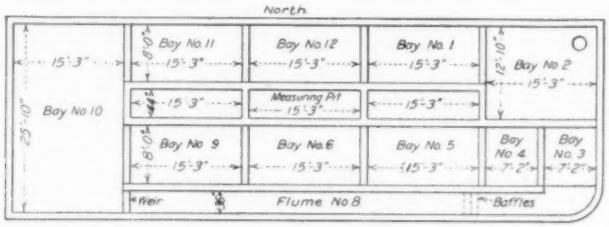


FIG. 10 ARRANGEMENT OF CONCRETE TANKS BELOW THE FLOOR OF THE LABORATORY

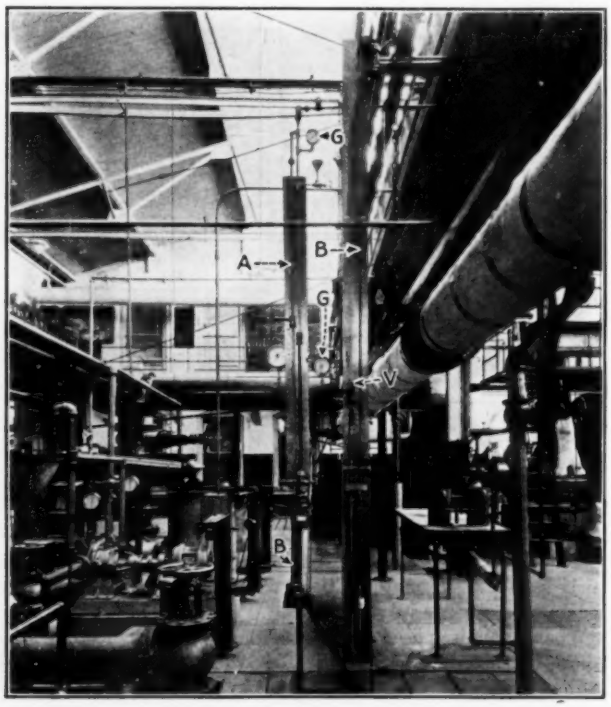


FIG. 11 MANOMETERS USED FOR DIFFERENTIAL AND STATIC PRESSURES OF STEAM

below this pipe. These tanks were of about 1200 lb. capacity and were set on platform scales. They were fitted with $2^1/_2$-in. quick-opening valves. This arrangement was used for orifices with diameter ratios from 15 to 35 per cent.

The volumetric method was used to determine the quantities passed through the larger orifices. For measurement, one of the cisterns or bays of the laboratory was used. These are concrete tanks below the floor of the laboratory arranged as shown in Fig. 10. For this work, bay No. 11 was used as a measuring bay and bay No. 2 was used as a suction bay for the pump. These bays are connected to $^3/_4$-in. gage glasses in the measuring pit in the center of the laboratory by 2-in. pipes, so that the level of the water in the bay can be read on a scale placed behind the gage glass. This scale is made of wood with a cross-section of about 2 in. × 4 in., and is kept well painted to prevent moisture from changing the length of the scale. The scale itself, reading in feet, tenths, and hundredths, is laid off directly on the white painted surface. The volume present in the bay for any gage reading is determined by calibration, and is read from a calibration chart.

When measuring larger quantities of water, the pipe was arranged as shown in Fig. 6 with a deflector pipe below the 6-in. gage valve. The deflector was arranged so that it could discharge the water either to the measuring bay or into a waste bay which was connected to the suction bay. By shifting this pipe quickly, it was possible to control accurately the lengths of test runs. The pressure in the pipe was regulated by means of the 6-in. valve.

The steam discharged by the orifices and nozzles was conveyed to a surface condenser through the piping indicated in Fig. 4. A valve was properly placed to control the pressure in the line. The condenser used was an Admiralty type of surface condenser with a cooling surface of approximately 465 sq. ft. It is fitted with a wet air pump and a circulating pump of the reciprocating type driven by a central steam cylinder. The cooling water was obtained from an outdoor reservoir, and was returned to the reservoir for cooling. The discharge of the condensate pump was piped to two tanks placed on calibrated scales of 1200 lb. capacity through a three-way cock and weighed. The cock was arranged so that the stream of condensate could be shifted from one tank to another quickly.



FIG. 12 DIFFERENTIAL HEAD MANOMETERS FOR STEAM

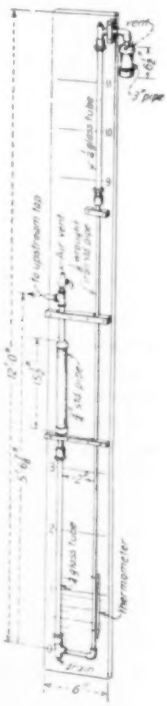


FIG. 13 STATIC PRESSURE MANOMETER FOR STEAM

The various pressure measuring devices used during the water tests may be seen in Fig. 3. The differential head across the orifice was measured by the multiple, inverted manometer shown at the right side of the illustration. This manometer consisted of four 10-ft. lengths of $^5/_8$-in. glass tubing connected to the nipples above the orifice, and six $2^1/_2$-ft. lengths connected below the orifice. The tops of these tubes were manifolded, and the manifolds were connected together by $^3/_8$-in. pipe. A $^1/_4$-in. air line was attached to the manifold at the top of the 10-ft. tubes for supplying air under sufficient pressure to balance the water pressure in the pipe. The valves in this air line were located at a convenient height, and were arranged so that the air pressure could be varied as desired, and thus the level of the water in the two sets of tubes could be set at any convenient position.

The glass tubes were connected to the nipples on the pipe by means of pieces of rubber tubing or garden hose, and pinch cocks were provided on the hose for throttling. The pipe was turned

so that the nipples were at an angle of about 45 deg. from the horizontal, and the tubes were placed above these so that the entire system was self-venting.

Thermometers were placed in the tubes as shown in Fig. 3. These were arranged at heights of 2 ft., 4 ft., and 9 ft. in the upstream tubes, and at about zero in the downstream tubes. These were used to determine the density of the water in the tubes.

Scales were laid off on the boards behind the tubes as shown. These were graduated in feet and hundredths, and the zeros were carefully leveled. The scale behind the upstream tubes was laid off directly on the board, and the board was then coated with shellac to prevent any appreciable change in length due to atmospheric conditions. The downstream scale was made of paper securely fastened to the boards. While this scale was subject to shrinkage, the customary method of operation was to adjust the

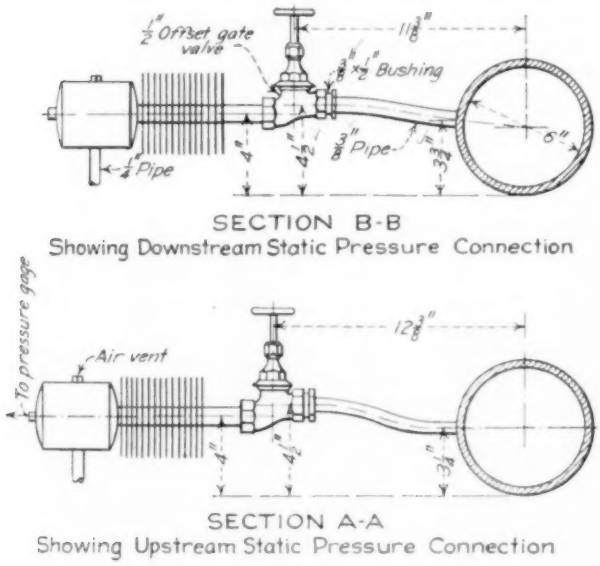


FIG. 14 DETAILS OF RADIATOR CONNECTIONS

height of the water in the tubes so that the downstream reading was near zero so that any shrinkage was negligible. With this arrangement of manometer, it was possible to get four upstream readings and six downstream readings simultaneously, and so determine the variation of pressure along the pipe both above and below the orifice.

The mercury manometer shown below the pipe in Fig. 3 was used to determine the difference in pressure between two diameters upstream and five diameters downstream, and so determine the pressure loss. Arrangements were also made to vent air from the lines leading to this manometer. The gage shown just below the manometer tubes was used for indicating the static pressure in the pipe.

Figs. 4, 11, 12, and 13 show the arrangement of the pressure-measuring devices for steam. For measuring the differential head across the orifice the two manometers "A" and "B" (Figs. 4, 11, and 12) were used. The manometer "A" was used when the differentials to be measured were less than 5 ft. of water, while manometer "B" was used for heads greater than 5 ft. "A" was a 5-ft. inverted high-pressure water manometer. It was made from gage glass and piping as shown in Fig. 12. The top of the manometer was connected to a high-pressure air line through the two tubes shown so that it was possible to adjust the pressure in the manometer approximately to the same pressure as that within the pipe before the manometer valves were opened. The gage "G" (Fig. 11) was used to indicate the pressure in the top of the manometer. The manometer scale was laid off upon the oak supporting board, and the board was then shellacked to prevent any change due to atmospheric conditions. The bottom of the manometer was connected to the pressure connections in the pipe through $^1/_2$-in. copper tubing and radiators which will be described later. The surface of the manometer was covered with non-shatterable glass during the tests so that the observers would be protected in case of a tube failure. The manometer "B" was connected in parallel with "A" so that it could be used when the differential head was too high to be recorded on the water manometer. It was a single-leg, high-pressure mercury manometer with a scale calibrated to read in inches and tenths. With this arrangement it was possible to get reasonable accuracy of readings for heads from $^1/_2$ ft. up to about 13 ft. of water.

Figs. 7 and 14 show the arrangement of the pressure connections at the pipe. The manometer legs were connected to the radiators by $^1/_2$-in. copper tubing and compression fittings, and a $^1/_8$-in. globe valve was mounted directly below the downstream radiator. A tee was placed between the upstream radiator and the valve for a connection to the static pressure manometer. The radiators were the standard high-pressure radiator condensers used by the Bailey Meter Company. They are built up of welded steel, and are arranged as shown in Fig. 14. The reservoir serves to keep a sufficient quantity of water to fill the tubes during any slight change of head, while the fins on the steel tube serve to condense enough steam to keep the reservoirs filled with water. A vent is provided in the top of each reservoir to permit the venting of air. These vents were operated manually while warming up the apparatus. A special offset gate valve was placed at the end of the reservoir, and this was connected to the main line through a $^3/_8$-in. extra-heavy nipple screwed into the pipe. These nipples were bent so that the upstream and downstream valves and reservoirs

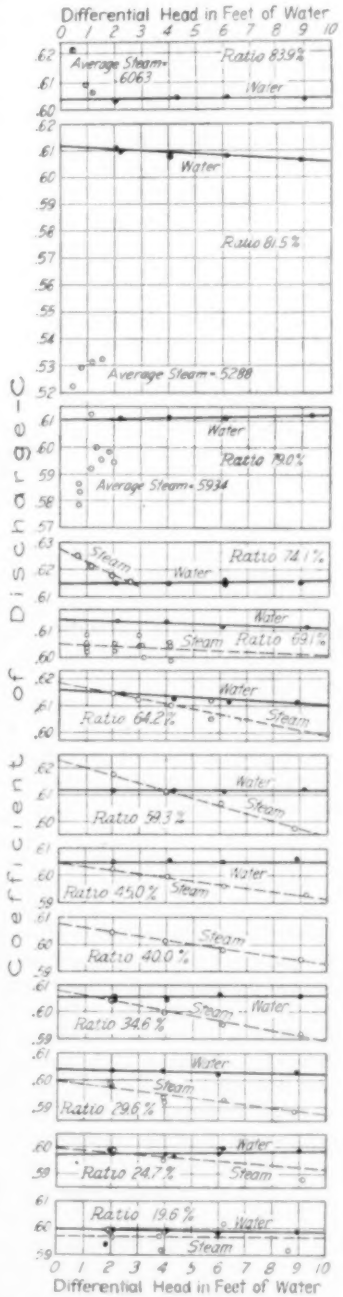


FIG. 15 VARIATION OF THE COEFFICIENT OF DISCHARGE WITH DIFFERENTIAL HEAD FOR CONCENTRIC ORIFICES

were level. This assured that as nearly as could be determined, the water head from the pipe to the manometer was the same on both upstream and downstream orifice connections. The ends of the nipples were carefully filed to give a smooth surface inside the pipe.

A plug was taken from the end of the upstream-reservoir, and a gage was connected to the reservoir to read the static pressure at the upstream connection. This gage was used for adjustment only, while the pressure during the test runs was measured by means of the mercury manometer shown in Fig. 13 and at "B" in Fig. 11. This was a two-leg mercury manometer with a range 6 ft. to 11.4 ft. of mercury corresponding to pressures 37 lb. per sq. in. to 59 lb. per sq. in. This manometer was made up of two sections of $^1/_4$-in. glass tubing mounted as shown in Fig. 13, and connected by $^1/_8$-in. pipe and fittings. A mercury reservoir made of $^1/_2$-in. pipe was mounted on the high-pressure side to prevent mercury from running into the pipe, and a trap was located at the top of the atmospheric leg to prevent any high pressure from blowing the mercury out. The piping and tube packing glands were mounted on a 1 in. × 6 in. board, and the manometer scale was laid out directly on this board. The board was then well coated with shellac. The top of the mercury reservoir was connected to the upstream radiator by $^1/_4$-in. pipe as shown at "P," Fig. 4. It was not possible to grade this line down so as to make it self-venting, and so it was graded up and a vent was placed at "V," Fig. 11, to remove any air which might gather in this line.

For measuring the lower static pressures for runs at 20 lb. absolute pressure, a tee was placed in the line between the radiator and the high-pressure manometer. This tee was connected through a valve to a standard 30-in. Merriam U-tube mercury manometer. The connection for this manometer is shown at "S," Fig. 4.

For measuring the pressure during the 115 lb. per sq. in. runs the pressure gage shown at "C," Figs. 4 and 13, was used. This was a 160-lb. Bourdon tube gage of the single-tube type connected to the pipe at two diameters upstream through a gooseneck connection.

Method of Taking Data

The method of taking the data was quite similar in all cases. For the volumetric tests with water, it was as follows: With the orifices or nozzle properly centered and bolted between flanges, the pumps were started and flow established at a rate to give the desired pressure drop across the metering element. This water was passed through the deflector pipe to the suction bay until a steady flow had been established, and all of the air eliminated from the apparatus. The level of water in the measuring bay was then read. After all of the conditions were satisfactory, at a given signal the deflector pipe was snapped to a position discharging into the measuring bay, and simultaneously two stop watches were started. Readings of head and temperatures were taken at the desired intervals throughout the run. The head was read by two observers reading simultaneously. The test run was ended by deflecting the stream of water again into the suction bay and taking the time by the stop watches. After a sufficient length of time had elapsed for surging to cease in the measuring bay, a final reading of elevation of water in the bay was taken. All readings of the bay elevation were checked by two observers. The length of the test was controlled by several factors. First, in order to get accuracy, the minimum quantity of water in the bay was limited to a depth of 4 ft. If the level of water in the bay increased more than 4 ft. in a half hour, the run was limited to 30 min. in duration. Otherwise the test ended when the measuring bay was completely filled. Observations were taken so that the minimum number of head readings was 20 while the maximum time between readings was 2 min. Temperature readings and static-pressure readings were ordinarily taken at the beginning and ending of each run.

For the weighed water test, the procedure in testing was similar to the displacement tests except that the water was run into one of the weighing tanks. At the beginning of a run the discharge was shifted into the other tank, and the two tanks were alternately filled, weighed, and emptied in the usual manner throughout the duration of the test.

When steam was used the condenser and superheater were usually started, and steam was turned into the line three or four hours before the beginning of a test. When the line was thoroughly warmed, the flow was adjusted to give approximately the desired pressure drop across the metering element, and the pressure in the pipe was adjusted to the approximate static pressure desired, and the apparatus was allowed to run at this rate for about 2 hr. so that all temperatures were brought up to a point where they would remain practically constant. While this was going on the discharge from the condensate pump was allowed to

TABLE 1 COEFFICIENTS FOR CONCENTRIC ORIFICES USING WATER

Test No.	Ratio $\frac{D_2}{D_1}$	H head corrected for temp.	Q = cu. ft. per sec.	K = coeff. discharge $= \frac{Q}{C\sqrt{H}}$	Location of vena cont., pipe diams. downstream
1057	0.1477	2.137	0.031625	0.6042	0.8
1058	0.1477	4.392	0.045260	0.6032	0.8
1059	0.1477	5.833	0.052180	0.6035	0.8
1060	0.1477	8.928	0.064535	0.6033	0.8
1042	0.2468	9.008	0.17992	0.5990	0.8
1043	0.2468	5.992	0.14650	0.5979	0.8
1044	0.2468	4.316	0.12408	0.5968	0.8
1045	0.2468	2.061	0.08596	0.5980	0.8
1061	0.2468	1.961	0.083926	0.5989	0.8
1062	0.2468	4.016	0.119905	0.5978	0.8
1063	0.2468	6.155	0.14902	0.6001	0.8
1038	0.3455	4.067	0.24033	0.6046	0.8
1039	0.3455	6.062	0.29433	0.6063	0.8
1040	0.3455	9.024	0.35892	0.6059	0.8
1041	0.3455	2.108	0.17346	0.6058	0.8
1046	0.3455	2.155	0.17525	0.6056	0.8
1064	0.3455	2.093	0.17295	0.6063	0.8
1065	0.3455	2.070	0.17150	0.6045	0.8
1033	0.4448	2.026	0.28495	0.6051	0.8
1034	0.4448	4.129	0.40733	0.6057	0.8
1035	0.4448	6.105	0.49472	0.6050	0.8
1036	0.4448	8.893	0.59822	0.6062	0.8
1029	0.4938	2.144	0.36653	0.6074	0.8
1030	0.4938	4.088	0.50633	0.6075	0.8
1031	0.4938	5.985	0.61258	0.6077	0.8
1032	0.4938	8.966	0.75022	0.6079	0.8
1025	0.5926	2.045	0.5377	0.6115	0.7
1026	0.5926	4.295	0.7795	0.6118	0.7
1027	0.5926	6.187	0.9341	0.6111	0.7
1028	0.5926	9.162	1.1393	0.6121	0.7
1021	0.6421	2.365	0.70122	0.6145	0.6
1022	0.6421	4.262	0.93811	0.6126	0.6
1023	0.6421	6.338	1.1424	0.6115	0.6
1024	0.6421	8.862	1.3499	0.6112	0.6
1017	0.6913	2.185	0.8096	0.6140	0.5
1018	0.6913	3.980	1.0914	0.6133	0.5
1019	0.6913	6.077	1.3447	0.6115	0.5
1020	0.6913	9.218	1.6557	0.6114	0.5
1047	0.7407	2.099	0.95833	0.6148	0.4
1048	0.7407	4.084	1.3369	0.6149	0.5
1049	0.7407	6.188	1.6445	0.6146	0.5
1050	0.7407	9.010	1.9874	0.6154	0.5
1053	0.7900	2.226	1.1936	0.6109	0.4
1054	0.7900	4.066	1.6132	0.6110	0.4
1055	0.7900	6.163	1.9865	0.6111	0.4
1056	0.7900	9.359	2.4513	0.6119	0.4
1005	0.8146	2.213	1.3195	0.6098	0.3
1006	0.8146	4.074	1.7878	0.6092	0.3
1007	0.8146	6.114	2.1852	0.6079	0.3
1008	0.8146	8.908	2.6336	0.6069	0.3
1051	0.8146	2.044	1.2712	0.6113	0.3
1052	0.8146	4.020	1.7728	0.6080	0.3
1001	0.8391	4.283	2.0354	0.6046	0.3
1002	0.8391	2.008	1.3907	0.6035	0.2
1003	0.8391	6.127	2.4326	0.6044	0.2
1004	0.8391	9.024	2.9483	0.6035	0.2
1066	0.2957	2.020	0.12344	0.6037	0.8
1067	0.2957	3.945	0.17243	0.6034	0.8
1068	0.2957	6.009	0.21233	0.6021	0.8
1069	0.2957	8.948	0.25957	0.6029	0.8
1070	0.1974	2.107	0.055626	0.5994	0.8
1071	0.1974	4.083	0.077411	0.5993	0.8
1072	0.1974	5.997	0.093634	0.5982	0.8
1073	0.1974	8.937	0.11436	0.5987	0.8

TABLE 2 COEFFICIENTS FOR ECCENTRIC ORIFICES USING WATER

Test No.	Ratio $\frac{D_2}{D_1}$	H head corrected for temp.	Q = cu. ft. per sec.	K = coeff. discharge $= \frac{Q}{C\sqrt{H}}$	Location of vena cont., pipe diams. downstream
1321	0.6420	4.140	0.93733	0.6211	0.8
1322	0.6420	2.263	0.69100	0.6195	0.8
1323	0.6420	6.110	1.14011	0.6219	0.8
1324	0.6420	9.231	1.4036	0.6230	0.8
1317	0.6913	2.115	0.79956	0.6165	0.7
1318	0.6913	4.275	1.1392	0.6179	0.7
1319	0.6913	6.296	1.3828	0.6179	0.7
1320	0.6913	9.198	1.6715	0.6178	0.7
1313	0.7402	2.212	0.98067	0.6142	0.6
1314	0.7402	4.083	1.3354	0.6154	0.6
1315	0.7402	6.207	1.6483	0.6163	0.6
1316	0.7402	9.087	1.9933	0.6160	0.6
1309	0.7903	2.059	1.1633	0.6183	0.5
1310	0.7903	4.224	1.6675	0.6189	0.5
1311	0.7903	6.295	2.0369	0.6192	0.5
1312	0.7903	9.324	2.4750	0.6183	0.5
1325	0.7903	2.085	1.1694	0.6176	0.5
1326	0.7903	4.124	1.6467	0.6183	0.5
1305	0.8392	2.340	1.5352	0.6171	0.4
1306	0.8392	4.238	2.0673	0.6174	0.4
1307	0.8392	5.960	2.4540	0.6182	0.4
1308	0.8392	9.010	3.0203	0.6156	0.4
1301	0.8690	2.198	1.7360	0.6198	0.4
1302	0.8690	4.127	2.3818	0.6206	0.4
1303	0.8690	6.161	2.9064	0.6200	0.4
1304	0.8690	9.189	3.5513	0.6204	0.4
1327	0.8690	2.115	1.6977	0.6182	0.4
1328	0.8690	4.019	2.3469	0.6198	0.4

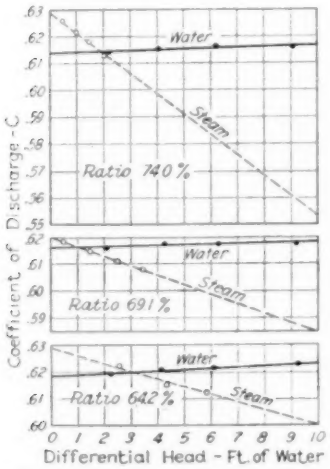


FIG. 16 VARIATION OF THE COEFFICIENT OF DISCHARGE WITH DIFFERENTIAL HEAD FOR ECCENTRIC ORIFICES

flow into the waste water trench; and while the line was warming, care was taken to eliminate all air and steam pockets from the manometer connections, and the water in the manometer tubes was allowed to cool approximately to room temperature. When the temperatures were constant, the test was started by throwing the three-way cock to discharge the condensate into one of the weighing tanks, and the time of start was noted on a reliable chronometer. Readings of head, temperatures, and pressures were taken at the desired intervals throughout the run. The differential head readings were made by two observers reading simultaneously when the water manometer was used, and by one observer when the mercury manometer was used. For runs at the 65-lb. and the 20-lb. pressures the mercury manometers were used, and were also read simultaneously by two observers. The test was ended at a certain time by shifting the condensate discharge into the waste trench again. During the runs the cans were alternately filled, weighed, and emptied as with the weighed water runs. Each run was made of sufficient length to get at least three tanks of water, the minimum amount in a tank being about 225 lb. The maximum length of run was 90 min. and the minimum (for 3000 lb. of water) was 15 min. Readings were taken of all pressures and temperatures at such intervals that the minimum number of readings was 20, and the maximum time between readings was 2 min.

CALIBRATION OF INSTRUMENTS

All instruments used were calibrated carefully.[3] The platform scales were calibrated before and after using over the entire range by means of standard weights. These weights were then checked by the Ohio Sealer of Weights and Measures, and found to be accurate within $^3/_4$ oz. in 100 lb., and the errors were found to be practically compensating.

The scales could be read to about $^1/_4$ lb., and were found to be so accurate that no correction was necessary. Care was taken at all times to keep the knife edges clean, and dirt and water off all working parts of the scales.

The measuring bay was calibrated by filling it with water. The water was weighed as it was put in, the height of the water in the bay gage glass noted, and the temperature of the water taken. The weight of water was found by using tanks on calibrated platform scales. A reading of the height of water in the bay was taken for about every 2000 lb. of water. A curve was then plotted showing the amount of water present in the bay for any gage height.

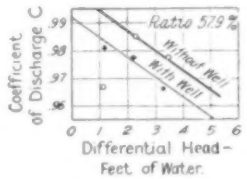


FIG. 17 VARIATION OF THE COEFFICIENT OF DISCHARGE WITH DIFFERENTIAL HEAD FOR FLOW NOZZLE

All thermometers were calibrated, the low-reading ones against a standard thermometer, and the high-reading ones by being placed in a well similar to the one in which they were used, and checked against steam temperatures.

The gages used were checked against a dead-weight gage tester. The gage tester was checked by carefully measuring the plunger diameter, and weighing the weights on a chemical balance.

All manometer scales were checked periodically against a steel tape or a steel scale while the zeros of the inverted water manometer were checked with a level at the same time. All manometers were checked for perpendicularity at frequent intervals. The mercury manometer was calibrated by comparison with a water column. The stop watches used were checked at intervals against a reliable chronometer which in turn was checked against Western Union time at intervals.

The orifice and nozzle diameters were measured carefully with inside micrometers. The diameters were measured at four places, and the averages considered as the average diameter of the opening. The nozzle diameter was measured at the outlet and also at the cylindrical portion. The diameter of the pipe was measured at the upstream pressure tap, and at a point about one-half pipe diameter downstream.

DISCUSSION OF RESULTS

Throughout these tests a serious attempt was made to install the orifices and nozzle in such a manner as to make the coefficients directly comparable to those which could be applied to actual commercial installations. The primary elements were all installed in accordance with the printed instructions of the meter manufacturer, and standard sized pressure connections were also used.

The results and some of the observed data obtained are shown in Tables 1, 2, 3, 4, and 5, and the results are also shown graphically in Figs. 15 to 21, inclusive.

Some idea of the accuracy of the testing methods used, and also some idea of the accuracy with which orifices can be used as metering elements can be gained by reference to Table 6. The first two tests of this table show how closely the value of the coefficient of discharge could be checked with water after the orifice had been removed from the pipe and reinserted at a later time. The last eight tests of this table show the same effect when steam was being used as the testing fluid. Tests for comparison have been chosen where the differential head was approximately the same since a change of head might cause a slight change of coefficient.

The greatest difference, about one-half of one per cent, shown

TABLE 3 COEFFICIENTS FOR CONCENTRIC ORIFICES USING STEAM

Run No.	Orifice diameter ratio D_2/D_1	Steam pressure, lb. per sq. in. abs. 1 P.D. upstream	Superheat deg. fahr. 5 diam. upstream	Diff. head at orifice ft. H_2O	Weight of steam, lb. per sec.	Quantity of steam, cu. ft. per sec.	C coeff. of discharge	Location of down-stream tap pipe diam.	Run No.	Orifice diameter ratio D_2/D_1	Steam pressure, lb. per sq. in. abs. 1 P.D. upstream	Superheat deg. fahr. 5 diam. upstream	Diff. head at orifice ft. H_2O	Weight of steam, lb. per sec.	Quantity of steam, cu. ft. per sec.	C coeff. of discharge	Location of down-stream tap pipe diam.
S-129	0.19588	59.977	137.25	8.6191	0.29273	2.5392	0.5918	0.8	*S-164	0.49984	63.113	156.11	2.0139	0.98722	8.3494	0.6063	0.8
S-130	0.19588	64.644	138.74	6.2292	0.26213	2.1233	0.6019	0.8	*S-165	0.49984	114.698	124.62	5.4871	2.17750	10.1108	0.6005	0.8
S-131	0.19588	62.371	135.72	3.8225	0.20069	1.6753	0.5969	0.8	*S-166	0.49984	115.408	123.47	3.9486	1.86375	8.591142	0.6035	0.8
S-132	0.19588	62.097	133.71	1.9107	0.14231	1.1900	0.5991	0.8	*S-167	0.49984	115.101	126.53	2.5666	1.50153	6.966198	0.6050	0.8
S-141	0.19588	62.921	100.32	3.9001	0.20625	1.6320	0.5918	0.8	*S-168	0.49984	114.561	129.85	1.0217	0.95292	4.45967	0.6113	0.8
S-142	0.19588	64.537	99.78	2.0630	0.15306	1.18139	0.5964	0.8	*S-179	0.49984	114.559	75.83	5.8566	2.31278	10.0856	0.5981	0.8
*S-143	0.19588	63.692	117.05	4.0217	0.21097	1.66693	0.5989	0.8	*S-180	0.49984	116.221	85.52	4.0166	1.93368	8.4409	0.6043	0.8
*S-144	0.19588	63.868	103.03	2.0010	0.15037	1.17704	0.5987	0.8	*S-181	0.49984	113.337	93.56	2.4795	1.49253	6.7415	0.6039	0.8
S-145	0.24702	63.443	136.40	9.1570	0.49076	4.03447	0.5883	0.8	*S-182	0.49984	115.138	90.02	0.9943	0.97153	4.30398	0.6148	0.8
S-146	0.24702	64.139	146.74	6.0392	0.40549	3.34149	0.5992	0.8	S-79	0.5926	62.422	167.58	8.7969	2.92028	25.2922	0.5976	0.7
S-147	0.24702	64.753	122.93	3.9614	0.33282	2.63893	0.5955	0.8	S-80	0.5926	63.537	174.44	6.0533	2.4706	21.2124	0.6069	0.7
S-148	0.24702	64.078	138.65	2.0978	0.24005	1.96048	0.5999	0.8	S-81	0.5926	62.069	176.00	3.9839	1.9986	17.5697	0.6124	0.7
*S-173	0.2957	64.313	112.64	8.8339	0.70694	5.56807	0.5882	0.8	S-82	0.5926	61.945	171.97	2.0157	1.4386	12.6126	0.6189	0.7
*S-174	0.2957	65.477	118.83	6.2116	0.60000	4.68348	0.5927	0.8	S-83	0.6421	62.612	172.87	5.6522	2.8875	25.0898	0.6118	0.6
*S-175	0.2957	64.334	137.26	3.9522	0.46994	3.81709	0.5937	0.8	S-84	0.6421	62.378	184.85	4.1470	2.4493	21.6540	0.6114	0.6
*S-176	0.2957	63.613	142.56	1.9783	0.33244	2.74662	0.5988	0.8	S-85	0.6421	63.168	184.65	2.9119	2.0730	18.1070	0.6137	0.6
*S-177	0.2957	61.679	118.07	3.9869	0.46667	3.85051	0.5919	0.8	S-86	0.6421	63.328	183.70	2.0019	1.7273	15.0351	0.6157	0.6
*S-178	0.2957	62.246	122.25	1.9982	0.33440	2.75005	0.5982	0.8	S-87	0.6421	64.308	187.19	5.6673	2.8719	24.7380	0.6051	0.6
*S-183	0.34553	64.654	121.86	9.0603	0.98333	7.79793	0.5916	0.8	S-88	0.6913	63.651	153.91	4.0804	2.9797	24.9431	0.6069	0.5
*S-184	0.34553	63.927	125.35	6.1508	0.808565	6.51081	0.5950	0.8	S-89	0.6913	62.857	155.37	2.9856	2.6253	21.4204	0.6054	0.5
*S-185	0.34553	63.714	129.60	3.9713	0.652083	5.29511	0.5997	0.8	S-91	0.6913	63.465	149.31	0.9705	1.4469	12.0775	0.6034	0.5
*S-186	0.34553	63.363	140.43	1.9861	0.460714	3.81043	0.6046	0.8	S-92	0.6913	63.061	139.47	2.9387	2.5517	21.1725	0.6096	0.5
*S-187	0.4001	64.258	116.36	9.033	1.3303	10.5377	0.5941	0.8	S-93	0.6913	62.991	137.13	1.0067	1.4781	12.4136	0.6069	0.5
*S-188	0.4001	64.189	127.04	6.129	1.0947	8.7989	0.5979	0.8	S-94	0.6913	63.092	161.41	4.1110	2.9610	25.2073	0.6056	0.5
*S-189	0.4001	63.843	136.46	4.009	0.88299	7.2170	0.6013	0.8	S-95	0.6913	63.573	165.92	3.0567	2.5567	21.7287	0.6059	0.5
*S-190	0.4001	62.944	142.63	1.999	0.62025	5.1751	0.6045	0.8	S-96	0.6913	63.828	167.14	2.0481	2.0881	17.7052	0.6038	0.5
*S-191	0.4499	64.189	143.45	8.978	1.04306	13.4760	0.5879	0.8	S-97	0.6913	63.214	169.60	1.0113	1.4720	12.6291	0.6094	0.5
*S-192	0.4499	64.289	127.83	9.210	1.09597	13.6247	0.5928	0.8	S-107	0.6913	63.728	125.80	4.1258	3.0158	24.3695	0.6002	0.5
*S-193	0.4499	64.327	136.27	6.159	1.38764	11.2586	0.5960	0.8	S-108	0.6913	65.432	136.52	3.1121	2.6632	20.8760	0.6013	0.5
*S-194	0.4499	64.070	141.64	4.054	1.12639	9.2341	0.5994	0.8	S-109	0.6913	63.614	142.64	2.0327	2.1141	17.4682	0.6064	0.5
*S-195	0.4499	63.774	147.23	1.989	0.78875	6.5383	0.6026	0.8	S-110	0.6913	65.294	137.45	0.9891	1.4930	11.9533	0.6045	0.5
S-157	0.49984	20.065	145.80	9.2966	1.174306	28.8202	0.5720	0.8	S-98	0.7407	63.879	160.49	2.6522	2.9371	24.6987	0.6166	0.4
S-158	0.49984	19.846	160.21	6.1347	0.90069	24.2454	0.5842	0.8	S-99	0.7407	64.094	165.75	1.9595	2.5315	21.3489	0.6192	0.4
S-159	0.49984	19.694	164.92	4.0510	0.79417	20.3082	0.5983	0.8	S-100	0.7407	63.986	161.79	1.1934	1.9887	16.7224	0.6223	0.4
S-160	0.49984	19.291	172.26	1.9985	0.56097	14.7570	0.6102	0.8	S-101	0.7407	64.435	157.50	0.6753	1.5137	12.5831	0.6261	0.4
S-153	0.49984	64.700	148.05	6.0663	1.72292	14.1047	0.5997	0.8	S-102	0.7900	62.648	160.79	1.9838	2.9606	25.3537	0.5956	0.3
S-154	0.49984	63.900	154.91	9.2085	2.08003	17.37058	0.5935	0.8	S-103	0.7900	63.780	140.71	1.5229	2.6519	21.8079	0.5967	0.3
S-155	0.49984	62.416	172.00	3.9650	1.35653	11.80980	0.6023	0.8	S-104	0.7900	63.196	165.78	1.1585	2.2564	19.2780	0.5933	0.3
S-156	0.49984	65.206	179.98	1.9682	0.97903	8.25820	0.6073	0.8	S-105	0.7900	62.705	170.25	0.7024	1.7056	14.7548	0.5796	0.3
S-149	0.49984	114.92	98.15	5.4481	2.19056	9.82880	0.5958	0.8	S-106	0.7900	63.945	140.67	0.7015	1.7742	14.5543	0.5874	0.3
S-150	0.49984	113.00	114.05	3.9402	1.83611	8.53369	0.5979	0.8	S-111	0.7900	63.705	142.84	1.1581	2.3733	19.5889	0.6135	0.3
S-151	0.49984	113.95	105.95	2.5183	1.49306	6.81819	0.6030	0.8	S-112	0.7900	64.634	144.20	0.7465	1.8268	14.8991	0.5846	0.3
S-152	0.49984	114.47	106.25	1.0358	0.96639	4.39591	0.6073	0.8	S-113	0.7900	64.882	150.52	1.8332	2.9303	23.9974	0.5998	0.3
*S-169	0.49984	19.821	205.70	8.9860	1.11111	29.62944	0.5735	0.8	S-114	0.7900	61.651	158.20	1.3471	2.4481	21.2187	0.6014	0.3
*S-170	0.49984	19.938	204.71	6.0680	0.92806	24.57809	0.5810	0.8									
*S-171	0.49984	20.186	208.82	4.0278	0.77153	20.29479	0.5908	0.8	S-115	0.8146	64.857	148.45	1.5562	2.9864	24.4040	0.5335	0.3
*S-172	0.49984	19.914	216.30	1.9471	0.53764	14.4444	0.5984	0.8	S-116	0.8146	64.068	153.36	1.1815	2.5747	21.4098	0.5325	0.3
*S-161	0.49984	65.385	142.37	9.2125	2.13194	17.1568	0.5970	0.8	S-117	0.8146	64.667	154.71	0.7861	2.0989	17.3307	0.5303	0.3
*S-162	0.49984	63.595	150.92	6.0745	1.71285	14.2985	0.6016	0.8	S-118	0.8146	64.335	154.12	0.4894	1.6308	13.5206	0.5232	0.3
*S-163	0.49984	63.688	156.94	4.0346	1.39514	11.7147	0.6031	0.8	S-122	0.83919	64.696	173.47	1.1781	2.9129	24.5686	0.6078	0.2
									S-123	0.83919	63.720	175.34	0.9195	2.5639	21.9764	0.6105	0.2
									S-124	0.83919	61.748	175.28	0.4558	1.3163	16.0378	0.6236	0.2

* No therometer well in place at 5 pipe diam.

in these results is that between S132 and S142. These were tests on a 20 per cent concentric orifice, using steam at a pressure of about 65 lb. per sq. in., but the pressure for run S132 was only about 62.1 lb. while for S142 and S144 it was 64.5 and 63.8 lb., respectively. This change in pressure would probably account for some changes in coefficient.

The six tests, 1014, 1062, 1041, 1046, 1064, and 1065, illustrate the close agreement of results when measuring the quantity of water by displacement, and when determining it by observing weight and temperature. The 25 per cent ratio concentric orifice was tested by both methods, and when tested at four foot head, a disagreement of only one part in six hundred was discovered. The 35 per cent ratio orifice was tested in a similar manner with a two-foot differential head, and the average value of the coefficient of discharge for two short weight runs was within one part in two thousand of the average value of the coefficient determined by the much longer displacement runs. The coefficients for the two displacement runs agreed with each other within one part in three thousand. These results are thought to be fairly conclusive proof that volume displacement tests of this nature when properly conducted have just as great a degree of accuracy as can be obtained from the direct weight of the water.

The curves in Figs. 15, 16, and 17 show the variation of the coefficient of discharge with differential head for the various orifices and the nozzle. These were plotted against the head in feet of water since most meters have head measuring devices which measure equal heads

TABLE 4 ECCENTRIC ORIFICES USING STEAM

Run no.	Orifice diameter ratio D_2/D_1	Steam pressure, lb. per sq. in. abs. 1 P.D. upstream	Superheat deg. fahr. 5 diam. upstream	Diff. head at orifice ft. H_2O	Weight of steam, lb. per sec.	Quantity of steam, cu. ft. per sec.	C coeff. of discharge	Location of downstream tap pipe diam.
S-125	0.6420	62.067	139.67	5.8225	2.9765	25.0850	0.6124	0.8
S-126	0.6420	64.956	171.17	4.3220	2.5894	21.6894	0.6166	0.8
S-127	0.6420	61.205	182.34	2.5449	1.9421	17.4324	0.6238	0.8
S-133	0.6913	63.518	180.19	3.4487	2.7031	23.3670	0.6081	0.7
S-134	0.6913	63.926	183.66	2.5020	2.23190	20.0040	0.6119	0.7
S-135	0.6913	63.170	183.58	1.4825	1.78363	15.5599	0.6148	0.7
S-136	0.6913	62.971	181.25	0.4656	1.00538	8.77326	0.6185	0.7
S-137	0.7402	64.809	171.71	2.0446	2.56056	21.5193	0.6130	0.6
S-138	0.7402	63.595	174.53	1.4678	2.16505	18.5750	0.6181	0.6
S-139	0.7402	61.700	178.04	0.9863	1.75694	15.5702	0.6219	0.6
S-140	0.7402	62.084	175.36	0.4780	1.23663	10.8610	0.6260	0.6

TABLE 5 COEFFICIENTS FOR NOZZLE USING STEAM

Run no.	Nozzle ratio D_2/D_1	Absolute pressure	Degrees superheat	Diff. hd. ft. H_2O	Lb. per second	Cu. ft. per sec.	K	Location downstream tap
*NS-1	0.5793	64.993	147.90	3.4698	2.93861	23.94909	0.9776	0.5
*NS-2	0.5793	63.707	156.63	2.2760	2.36343	19.83269	0.9851	0.5
*NS-3	0.5793	63.162	158.20	1.1135	1.61476	13.68119	0.9669	0.5
NS-4	0.5793	63.967	148.85	3.2879	2.80361	23.22091	0.9661	0.5
NS-5	0.5793	63.715	157.53	2.2001	2.30463	19.35774	0.9777	0.5
NS-6	0.5793	63.780	160.04	1.1784	1.69080	14.23111	0.9811	0.5

* No thermometer well in place at 5 pipe diam.

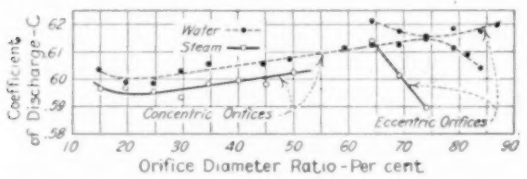


FIG. 18 VARIATION OF THE AVERAGE COEFFICIENT WITH DIAMETER RATIO FOR ALL THE ORIFICES

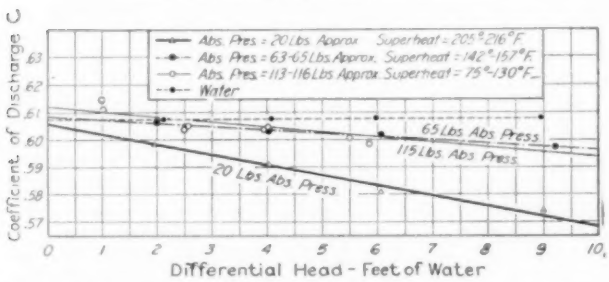


FIG. 19 VARIATION OF COEFFICIENT WITH HEAD FOR WATER AND STEAM AT VARIOUS PRESSURES USING 50 PER CENT CONCENTRIC ORIFICE

no matter what fluid is being metered, and in most of the test runs the head was actually measured in feet of water.

An inspection of the curves for the concentric orifices in Fig. 15 for water shows that they are, with the exception of the 64 per cent and 81.5 per cent diameter ratio orifices, practically horizontal, straight lines. The 64 per cent and 81.5 per cent diameter ratio coefficients of discharge seem to decrease with an increase of head. This, however, was only a tendency and was probably due to some eccentricity of installation. This peculiarity was noted with the 74 per cent and 79 per cent diameter ratio orifices which when first tested showed quite a variation of coefficient with head. When removed from the pipe and carefully inspected and reinstalled in the pipe, the variation did not appear, and the points which are plotted were determined. It seems to be apparent that if carefully installed, there were no appreciable variations in coefficient of discharge with head for a concentric orifice using water for the head ranges between two and nine feet of water.

The curves of the eccentric orifices for the tests with water seem to show that the coefficient has a tendency to increase with an increase of head. This tendency, however, is small, averaging for the three orifices tested only about 3 parts in 600 with a head differential variation of from two feet to nine feet of water. All of the curves for steam, however, except for the 19.6 per cent diameter ratio orifice show that the coefficient decreases rather rapidly with head. This is probably due to the effect of the expansion factor for steam which will be discussed later. The curves for all orifices below 70 per cent diameter ratio are practically parallel lines. The coefficients for the concentric orifices below 50 per cent diameter ratio are probably more accurate than the others as these were determined when using the "thick" orifice, and with no thermometer well placed in the pipe at five diameters upstream. It was felt that the thin orifices especially in the smaller diameter ratios might buckle when subjected to the pressures used. For this reason all results for test runs on "thin" small diameter ratio orifices were not considered accurate.

TABLE 6 COMPARISON OF COEFFICIENTS FOR ORIFICES REPLACED IN PIPE

Run no.	H Differential head	Superheat, deg. fahr.	C Coefficient of discharge	Remarks
1006	4.079	...	0.6092	6-in. concentric orifice in line twice—water
1052	4.020	...	0.6080	
1044	4.316	...	0.5968	Displacement test—water
1062	4.016	...	0.5978	Weighed water test
1041	2.108	...	0.6058	Displacement test
1046	2.155	...	0.6056	Displacement test
1064	2.093	...	0.6063	Weighed water test
1065	2.070	...	0.6048	Weighed water test
S-156	1.968	180	0.6073	Steam test, 6-in. concentric orifice
S-164	2.014	156	0.6063	in line twice
S-155	3.965	172	0.6023	6-in. concentric orifice in line twice—steam
S-163	4.035	157	0.6031	
S-132	1.911	134	0.5991	6-in. concentric orifice in line twice—steam
S-142	2.063	100	0.5964	
S-144	2.010	103	0.5987	

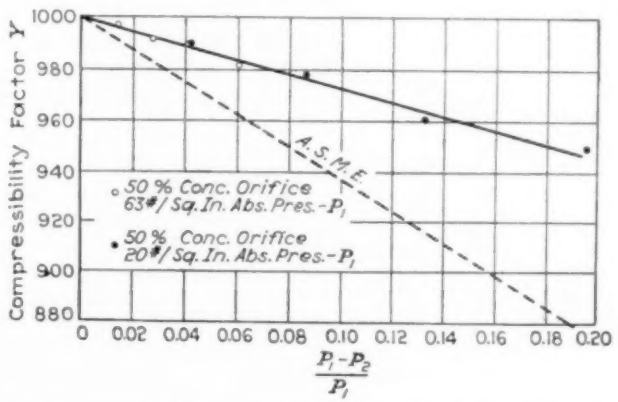


FIG. 20 GRAPH OF RATIO OF THE COEFFICIENT AS DETERMINED TO THE COEFFICIENT WITHOUT EXPANSION PLOTTED AGAINST THE RATIO OF THE PRESSURE DROP ACROSS THE ORIFICE TO THE ABSOLUTE PRESSURE UPSTREAM

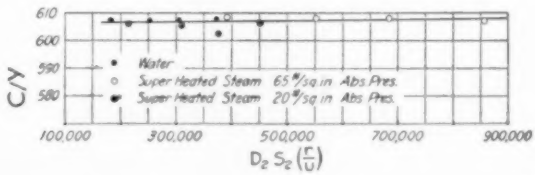


FIG. 21 GRAPH WITH VALUES OF C/Y PLOTTED AGAINST $D_2S_2\left(\frac{\gamma}{u}\right)$

The effect of a "finned" thermometer well in the pipe five pipe diameters above the orifice, upon the coefficient of discharge is not known, but it probably had some effect, and this effect would probably increase with an increase in the velocity of approach. For this reason, the absolute accuracy value of the coefficients as found for the orifices above 50 per cent diameter ratio is somewhat questionable.

Why the curve for the 20 per cent diameter ratio orifice should not be parallel to the others is not apparent, but for this orifice of small diameter the flow in the line was so small that it was difficult to maintain temperature conditions constant, especially for the low heads, and this no doubt would affect the accuracy of the coefficient.

For the larger orifices and for the one flow nozzle tested, it was impossible with the condensing equipment available to get high enough rates of flow to extend the curves far enough to make the results comparable. These points, however, have been included in the data, and the results are plotted to show that the tendency was for the coefficients for steam to fall close to those for water over the entire range of orifice sizes tested.

The data and curve for the nozzle are included only as a matter of interest, although there were not enough tests made to draw any conclusions. However, the effect of the thermometer well at five diameters upstream is shown by the higher coefficient curve obtained without the well as compared with the tests with the well inserted.

Fig. 18 shows the variation of the average coefficient for all the orifices with variation in diameter ratio. Compared with the curves published in the 1927 report of the Fluid Meters Committee, page 38, the coefficients determined for water seem to be about $1^1/_2$ per cent lower than those of curve "C" obtained by Spitzglass using air in a 6-in. pipe, though the general shape of the curve is about the same as those of Spitzglass for the 3-in. and the 6-in. pipe. The actual values of the coefficient agree very closely with those read from curve "A" obtained by Judd using water in a 5-in. pipe except for the diameter ratios above 60 per cent. Since Judd's results were based on downstream pressure readings at one-half pipe diameter below the orifice, and these are based on readings at the vena contracta, a disagreement might be expected especially at the higher values of orifice ratio.

The curve showing the variation of the coefficient for steam with orifice ratio appears to be about parallel, and about 1 per cent lower than the corresponding curve for water. This is due to the variation of the coefficient with head in the case of steam as shown in Fig. 15.

Fig. 19 shows the variation of coefficient with head for water, and for steam at 20 lb. per sq. in., at 65 lb. per sq. in., and at 115 lb. per sq. in. absolute pressure using a 50 per cent concentric orifice. All of these curves except the one for steam at 115 lb. per sq. in. seem to intersect the zero head line at the same place within the limits of accuracy of the experimental work. The curve for 115 lb. per sq. in. absolute pressure intersects the zero head line at a point about 1 per cent higher than do the others. The curve for water is practically a horizontal, straight line while those for steam at 20 and 65 lb. per sq. in. absolute pressure slope away from this, the 20-lb. curve sloping more than the 65-lb. curve. This agrees with theory as the effect of the compressibility of steam is greater for the lower absolute pressures with the same differential head. The slope of the 115-lb. line, however, lies between the 65- and 20-lb. lines. This apparent disagreement seems to throw doubt upon the accuracy of the values of the coefficient determined at 115 lb. per sq. in. For these tests the same apparatus was used for measuring everything except the static pressure. This pressure was measured by a 160-lb. pressure gage mounted two diameters upstream in place of the mercury manometer at one diameter upstream used for the other pressures. This gage was calibrated each day it was used, before and after being placed on the pipe, and so should be accurate, but the fact that it was used and that the results do not line up with the others tend to cause a belief that some inaccuracy in the pressure measurement is the cause of this error.

TABLE 7 6-IN. ORIFICE DIMENSIONS

Pipe Diameter = 6.121 in. = D_1

Orifice no.	D Average diam., in.	f Factor velocity of approach	R Ratio D_2/D_1	Remarks
		Thin Concentric Orifices		
18909	0.9043	1.0002	0.1477	
18910	1.5108	1.0019	0.2468	
18911	2.1149	1.0072	0.3455	
18912	2.7224	1.0202	0.4448	
17741	3.0225	1.0311	0.4938	
17660	3.6272	1.0680	0.5926	
17742	3.9300	1.0977	0.6421	
17743	4.2316	1.1384	0.6913	
17661	4.5337	1.1961	0.7407	
17744	4.8355	1.2799	0.7900	
19511	4.9861	1.3367	0.8146	
19512	5.1367	1.4085	0.83919	
19607	1.8098	1.0039	0.2957	
19606	1.2080	1.0008	0.1974	
		Eccentric Orifices		
17823	3.9294	1.0976	0.6420	
17662	4.2315	1.1384	0.6913	
17824	4.5306	1.1954	0.7402	
17663	4.8373	1.2805	0.7903	
17825	5.1365	1.4084	0.83916	
17826	5.3190	1.5255	0.8690	
		Thick Concentric Orifice		
23458	0.916	1.00025	0.14965	1st Reboring
23458	1.199	1.00073	0.19588	
23458	1.512	1.0019	0.24702	2nd Reboring
23458	1.810	1.00387	0.2957	3rd Reboring
23458	2.115	1.00725	0.34553	4th Reboring
23458	2.449	1.0131	0.4001	5th Reboring
23458	2.754	1.0212	0.4499	6th Reboring
23457	3.0595	1.03274	0.4999	
		6-In. Flow Nozzle Dimensions		
...	3.5458	1.0616	0.5793	

In Fig. 20 these tests are also used for a further study of the effect of compressibility. In it the compressibility factor Y, which is the ratio of the coefficient as determined to the coefficient without expansion, is plotted against the ratio of the pressure drop across the orifice to the absolute pressure upstream. Since the 115-lb. absolute-pressure line does not agree with the other pressure lines in Fig. 19, it was neglected in the determination of this curve. On the same sheet is plotted a curve of the calculated values of Y taken from the table, page 23, of the Fluid Meters Report. From these curves it will be seen that the values of Y determined experimentally do not check the calculated values. In getting the values for the curve the values of Y were calculated from the experimentally determined coefficients using the point of intersection of the head coefficient curve with the zero head axis as the value for the coefficient C where there was no expansion effect, and assuming that the coefficient for the orifice was constant over the head range used. The value of C for water was constant over this head range, and as water is practically incompressible this assumption should be correct. The points for both the 65 lb. per sq. in. and the 20 lb. per sq. in. absolute pressure tests fall fairly close to the same straight line, and seem to show that actual correction for compressibility is much less than the theoretical value. However, the evidence of eight runs is not enough to prove this fact, and an attempt is now being made to gather more data concerning it.

Apparently the amount of superheat had no effect on the coefficient as long as it was high enough to insure dry steam, and was constant during any run. This was true at least over the range of temperatures which was covered. An inspection of the last seven runs in Table 6 will show that a temperature difference of as much as 30 deg. fahr. had little, or no, effect on the value of the coefficient. However, any great change of temperature during a test tended to give a value of coefficient that was not

comparable since the change in temperature affected the pressure and the flow of steam a great deal.

An attempt was made to discover if it was possible to apply the viscous flow criterion to both steam and water, and to determine if this criterion would apply to orifices of different sizes. It was found that for the 50 per cent diameter ratio orifice using water and steam at 20 and 65 lb. per sq. in. absolute pressure, the curve as shown in Fig. 21 could be drawn. In this curve the values of C/Y have been plotted against $D_2S_2\left(\frac{\gamma}{u}\right)$, where D_2 is the orifice diameter in feet, S_2 the velocity of flow through the orifice in feet per second, γ the density in pounds per cubic foot, and u the absolute viscosity in foot-pound-second units of the fluid being metered. C/Y was determined by using the value of Y read from the curve plotted in Fig. 20 and the experimental values of C. The values of u were taken from the paper on the "Laws of the Similarity for Orifice and Nozzle Flows" by J. L. Hodgson, a paper presented by the committee on fluid meters at the 1928 annual meeting of the A.S.M.E.

It will be seen that all but one of the twelve points plotted fall very nearly on a straight line with the values of C/Y increasing slightly with increase of $D_2S_2\left(\frac{\gamma}{u}\right)$. Any attempt to apply this to orifices of other diameter ratios, however, gave simply a collection of points with no apparent order. This was more noticeable when the larger orifice coefficients were plotted. It appears from this study that this method of analysis can be applied only to orifices of the same diameter ratio. For any other ratio the change of the contour of the stream is enough to change the flow conditions.

As mentioned in an earlier part of this paper the Ohio State Engineering Experiment Station is carrying on this work using a three-inch test line. This will make possible a comparison between different pipe sizes, and make it possible to prove or to disprove the results of this first research, some of which are rather questionable.

This paper is submitted in the hope that the data already obtained may be of some value when applied to the art of metering fluids, and also in the hope that much interest will be aroused which will result in constructive criticism and many suggestions which will make the continuation of this research of greater value.

Appendix

Method of Calculation

The formula used for the calculation of the coefficient of discharge was the same for both water and steam and was as follows:

$$C = \frac{Q}{0.04376 A D_2^2 \sqrt{h}} \quad \ldots\ldots\ldots\ldots\ldots\ldots [1]$$

where C = coefficient of discharge
Q = quantity cubic feet per second
A = correction for velocity of approach
$= \sqrt{\frac{1}{1 - r^4}}$ where r equals ratio of orifice or nozzle diameter to pipe diameter
D_2 = diameter of orifice or nozzle, inches
h = differential head in feet of fluid flowing

This was derived from the equation for a head meter for liquids as found on page 19 of the A.S.M.E. Fluid Meters Report of 1927. In using this formula Q and h were determined experimentally and D and r were determined by measurements of pipes, orifices, and nozzles.

For tests using water the differential head was measured directly in feet of water, however the temperature of the water in the tubes was different from that of the water in the pipe, and the observed differential head was corrected by use of the formula:

$$h = h_0 \frac{r_0}{r} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots [2]$$

where h = actual differential head, feet
h_0 = head as read on manometer, feet
r_0 = density of water in manometer
r = density of fluid in pipe.

This formula is found on page 20 of the 1927 Fluid Meters Committee Report.

The density of the water was found from the tables of density found in Marks handbook, second edition, page 300.

The same formulas were used for the determination of the coefficients for steam. The value of the density of the steam at the upstream pressure tap, and the temperature at five diameters upstream was used as the value of r in formula [2].

This density was found from the steam tables published in Mechanical Engineering of February, 1926. The temperature of the water in the tubes was assumed to be the same as that of the surrounding air. A correction was applied for the unbalanced air column on the two sizes of the water manometer.

For some of the later tests using steam, the steam temperature was taken only at a point 30 diameters upstream. Enough tests were run previously to determine temperatures at both points, and to make possible the determination of the temperature at five diameters, and the proper correction was read from a curve.

HYD-52-7b

Quantity-Rate Fluid Meters

Correlation of Coefficients and Expansion Factors, Using the Reynolds Number and Acoustic Velocity Ratio

By ED S. SMITH, JR.,[1] PROVIDENCE, R. I.

PREFACE

Critical consideration is invited to the first section of this paper which is offered as of use in preparing a revised edition of the A.S.M.E. Fluid Meter Report—Part One. In it, the venturi tube and square-edged orifice are placed upon a common basis and simple graphical methods of use are shown. This common basis is rigorously derived from the laws of flow similarity and dimensional homogeneity in the body of the paper.

The dimensionally correct "hydraulic" formula is used as the common basis in metering fluids ranging from highly viscous oils to expansible fluids (gases) with both types of meters as follows:

$$Q = CY \frac{A_1}{\sqrt{(D_1/D_2)^4 - 1}} \sqrt{2PW_1}$$

which does not involve gravitation "g" in any way. This formula is derived for the venturi tube (or nozzle) with a frictionless liquid, in which case the coefficient "C" and the expansion factor "Y" are both unity.

Flow similarity and the correction factors "C" and "Y" depend upon two dimensionless ratios: the familiar Reynolds number "$V_1D_1W_1/U_1$" and the ratio "x/n," which has been christened the "acoustic ratio" by the author since it is derived from the acoustic velocity ratio "V_1/V_a" ("x" is the ratio of the differential to inlet pressure, "n" is the specific-heat ratio, "V_1" is the fluid velocity at the inlet, "V_a" is the acoustic velocity—speed of sound in the fluid at the inlet). These ratios are used for correlating data on "C," "Y," or "CY" experimentally obtained with whatever fluids are convenient and then the correlated data are used with whatever fluid is being metered commercially.

For liquid meters, the coefficient "C" depends only on the Reynolds number including when the coefficient is substantially flat, which it generally is, except for unusually small meters or for those measuring viscous fluids such as heavy oils or hydrogen gas. For a liquid, the coefficient "C" does not depend at all upon the acoustic ratio "x/n," since there are practically no expansibility effects involved. This is also true for a gas when the differential is relatively low. Also where the differential is relatively high, exactly the same coefficient would be used for a gas as for a liquid having the same Reynolds number, or as though the gas were a liquid having the same density and viscosity. The net expansion factor "Y" corrects for all of the expansion effect which exists due to the gaseous nature of the fluid.

In the usual measurement of gaseous fluids, the coefficient "C," defined as above, is "flat" with any given meter, and so the expansion factor "Y" is the only one which varies appreciably with the rate of flow, being unity at the low rates. Thus, gaseous fluids are most simply metered in practice by using the "hydraulic" equation with the necessary correction factor "Y" for expansion, and then the expansion factor "Y" may be taken as a straight-line function of the acoustic ratio "x/n." The conditions at the inlet are used throughout to make the method universal for venturi tubes and orifices and to gain other important advantages.

An original example illustrates the relation between flow similarity, Reynolds number, and coefficient by using parallel streams of fluid following "viscous" and "inertia" resistance laws, respectively.

The paper also presents material on factors which cause departure from the theoretical flow conditions, illustrations of the flow with venturi tubes and orifices, viscosity, friction loss, and flow in pipes, as well as other related matter.

INTRODUCTION BY R. J. S. PIGOTT

Chief Engineer, Research Dept., Gulf Prod. & Pipe Line Cos., Pittsburgh, Pa., and Chairman of A.S.M.E. Special Research Committee on Fluid Meters.

WITH a view to showing the great value of this paper by Mr. Smith, the events which led up to its publication may be of interest. The Special Research Committee on Fluid Meters of The American Society of Mechanical Engineers was formed in 1915, for the purpose of establishing, if possible, a clear understanding of fluid measurement. Knowledge of various meters, their coefficients, and general behavior was incomplete and scattered. The Committee very early decided to produce a treatise on the theory and application of fluid meters which would contain all of the authoritative data available.

In the course of this work some research was undertaken on pulsating flow: the remainder of the data has become available largely without actual laboratory work by the Committee.

The first report, Part I, on theory, was published in 1923, having taken eight years to assemble. Parts II and III on application and commercial meters are now in preparation.

The work of the Committee has been the means of bringing out several important papers on the subject. The latest was a most important paper by John L. Hodgson, summarizing his long experience in the subject; the present contribution by Mr. Smith likewise represents the results of many years' work, and is another stepping stone in the establishment of reliable knowledge of fluid measurement.

Simple relations exist between the fundamental quantities involved in metering fluids. The modern science of hydrodynamics, with its use of simple dimensionless ratios and basic formulas, has largely cleared the "art of fluid measurement" of involved, and sometimes even dimensionally wrong, formulas expressing the necessarily empirical relations between pressure differential and fluid velocity. Most of the formulas of the older conventional hydraulics were based on the relation $H = V^2/2g$, and had no proper method of making corrections for appreciably viscous and expansible fluids.

[1] Hydraulic Engineer, Builders Iron Foundry. Jun. A.S.M.E.

Contributed by the Special Research Committee on Fluid Meters and presented at the Annual Meeting, New York, N. Y., December 2 to 6, 1929, of The American Society of Mechanical Engineers. Previously discussed at an open meeting of the A.S.M.E. Special Research Committee on Fluid Meters, Rochester, N. Y., May 13 to 16, 1929, and subsequently presented at the World Engineering Congress, Tokyo, Japan, October 29 to November 22, 1929, as the A.S.M.E. contribution. For discussion and author's closure, see pp. 123–135.

Innumerable experiments on geometrically similar models in both ship and airplane design have confirmed the modern theory of flow similarity in which the basic dimensionally correct formula states that the resistance (or pressure differential for a quantity-rate meter) is merely proportional to the product of the square of the velocity and the density of the fluid, with the necessary empirical constant simply a function of the two dimensionless ratios, namely, the Reynolds number and the acoustic-velocity ratio.[2]

In 1919 and 1920 the author pioneered in making practical use of the Reynolds number with viscous-liquid measurements, obtaining data with one liquid and then using these data with other liquids. Now, ten years later, he has again pioneered in making practical use of the acoustic-velocity ratio (or more simply, the acoustic ratio) with expansible fluids, obtaining data with one gaseous fluid and then applying these data to other gaseous fluids. In both cases his contributions are of the useful engineering nature of "reduction to practice" of established physical laws, rather than of a purely scientific nature.

In 1919 the author made use of the Reynolds number with data on the friction loss in oil pipe lines. Although anticipated by numerous experimenters in this field, still his personal experience confirmed definitely the soundness of this method for correlating data for various fluids for flow in pipes. Later, while testing oil pumps, it became economically necessary to calibrate a venturi meter and use this to obtain instantaneous measurements of quantity rate instead of using the flow into available large cylindrical storage tanks over a considerable period of time.

The author undertook this problem for graduate research work at the University of California, and in 1920 filed a thesis progress report which correlated full-scale venturi-tube coefficients with the Reynolds number, confirming the use of this method for various liquids, and reaching a low coefficient of 0.91.

Continuation of this work on a wide variety of fluids with a geometrically similar glass model of a venturi tube showed that this method was basically correct for various sizes of meters, and a low coefficient of about 0.012 (the usual coefficient for water is about 0.990) was reached. The equation relating coefficient and Reynolds' number for the streamline régime was definitely formulated, and the relation established between the value of the coefficient and the turbulence—as defined by the Reynolds number. Also similarity of flow was observed by visual study of dye streams in the glass venturi tube and pipe over a very wide range of Reynolds' numbers.

These results, along with a practical method of metering viscous fluids, were stated in "The Oil Venturi Meter," a paper published in 1923 by The American Society of Mechanical Engineers (R-1)[3] and attention was called to the possibility of correlating the expansion correction with the acoustic-velocity ratio for metering gases. This method of metering viscous liquids has been successfully used by the author since that time and has proved reliable from his subsequent experience.

In 1924, the "Fluid Meter Report—Part One," which was issued by the A.S.M.E. Special Research Committee on Fluid Meters (R-2), presented an independent correlation of venturi-tube coefficients, for water, with the Reynolds number made by Dr. E. Buckingham, which agreed substantially with the author's 1923 correlation using data obtained on various oils, various solutions of corn syrup and water, with both model and full-scale venturi tubes.

In 1925 the author assisted (R-3) Prof. Robert L. Daugherty in correlating the coefficients for full-scale orifices, for various oils, with the Reynolds numbers from the original data, and this work again directly confirmed the use of Reynolds' numbers for this differential producer. It is thus believed that the author is among the pioneers in the "reduction to fluid-meter practice" of dimensionless ratios, as he had contact with what are believed to be the first published correlations for full-scale venturi tubes and square-edged orifices using a variety of liquids.

In the "Fluid Meter Report—Part One," the venturi-tube theory was adequately shown with the correlation of coefficient and Reynolds' number for viscous fluids, and the expansion factor for gaseous fluids was determined by the conventional "adiabatic" expansion formula, using St. Venant's equation. For orifices, however, neither the variation of coefficient with low Reynolds' numbers for viscous fluids nor the increase of the *vena contracta* with expansion of gaseous fluids was presented.

As a member of the A.S.M.E. Special Research Committee on Fluid Meters, the author has presented a tentative revision of the "Fluid Meter Report—Part One," to the Committee, and the present notes are taken chiefly from this revision. In this the venturi tube and orifice are placed on a common basis, and a practical method of using it is also shown.

The acoustic-velocity ratio has been developed and its use simplified by the author until it forms a practical basis for correlating expansion corrections for both the venturi tube and orifice for various fluids. In the past several years a number of investigators have published material which establishes confidence in the soundness of this use of the acoustic-velocity ratio for fluid meters.

The art of metering fluids is becoming of increasing importance since accurate quantity-rate meters are required for modern industrial uses and for the control of continuous processes in the manufacture of enormous quantities of highly uniform products.

Critical consideration and discussion of both the basic theory and method of practical use are invited so that the revised edition of the "Fluid Meter Report," will be accurate and of the highest usefulness.

R. J. S. PIGOTT.

Quantity-Rate Fluid Meters

INTRODUCTION

VENTURI tubes and square-edged orifices are increasingly used in commercially metering a wide range of fluids, including highly viscous liquids and expansible gases. The following notes are offered to place the common fundamental relations upon which both forms of meters depend in convenient form for general commercial use.

For commercial metering the "hydraulic" equation is convenient, that is,

$$Q_1 = CY \frac{A_1}{\sqrt{(D_1/D_2)^4 - 1}} \sqrt{2P/W_1}$$

This is originally derived for the venturi tube (or nozzle) with a frictionless and non-expansible liquid, in which case the coefficient C and expansion factor Y are both equal to unity.

Flow similarity and the combined coefficient and expansion factor depend, respectively, upon two dimensionless ratios: the Reynolds number and the acoustic-velocity ratio. For liquid

[2] The Reynolds number $= VDW/U$, and the acoustic-velocity ratio $= V/V_a$, where V = velocity of fluid, D = a linear dimension, W = density of fluid, U = viscosity of fluid, and V_a = velocity of sound in fluid. These ratios are significant of the actual state of flow, i.e., indicate truly the "flow similarity." The acoustic-velocity ratio can be more simply expressed as the acoustic ratio x/n, where x is the ratio of differential to inlet pressure and n is the specific-heat ratio.

[3] This and subsequent similar notation in the text refer to numbered references in Appendix No. 4.

meters the acoustic-velocity ratio naturally does not enter since it measures only expansion effects, and so the flow similarity and the coefficient C depend only upon the Reynolds number including when the coefficient is substantially "flat" (or constant), which it generally is, except for unusually small meters or for those measuring viscous oils.

For gaseous fluids at low differentials there is no appreciable effect of expansion, and so the Reynolds number determines the coefficient C, using the kinematic viscosity of the gas exactly as though it were for a liquid. Hydrogen gas, for example, has approximately one hundred times the kinematic viscosity of water, and so behaves similarly to a moderately viscous oil. For most gas measurement, however, the coefficient in the hydraulic formula is "flat," and the expansion effect may be corrected by a factor which depends only upon the acoustic ratio: a simplified form of the acoustic-velocity ratio.

In practice, coefficients and expansion factors may usually be determined separately and then combined as above with satisfactory accuracy, even where there are considerable departures from the "flat" coefficient and an expansion factor of unity. Fundamentally, however, the coefficient and expansion factor are interdependent, and for a rigorous treatment must be considered simultaneously; for example, say, plotting a family of "solid" curves of combined coefficients and expansion factors for various acoustic ratios against the Reynolds number.

Flow similarity in these notes refers to geometrical similarity of flow lines and velocity distribution, and is determined by the Reynolds number together with the acoustic ratio. For example, with square-edged orifices, the size of the *vena contracta* increases with either a decrease of the Reynolds number or an increase of the acoustic ratio. Also the expansion of the fluid itself is a function of the acoustic ratio, and therefore the combined correction for the expanded *vena contracta* and the fluid expansion is a function of the acoustic ratio x/n, where x is the ratio of the differential pressure to the absolute pressure at the inlet, and n is the ratio of the specific heats. The combined expansion factor Y is thus found to be a nearly straight-line function against this simple acoustic ratio x/n in the usual differential range for gaseous fluid meters, with either venturi tubes or orifices. The data presented are purely for the illustration of the use of this method.

The above use of the "hydraulic" equation with factors depending upon the dimensionless ratios noted is rigorous and also simple. The acoustic ratio x/n is practical in use and further is particularly valuable in metering one gaseous fluid, using data obtained with another gaseous fluid.

The upstream (pipe) diameter governs the character of the flow and amount of turbulence in the pipe, and this diameter has been used as a common basis throughout, making the basis universal for pipe-friction loss, venturi tubes, orifices, and pitot tubes.[4] Otherwise, for orifices, for example, the diameter of the variable and generally indeterminate *vena contracta* would have to be used to correspond strictly with the downstream (throat) diameter of the venturi tube, and there is no corresponding constriction for the pitot tube. Also the fluid conditions may be exactly known at the upstream connection, but those at the downstream connection depend also upon the shape of the differential producer, and so are generally more or less indeterminate—as is shown, in fact, by the necessity of using coefficients.

All of the relations that can be expressed using the downstream diameter as a basis can be converted to use the upstream diameter equally well, although sometimes at a partial loss of the physical significance of critical phenomena. For example, coefficients for different shapes of venturi tubes are more closely correlated using the throat diameter as the linear dimension in the Reynolds number. However, the results may then be transferred to the pipe-diameter basis for convenience in actual use.

An additional advantage of using the upstream diameter in the Reynolds number as a convenient basis appeals to any one in selecting a differential producer to produce a given differential for a known flow through a certain size of pipe, the usual case in practice. From these conditions an equivalent "effective area" can be simply found, and from its ratio to the "pipe area" the required venturi tube throat (or orifice) diameter can be simply found by referring to the relation between this ratio of areas and the ratio of the required diameter to the pipe diameter.

It is believed that one portion of these notes presents a clear and original picture of fluid flow: an example illustrating the relation between flow similarity, Reynolds' number, and coefficient by the use of parallel streams of fluid following "viscous" and "inertia" resistance laws, respectively.

The basis for the author's revision is, briefly, the "hydraulic" equation, using the coefficient and expansion factor as determined from the Reynolds number and acoustic ratio. These notes also include their derivation from fundamental theory. A discussion of other factors affecting coefficients, along with some material on related subjects, is also included.

Grateful acknowledgment is due R. J. S. Pigott, Prof. W. S. Pardoe, and J. M. Spitzglass, colleagues on the Fluid Meter Committee, to Alton C. Chick and to John L. Hodgson, of Geo. Kent, Ltd., London, outstanding authority on fluid measurement, for their direct assistance in critically discussing these notes; to the Builders Iron Foundry for their willing cooperation; also to Winslow H. Herschel and Edgar Buckingham, who generously offered pertinent references and suggestions; and finally to Prof. Robert L. Daugherty for his friendly interest over a period of years, which has assisted in making this work possible.

I—FLUID METERING IN PRACTICE

(This chapter is offered as a basis for the revision of the A.S.M.E. Fluid Meter Report—Part I)

The Hydraulic Equation

It is convenient to use the "hydraulic" formula, which assumes that the fluid is neither appreciably viscous nor expansible, with experimentally determined coefficients for the effects of viscosity and expansibility.

This hydraulic formula is

$$Q_w = CY \frac{A_1}{\sqrt{(D_1/D_2)^4 - 1}} \sqrt{2PW_1} \quad \ldots\ldots\ldots [1]$$

or

$$Q_w = CYA \sqrt{2PW_1} \quad \ldots\ldots\ldots\ldots [2]$$

in which the following notation is used:

Symbol	Dimension	Unit	Quantity
Q_w	M/T	Lb. per sec.	Mass discharge
C	....	.....	Coefficient, correcting for viscosity, as determined by data correlated with the Reynolds number

[4] The use of simple graphical methods is ideal for correlating the experimentally determined coefficient and expansion factor with the Reynolds number and acoustic ratio. There has been much confusion due to attempts, including extremely clever ones, to express empirical data by complicated formulas instead of by simple graphs, using proper scales (logarithmic, semi-logarithmic, or whatever is most suitable). In oil flow there are three distinct régimes: streamline, superturbulent, and the usual turbulent flow (hydraulic), and this condition practically forces the use of graphical methods.

Y	....	.....	Factor correcting for expansion of gaseous fluids (Y = 1.000 for liquids)
A_1	L^2	Sq. ft.	Area of inlet (pipe)
A	L^2	Sq. ft.	Effective area of section of maximum constriction
D_1	L	Ft.	Inlet diameter (pipe)
D_2	L	Ft.	Diameter of throat of venturi tube or diameter of orifice
P	M/LT^2	Poundals per sq. ft.	Differential pressure
W_1	M/L^3	Lb. per cu. ft.	Density of fluid at inlet

For convenient use, in ordinary English units,

$$Q_w = 457.2\, CYa\sqrt{hW_1} \quad \text{.....[3]}$$

in which the following notation is employed:

Symbol	Dimension	Unit	Quantity
Q_w	M/T	Lb. per hr.	Mass discharge
457.2	$M^{1/2}/T$	.....	A physical constant
C	....	.....	Coefficient correction for viscous effects, as determined by Reynolds' number
Y	....	.....	Factor correcting for expansion of gaseous fluids
a	L^2	Sq. in.	Area of venturi tube throat or orifice corrected for approach velocity
a_1	L^2	Sq. in.	Area of inlet (pipe)
a_2	L^2	Sq. in.	Area of venturi tube throat or orifice
h	L	In.	Water differential
W_1	M/L^3	Lb. per cu. ft.	Density of fluid at inlet

$$a = \frac{a_1}{\sqrt{(d_1/d_2)^4 - 1}} = a_2\sqrt{\frac{(d_1/d_2)^4}{(d_1/d_2)^4 - 1}} \quad \text{.....[4]}$$

The absolute system of mass units is quite satisfactory, since using "pounds" for the mass unit and "poundals" for the force unit (grams and dynes correspondingly in the absolute c.g.s. system) gives the discharge in familiar mass units, while the unfamiliar force unit (poundal) is replaced in practical use by the differential head h.

The Coefficient

Fig. 1 shows the relation between the coefficient C and the Reynolds number for various differential producers. It is apparent that for a velocity range of 1 to 10, which is customary in commercial metering, the coefficient is generally substantially "flat" for commercial sizes of meters, the exceptions being for unusually small pipe sizes or for fluids having high kinematic viscosity, such as heavy oils or hydrogen gas.

The Expansion Factor

Fig. 2 shows the expansion factors for use with gaseous fluids for various differential producers.[5]

The orifice expansion correction is considerably less than that for nozzles. In both cases the factor depends upon the "acoustic-velocity ratio," which may be more simply expressed as the "acoustic ratio" x/n, where x is the ratio of the differential to the inlet pressure, and n is the ratio of specific heats for constant pressure and constant volume.

Although with the venturi tube these factors are computed for frictionless adiabatic expansion, it will be found that the above equation applies nearly exactly when using the friction coefficient C and the expansion factor Y here, as above stated, for metering various gaseous fluids with the usual commercial sizes of meters.

For the orifice, the expansion factor Y, as experimentally determined, includes the effect of both the increase in the size of *vena contracta* and also the expansion of the fluid itself. This expansion correction Y for orifices is best determined from high-pressure tests (R-4) in which the coefficient C is flat, i.e., has no appreciable variation with the Reynolds number.

The relation between this flat coefficient C for square-edged orifices having flange taps and various ratios of orifice diameter to pipe diameter is shown in Fig. 3.

The expansion correction in Fig. 2 is plotted against the acoustic ratio, which basis is entirely satisfactory for commercial metering with the differentials generally used. This graph shows

[5] The computations throughout these notes have been made with a 20-in. slide rule, and this is adequate, since the data are only presented to illustrate the method.

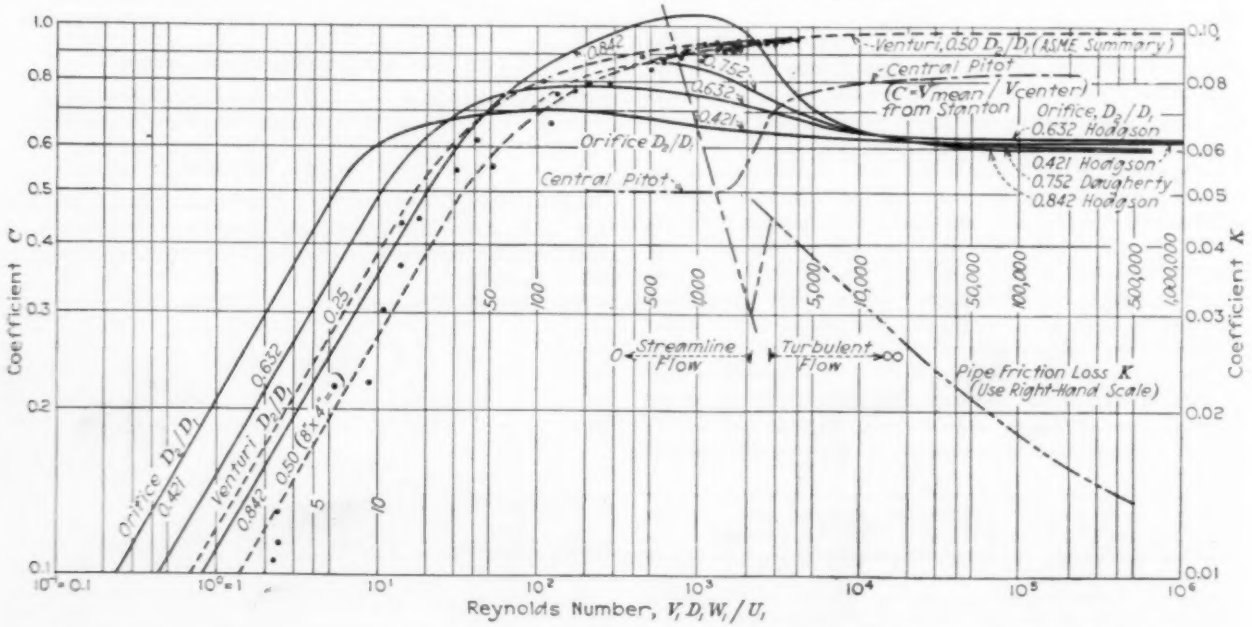


Fig. 1 Fluid-Meter Coefficient C and Pipe-Friction Loss K Plotted Against Reynolds' Number $V_1 D_1 W_1 / U_1$

that a nearly straight-line relation exists for all values of the expansion correction y and the acoustic ratio x/n. The expansion factor Y depends on the correction y as follows:

$$Y = 1 - y \quad [5]$$

It is generally most convenient to apply this nearly straight-line relation by using one of the following equations:

$$y = bx/n \quad [6]$$

$$Y = 1 - bx/n \quad [7]$$

since b is practically constant. (R-5, R-6.)

Fig. 4 shows the relation between b and the ratio of diameters d_2/d_1 for the venturi tube or nozzle, and also for square-edged orifices with flange taps.

Another familiar form involves taking the square root of a similar factor:

$$Y = \sqrt{1 - b'x/n} \quad [8]$$

However, the simpler form (which does not include the radical), $Y = 1 - bx/n$, as used above, is satisfactory for general commercial

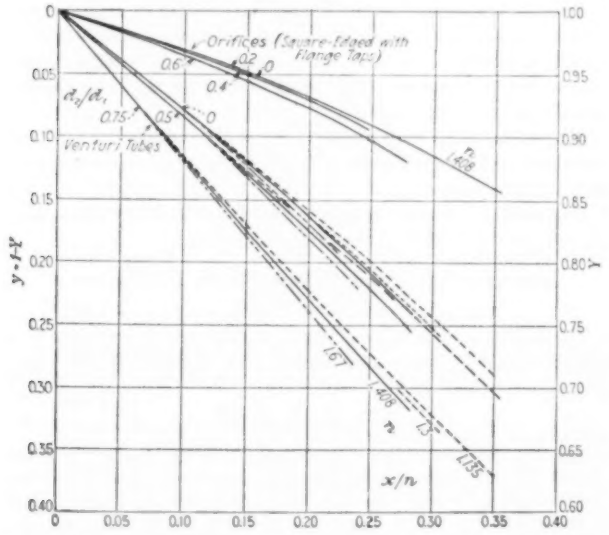


FIG. 2 RELATION BETWEEN EXPANSION FACTOR Y AND ACOUSTIC RATIO x/n

(Orifice data from U. S. Bureau of Stds. R.P.49.)

use, since the error produced by this approximation is generally less than 0.2 per cent; and also it will be observed that some adiabatic corrections for venturi tubes are concave upward and some the reverse.

In 1914, G. B. Upton used the following formula for venturi tubes: $b = (3 + R^4)/4(1 - R^4)$, where $R = d_2/d_1$.

For most orifices the use of the separate factors C and Y determined above will be entirely satisfactory. However, for very small orifices or highly viscous gaseous fluids it may be necessary to use the experimentally determined combined factor CY plotted against the Reynolds number for various acoustic ratios.

Although the use of the simple acoustic ratio x/n is only shown for nearly adiabatic expansions, this method will be found useful where the flow through the differential producer follows a somewhat different expansion law (as with orifices with pipe connections which really measure the "loss" due to the throttling orifice). This is due to the fact that the acoustic ratio x/n is primarily based upon the use of the velocity of the fluid in the pipe at the inlet and the acoustic velocity also at the same point, and at the usual low differentials the velocity of sound will not change appreciably along the differential producer.

The above method of metering uses only known quantities in the hydraulic formula, and the experimentally determined factors, coefficient C and expansion factor Y, are obtained from two dimensionless ratios (Reynolds' number and acoustic ratio), both of which depend only upon known quantities. Since both the Reynolds number and acoustic ratio are based upon the conditions in the pipe at the inlet of the differential producer, the above method is suitable for use with any geometrically similar differential producers.

STANDARD DIFFERENTIAL PRODUCERS—DEFINITIONS

Venturi Tube. The term "venturi tube" refers specifically to a constant-angle conical contracting inlet section having a

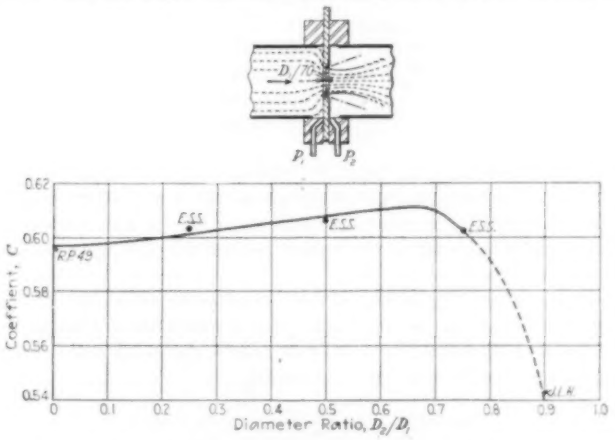


FIG. 3 RELATION BETWEEN "FLAT" COEFFICIENT C FOR SQUARE-EDGED ORIFICES WITH FLANGE TAPS AND DIAMETER RATIO D_2/D_1 AT HIGH VALUES OF REYNOLDS' NUMBER

cylindrical inlet and throat with gently curving transition surfaces, and having an annulus at both inlet and throat. A downstream pressure-recovery constant-angle cone is generally added to reduce the loss of head to a minimum, the overall length depending also upon the throat diameter.

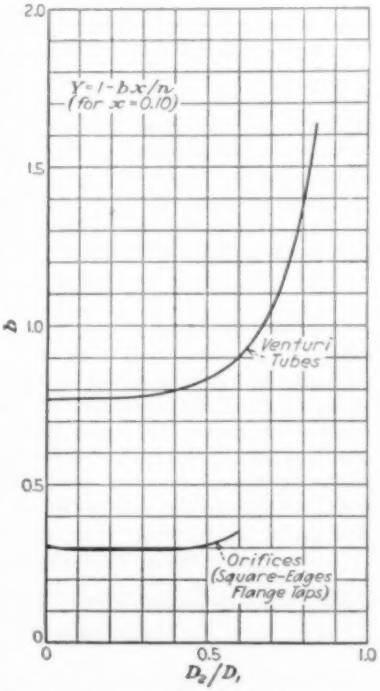


FIG. 4 RELATION BETWEEN EXPANSION CONSTANT b AND DIAMETER RATIO D_2/D_1 FOR VENTURI TUBES AND SQUARE-EDGED ORIFICES

Venturi Nozzle. The venturi nozzle consists specifically of a conoidal contracting section containing an inlet and throat connections each having an annulus (as with standard venturi-tube construction, hence "venturi" nozzle). Although there is generally no downstream recovery cone, a short cone is sometimes added which cuts the loss practically in half. This venturi nozzle is generally somewhat shorter between inlet and throat connections than the standard venturi, one pipe diameter being the standard distance between connections; the overall length is the same for a given pipe size regardless of the size of the throat.

Venturi Nozzle-Tube. This is merely the venturi nozzle with an added short recovery cone (of standard length regardless of diameter ratio).

Flow Nozzle. The flow nozzle consists merely of a nozzle-

shaped constriction which is placed between two pipe flanges. There is no annulus at either section, both connections being generally made in the wall of the pipe near the nozzle, as with square-edged orifices.

Square-Edged Orifice With Flange Connections. This consists generally of a concentric circular aperture in a thin-plate diaphragm, having single upstream and downstream pressure connections in the flanges immediately adjacent to the orifice, or in the face of the orifice itself. The leading edge of the orifice is square-edged, while the downstream edge may be beveled to make the thickness at the edge less than one-seventieth of the pipe diameter.

Square-Edged Orifice With Throat Connections. With an orifice between flanges as before, single connections are made to the pipe wall at 1 pipe diameter upstream and $^1/_2$ pipe diameter downstream, the latter corresponding roughly with the *vena contracta*.

Square-Edged Orifice With Pipe Connections. With an orifice between flanges as before, single-point connections are made in the wall of the pipe at $2^1/_2$ pipe diameters upstream and 8 pipe diameters downstream from the orifice.

II—BASIC FORMULA FOR QUANTITY-RATE METERS

General Principles

The common characteristic which classes the quantity-rate meters together is that the flow through the differential producer causes a pressure difference which the secondary element then uses as an indication of the rate of flow. This difference may be between the pressure at two different sections of the stream, as in the venturi tube, flow nozzle, and orifice meters, or it may be the difference between pressures at two points in the same section of a curved stream, as in the centrifugal meter and the flow bend, or, finally, it may be the difference between pressures at the same point, as in the pitot or impact-tube meter. To have calibrations apply generally, it is necessary to use geometrically similar differential producers.

Relation of Differential to Rate

Generally there is a simple relation between the quantity-rate and the differential: it is, as a rule, almost exactly proportional to the square root of the differential. The proper multiplier is usually determined by calibration against a basic standard, measuring the actual volume or weight in a certain period of time. The multiplier depends primarily on the shape of the differential producer, including the usually slight effect of the roughness, the size of the pipe and the density of the fluid.

Basic General Formula

The above relation can be expressed in the following equation:

$$Q_1 = kD_1^2 \sqrt{P/W_1} \quad \ldots\ldots [9]$$

which does not involve gravitation in any way.[6] The following notation is employed:

Symbol	Dimension	Unit	Quantity
Q_1	L^3/T	Cu. ft. per sec.	Quantity-rate of flow under upstream conditions
k	....	.....	Constant determined by calibration
D_1	L	Ft.	Diameter of pipe
W_1	M/L^3	Lb. per cu. ft.	Density of fluid under upstream conditions
P	M/LT^2	Poundals per sq. ft.	Differential pressure

However, Equation [9] may be rewritten as

$$Q_1 = kD_1^2 \sqrt{gH_1} \quad \ldots\ldots [10]$$

where

H_1	L	Ft.	Differential head of the fluid itself (of density w_1) acted upon by gravity
g	L/T^2	Ft. per sec. per sec.	Acceleration of gravity

The constant k, as determined by calibration, is found to depend upon the size and shape of the differential producer—including the usually slight effect of roughness in the differential producer and also of the pipe itself as well as the influence of fittings, upstream especially, and the amount of straight pipe between them and the differential producer; this constant k being expressed as a function of the Reynolds number, which relates the quantity-rate of flow, size of pipe, and the density and viscosity of the fluid.

Where the fluid is gaseous, there is the additional effect of the usually slight expansion of the fluid as it passes through the differential producer.

Differential producers generally used for closed conduits (pipes) are the venturi tube and the thin-plate orifice.

Derivation of Basic Formula From Dimensional Homogeneity

The basic formula for any shape of differential producer is as follows, being dimensionally correct, and does not depend upon gravitation in any way:

$$Q_1 = kD_1^2 \sqrt{P/W_1} \quad \ldots\ldots [9]$$

---

[6] In the basic formulas throughout this work, the unit "pounds" means "pounds mass" (the British "absolute" unit) instead of "pounds force" (the customary "engineering" or "gravitational" system, also known as the "foot-pound-second" system). This absolute system has been used in these notes to avoid the misleading appearance of the symbol g in a formula, e.g., Equation [9], expressing a physical relation into which gravitation absolutely does *not* enter.

Referring to the unfortunate customary use of "pounds force," when it falsely appears to involve gravitation in such equations, Lord Rayleigh wrote (R-7):

"Engineers, who might make much more use of it (the principle of similitude) than they have done, employ a notation which tends to obscure it. I refer to the manner in which gravity is treated. When the question under consideration depends essentially upon gravity, the symbol of gravity g makes no appearance; but when gravity does not enter into the question at all, g obtrudes itself conspicuously."

Since it is necessary to make use of viscosity in metering fluids, it is important that the relatively unfamiliar units for viscosity be clearly expressed in absolute units, their use for this quantity having been established by physicists in practically all available published data.

"It is not desirable to tabulate viscosities in gravitational or engineers' units, as such values involve the value of g, which varies from place to place (R-8)."

The British "absolute" system of units has been used in this work since it corresponds dimensionally throughout with the metric "absolute" or c.g.s. system of units. Although there should be no confusion concerning the units for length and time, it may be well to make the following statement concerning mass and force units, as related by the gravitational acceleration g:

In (metric) absolute c.g.s. units, a force of one dyne will give a mass of one gram an acceleration of one centimeter per second per second. Correspondingly in (British) absolute f.p.s. units, a force of one poundal will give a mass of one pound an acceleration of one foot per second per second.

The international gravitational standard at sea level and 45 deg. latitude is that it takes 32.174 poundals to make one pound of force, in the customary "engineering" units.

A fluid, water, may have a density (mass per unit of volume) of 62.37 lb. per cu. ft. at 60 deg. fahr. A cubic foot of this fluid would then weigh (exert a force of) 62.37 lb. in the customary engineering or gravitational units under the acceleration of gravity g. This weight-force would then equal $g \times 62.37$ or $32.174 \times 62.37 = 2{,}007$ poundals, i.e., expressed in absolute force units. This system gives the mass discharge in customary units, and since the differential pressure is ordinarily expressed as head, there is practically no use made of the unfamiliar "poundal."

where P and W_1 are expressed in mass units. However, $P/W_1 = gH_1$, as will be immediately recognized since $P = W_1gH_1$. Writing the formula for both the quantities and their dimensional units:

$$Q_1 = kD_1^2\sqrt{2gH_1} \quad \ldots\ldots\ldots\ldots\ldots [11]$$

and

$$\frac{\text{ft.}^3}{\text{sec.}} = \text{ft.}^2\left(\frac{\text{ft.}}{\text{sec.}^2} \times \text{ft.}\right)^{1/2} \quad \ldots\ldots\ldots\ldots [12]$$

$$= \frac{\text{ft.}^3}{\text{sec.}}$$

It is seen that the dimensions on both sides of the equation are equal.

The factor k may equal the coefficient C or may include some constant which will make the equation either more convenient to use, or cause the value of the coefficient C to have some particular physical significance, such as the correction for friction in the inlet cone of a venturi tube or the correction for the combined effects of contraction and friction in the square-edged orifice, having pressure connections one diameter upstream of the orifice and at the *vena contracta* downstream.

It should be observed here that the relation between differential pressure and quantity is perfectly general for any shape of differential producer which is exposed to the dynamic action of a flowing stream of fluid in a closed conduit. This formula may be used in connection with such diverse differential producers as friction loss in a length of straight pipe, square-edged orifice, venturi tube, pitot tube, elbow, or other fixed shape.

III—COEFFICIENT AND EXPANSION FACTOR CORRELATED BY DIMENSIONLESS RATIOS

Coefficient and Reynolds' Number

Physical Significance of Reynolds' Number. The various factors entering into the Reynolds number are qualitatively related as follows (R-4):

Consider a fluid without appreciable viscosity flowing in a closed conduit. This fluid would have only inertia forces called into play by its movement. These inertia forces are proportional to the density.

Density and Viscosity. In any real fluid there will necessarily be some effect of viscous forces. The degree to which they modify the motion of the liquid depends on their relative magnitude in comparison with the inertia forces. Hence with fluid flow the significant property of the fluid is not simply its viscosity U nor its density W, but the ratio of viscosity to density, or

$$U/W = \underline{KV} \quad \ldots\ldots\ldots\ldots\ldots [13]$$

which is known as its "kinematic" viscosity.

In flow through a nozzle or orifice the viscous forces increase in direct proportion to (V) the velocity of flow of the fluid. The kinetic energy, or inertia forces, increase with (V^2) the square of the velocity. Hence the effect of viscosity is relatively less at high than at low velocities; thus, increasing the rate of discharge has qualitatively the same effect on the discharge coefficient as decreasing the kinematic viscosity.

Diameter. Consider another nozzle or orifice, geometrically similar to the one just discussed, i.e., of the same shape but smaller. Reducing the size of the differential producer without reducing the velocity of flow through it increases the viscous shear, or the transverse gradient of velocity across the flow, and thus increases the viscous forces which are proportional to the shear.

The influence of viscosity is therefore greater on small than on large differential producers. Thus, reducing the diameter has qualitatively the same effect on the coefficient as increasing the kinematic viscosity of the fluid.

Combination as Reynolds' Number. Since the foregoing conclusions are relative to the effect of the kinematic viscosity, the coefficient for differential producers of any one shape can be expressed by some function of the ratio

$$R = VD/\underline{KV} = VDW/U \quad \ldots\ldots\ldots\ldots [14]$$

in which V is the mean axial velocity through the nozzle or orifice —computed from the known diameter D and the actual discharge-rate A, $\underline{KV}$ is the kinematic viscosity, U is the viscosity, W is the density of the fluid, and R is known as the Reynolds number.

The important conclusion is that the general relation holds for differential producers of any one shape, and that the coefficient C is a function of the Reynolds number, or

$$C = f(R) \quad \ldots\ldots\ldots\ldots\ldots [15]$$

The relation between fluid-meter coefficients and the Reynolds number has been shown in Fig. 1, which gives an airplane view of the general situation. Although the central pitot tube would seem to make an ideal oil meter in the streamline-flow régime still the maximum velocity is liable not to be in the center of the pipe, and such a meter would not be accurate. In the usual turbulent régime the venturi-tube coefficients are seen to be quite close together while there is considerably more variation for the orifices.

Derivation From Dimensional Homogeneity. For geometrically similar differential producers in closed conduits, through which fluids flow, having no appreciable effects of compressibility or change of density, the discharge coefficient for a fluid will depend in part upon the rate of discharge. The actually effective properties of the fluid are its density and viscosity.

The relation connecting the discharge coefficient C with the velocity of flow V, the density W, and the viscosity U of the fluid, and the diameter D of the differential producer may be expressed by the equation

$$C = f(V, D, W, U) \quad \ldots\ldots\ldots\ldots\ldots [16]$$

The value of the function f must be determined by actual experiment; however, it is certain that some relation exists between this function f and these variables, since all of the variables which have actually been found by experiment to affect the coefficient are included, and nothing else of importance enters in usual practical metering of fluids.

For a differential producer of another shape the function f would obviously be changed somewhat, but where the form is limited to one shape there can only be one relation of the function f to the other factors. Since the value of the function f, which takes complete account of the shape, must be determined by actual experiment, it makes the relation, which is entirely independent of the shape, perfectly general for any type of differential producer.

The essence of the law of dimensional homogeneity is that the dimensions on both sides of an equation must be equal as well as the numerical values of the various quantities.

The discharge coefficient C is dimensionless, for it is a pure ratio of the actual value of the rate of flow V as compared with the theoretical value determined from the fundamental relation without any coefficient.

Hence each combined term of the second member of the equation must be dimensionless, regardless of the unknown form of the function f itself. While none of the quantities V, D, W, or U themselves satisfy this requirement, still it was

early found by Reynolds that the combination R (the Reynolds number) does.

$$R = VDW/U \quad [14]$$

as can be seen by equating the units:

$$\frac{\frac{\text{ft.}}{\text{sec.}} \times \text{ft.} \times \frac{\text{lb.}}{\text{ft.}^3}}{\frac{\text{lb.}}{\text{sec.} \times \text{ft.}}} = 1 \quad [14a]$$

This is the only combination that does satisfy this requirement,[7] and consequently the coefficient must be a function of the Reynolds number only,[8]

or

$$C = F(VDW/U) = F(R) \quad [15]$$

The value of F must be found from experiment, but there is now only one independent variable R instead of four: V, D, W, and U.

Expansion Factor and Acoustic-Velocity Ratio

The correction for expansion must also be dimensionless as it is also a true ratio or coefficient, and consequently must depend upon terms which are themselves dimensionless. It is obvious that the ratio of the differential pressure to the inlet pressure, P/P_1, would be dimensionless, and the expansion correction should be some function of P/P_1. Where the acoustic velocity (velocity of sound) is significant, it will be seen that the ratio of the velocity of flow and the acoustic velocity, V/V_a, is also dimensionless, and consequently this acoustic-velocity ratio should be useful for this type of flow. The usual adiabatic-expansion correction is seen to be composed purely of dimensionless ratios—for the pressures and the specific heats, which correspond with the relations obtained if the velocities of flow and sound are expressed mathematically.

It should be realized, however, that there is a fundamental difference in the performance of fluid meters measuring liquid and those measuring gaseous fluids, in that liquids do not change density appreciably with the usual small differential pressures occurring in fluid meters, whereas gaseous fluids may, particularly where the initial pressure is low.

A further limitation for gaseous-fluid meters is that the absolute pressure at the greatest flow constriction should exceed approximately one-half of the inlet pressure, as otherwise the rate of flow will depend entirely upon the inlet pressure, being practically directly proportional to it, this being generally known as Napier's relation. At this critical pressure the velocity of the fluid at the point of maximum constriction becomes equal to the velocity of sound, and the acoustic-velocity ratio thus becomes unity. Under these conditions the usual differential pressure relation becomes useless for metering purposes.

Derivation of Expansion Factor "Y" From Dimensional Homogeneity. For use with any shape of differential producer, the known upstream conditions are used. The dimensionless expansion factor Y is seen below to be a function of the dimensionless ratio V_1/V_a, where V_1 is the mean velocity of the fluid at the inlet and V_a is the acoustic velocity (velocity of sound) under the inlet conditions.

The expansion factor Y is dimensionless and consequently must depend upon that dimensionless ratio which contains all of the pertinent variables which affect the expansion of the fluid. The velocity V_1 of the fluid and the acoustic velocity V_a together contain all of these pertinent variables. The ratio of the two velocities V_1 and V_a is necessarily dimensionless, being known as the acoustic-velocity ratio V_1/V_a, which forms the basis upon which the expansion factor Y must necessarily depend.

$$V_1 = \frac{Q_1}{A_1} = \frac{CY}{\sqrt{(D_1/D_2)^4 - 1}} \sqrt{2P/W_1} \quad [17]$$

and

$$V_a = \sqrt{\frac{nP_1}{W_1}} \quad [18]$$

so that for a "flat" coefficient C,

$$V_1/V_a = k_1 Y \sqrt{\frac{x}{n}} = \text{the acoustic-velocity ratio} \quad [19]$$

where

$$x = P/P_1 \quad [20]$$

is the ratio of differential to upstream pressure, n = the ratio of the specific heats, and k_1 a constant.

However, for each value of the "acoustic-velocity ratio" there is a corresponding value of x/n, namely, the "acoustic ratio" R_{a1}, or

$$Y = f_1(V_1/V_a) = f_2(Y^2 x/n) = f(x/n) \quad [21]$$

by dimensional homogeneity.

The acoustic ratio x/n for the inlet conditions will be used since it involves only known quantities and is directly proportional to the square of the acoustic-velocity ratio at ordinary low differentials.

If the acoustic-velocity ratio is taken at the "throat" conditions—Section 2—for a venturi tube (or the corresponding *vena contracta* conditions for an orifice), the corresponding factor (R-8) for the acoustic ratio R_{a2} under these conditions, assuming adiabatic expansion, is

$$\frac{(P_1/P_2)^{\frac{n-1}{n}} - 1}{(n-1)\left[1 - (D_2/D_1)^4 (P_2/P_1)^{2/n}\right]} = R_{a2} \quad [22]$$

The relation of the acoustic ratios[9] at the inlet and throat, R_{a2}/R_{a1}, is shown on Fig. 5. Although (for infinite D_1) $R_{a2} = 0.5$ when $V_2 = V_a$, regardless of the value of the specific-heat

[7] See Appendix No. 3 for conventional derivation, equating dimensional exponents.

[8] Where V, D, W, and U and the geometrical shape of the differential producer are the only pertinent variables.

[9] If the ratio R_{a1}/R_{a2} was a single-valued function in spite of variation of the specific-heat ratio n, exact correspondence could theoretically exist between the coefficient and R_{a2} at the throat, R_{a1} at the inlet, or R_a at any section between them. (J. L. Hodgson states that the coefficient does not depend exactly upon the acoustic-velocity ratio taken at either the throat or upstream, but instead at some point between them.)

The choice of "corresponding" functions, such as the acoustic ratio x/n for the acoustic-velocity ratio V_2/V_{a2}, requires careful judgment since the simple mathematical laws underlying "correspondences" are not adequately covered in conventional engineering courses, and since such a change of function is usually made, in fact, to avoid involved mathematical expressions which are difficult to visualize.

The basis for primary "correspondence" is (as its name implies) that for each point on one surface there will be a single corresponding point on another surface. This essential requirement was not met by Odquist, who attempted to substitute the simple pressure ratio P_2/P_1 (a function of pressure only) for the acoustic-velocity ratio V/V_a (a function of both the pressure ratio P_2/P_1 and the specific-heat ratio n). Odquist also substituted the acoustic velocity V_a for the fluid velocity in the Reynolds number, and thus obtained (R-9) a physically meaningless ratio $V_a DW/U$, which with P_2/P_1 becomes absurd when the fluid is a liquid.

It is desirable to use only basic dimensionless ratios which have clear physical meaning; the Reynolds number VDW/U (or its equivalent QW/DU) which defines the degree of "turbulence" in the fluid, and the acoustic-velocity ratio V/V_a (or its equivalent, and more convenient, acoustic ratio x/n). Both of the ratios ($V_1 D_1 W_1/U_1$ and x/n) finally chosen by the author retain their physical meaning for either liquid or gaseous fluids.

ratio, still it will be found that Y then varies considerably, as below:

$$Y = \sqrt{\frac{n}{n-1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} \frac{1-\frac{2}{n-1}}{1-\left(\frac{2}{n-1}\right)^{\frac{n}{n-1}}}} \quad \ldots [23]$$

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}} \text{ for } D_2/D_1 = 0 \ldots\ldots [24]$$

assuming adiabatic expansion as above.

In fact, the expansion factor Y varies less with changes of the specific heat ratio when R_{a1} is used, although R_{a1} does not have a constant value when the velocity of sound is reached at the throat.

With the velocity of sound at the throat, for $D_2/D_1 = 0$,

with	n =	1.135	1.408	1.67
	x =	0.422	0.473	0.513
	x/n =	0.372	0.336	0.307

Obviously the only basis which would give a constant value of the expansion correction at the acoustic velocity, regardless of

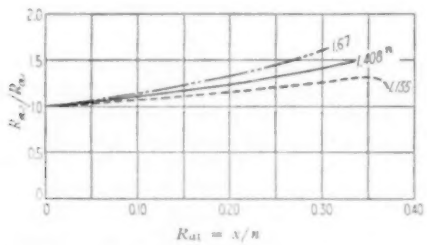


FIG. 5 RELATION OF ACOUSTIC RATIOS AT INLET AND THROAT CONDITIONS FOR VENTURI TUBES

changes of the specific-heat ratio, would be the complete adiabatic expression itself, and this would hardly be convenient or practicable to use for usual commercial metering. Besides, this expression is based on the assumption of purely adiabatic flow and so is not suitable for general use. The "throttling" orifice with distance pressure connections, which measures only the "loss," may possibly do violence to this assumption.

In view of the foregoing, the use of the known conditions upstream has been adopted in these notes for the acoustic-ratio basis for the expansion correction factor. Incidentally, it would be difficult in practice to measure the temperature at the throat.

REYNOLDS' NUMBERS, SIMILARITY, AND COEFFICIENTS

The following example illustrates the relation which exists between Reynolds' numbers, similarity of flow and structure, and coefficients. It may also assist in defining these terms as used in this work.

The flow Q through a closed conduit is divided into two streams, see Fig. 6, one of which flows through a long capillary tube M in which the resistance to flow is due to viscous forces, and the other stream through a nozzle N or orifice in which the resistance to flow is due to the inertia of the fluid.

The pressure differential P, due to viscous friction, across the ends of the tube is proportional to the viscosity and to the rate of flow Q_m through the tube. This same pressure differential P, due to inertia forces, across the ends of the passage containing the nozzle is proportional to the density and to the square of the rate of flow Q_n through the nozzle.

Relation of Similarity to Reynolds' Numbers. Similarity of flow requires that the streamlines, which necessarily follow the flow distribution, be similar in shape and that the velocities or ratios of flows be in the same proportion at all places in the conduit. The ratio of the two branch flows, Q_m/Q_n, through the capillary tube and the nozzle may be found from the relation that the same differential pressure exists across the ends of both branches.

For a capillary tube J diameters long, the differential pressure P depends upon the flow Q_m, the viscosity U, and the capillary-tube diameter D_m as follows:

$$P_m = J\,k\,U\,V_m/D_m \ldots\ldots [25]$$

and since $Q_m = k_1\,V_m D_m^2$,

$$P_m = K_m\,U\,Q_m/D_m^3 \ldots\ldots [26]$$

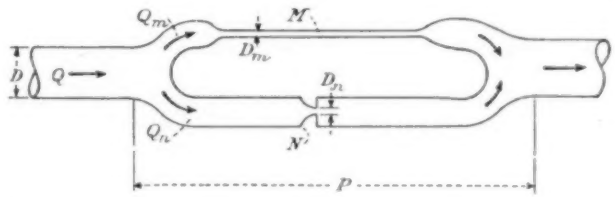


FIG. 6 DIVIDED CONDUIT WITH VISCOUS AND INERTIA FORCES ACTING IN PARALLEL

For the nozzle, the differential pressure depends upon the flow Q_n, the density W, and the nozzle diameter D_n as follows:

$$P_n = K\,W\,V_n^2 \ldots\ldots [27]$$

and since $Q_n = K_2\,V_n D_n^2$,

$$P_n = K_n W\,Q_n^2/D_n^4 \ldots\ldots [28]$$

Since these two differential pressures are equal, or $P_m = P_n = P$, the ratio S of flows Q_m/Q is defined as follows:

$$Q = Q_m + Q_n \ldots\ldots [29]$$

Letting $D_m = mD$ and $D_n = nD$, $Q_m = SQ$ and $Q_n = (1-S)Q$.

$$P_m = K_m\,U\,S\,Q\,/m^3D^3 \ldots\ldots [30]$$

and

$$P_n = K_n W\,(1-S)^2\,Q^2/n^4D^4 \ldots\ldots [31]$$

Then equating P_m and P_n,

$$\frac{(1-S)^2}{S} = \frac{K_m n^4\,UD^4Q}{K_n m^3\,WD^3Q^2} = \frac{K_m n^4\,UD}{K_n m^3\,WQ} \ldots\ldots [32]$$

and since $Q = k_2\,V\,D^2$,

$$\frac{(1-S)^2}{S} = b\,\frac{U}{WVD} = b/R \ldots\ldots [33]$$

where $R = VDW/U$ or the Reynolds number.

$$S^2 - (-2 - b/R)\,S + 1 = 0 \ldots\ldots [34]$$

$$S = \frac{(2 + b/R) - \sqrt{(2 + b/R)^2 - 4}}{2} \ldots\ldots [35]$$

following the usual quadratic solution. However, the value of S must be from zero to unity and so the positive value of the radical is omitted as having no physical meaning.

$$S = f_1(R) \quad \text{and} \quad (1-S) = f_2(R) \ldots\ldots [36]$$

Thus the ratio of flows, i.e., "flow similarity," is seen to depend only on the Reynolds number. Consequently for closed conduits, gravity has no effect upon the flow distribution; another proof of this is that it is well established that the direction of the axis of the flow in a nozzle with respect to the vertical has no effect upon the performance of the nozzle.

The direction of the flow lines themselves at the ends of the branch passages is also seen to be entirely dependent upon the Reynolds number.

Relation of Coefficient to Reynolds' Numbers. The differential pressure across the nozzle may be used to determine the rate of flow.

$$P = K_n W Q_n^2/D^4 = K_n W (1 - S)^2 Q^2/n^4 D^4 \ldots\ldots [37]$$

$$P = K_n W Q^2 f_2 (R)/n^4 D^4 \ldots\ldots\ldots\ldots [38]$$

and taking the square root of both sides,

$$Q = KD^2 \sqrt{P/W} \ldots\ldots\ldots\ldots [39]$$

where

$$K = f (R) \ldots\ldots\ldots\ldots [40]$$

or the coefficient depends upon the Reynolds number.

This is recognized as the fundamental equation for fluid meters in which the differential is created by the rate of flow. This formula holds rigorously for purely turbulent (non-viscous) flow, for mixed viscous and turbulent flow, and also for purely viscous flow—in other words, the formula is perfectly general.

While the mathematical treatment of the above derivation is based upon having purely viscous flow in the capillary tube and purely turbulent flow in the nozzle (or orifice), still it will be observed that the nozzle and the capillary tube are in parallel, and in case the flow in the capillary tube should have a slight inertia effect, or the flow through the nozzle should have a slight viscous effect, there would not be any change in the form of the equations as both types of flow are already in parallel and only the constants would be slightly affected. It may be noted that in ordinary turbulent flow in a closed conduit with ordinary smooth pipes, the turbulence is not complete but increases with the Reynolds number, the quantity of flow at the walls of the pipe in viscous motion decreasing and the quantity of flow in the center of the pipe in turbulent motion increasing correspondingly, both being in parallel.

The same fundamental equations involving terms of inertia—proportional to the density and the square of velocity, and terms of viscous resistance—proportional to the viscosity and the first power of the velocity, have been used heretofore to derive the relation between the type of flow and the Reynolds number. Although the equations and their results were undoubtedly correct, the physical significance of the various factors has not been entirely convincing. The present derivation from purely physical considerations constitutes a complete confirmation of the fact that both the similarity of flow and the coefficients depend only upon the Reynolds number, which number was derived originally from the simple law of dimensional homogeneity, i.e., the units on both sides of an equation must be the same. This latter purely mathematical derivation has also failed to satisfy completely the need for attaching a physical meaning to the Reynolds number. Consequently the present physical derivation may be of interest as presenting fairly and completely the relationship actually known from experiment to exist between Reynolds' numbers, similarity, and coefficients.

Notes on Fundamental Considerations for Flow Similarity

The rate of quantity discharge can depend only upon the shape and size of the differential producer, the various significant properties of the fluid, and the initial physical conditions, such as initial pressure and temperature besides the differential.

At the upstream connection the pertinent conditions may be exactly known.

At the downstream connection, with an actual fluid and shape where there is friction loss—involving viscosity, or a contraction—the amount of which depends upon the shape and also upon all of the factors affecting similarity of flow, the conditions of the fluid at the downstream connection will not in general be exactly determinable, and consequently may not be used as a rigorous basis for the fundamental relation between the quantity-rate of flow and the other pertinent variables.

Similarity of Flow. For identical coefficients, there must be exact geometrical similarity of shape of the differential producer and exact similarity of flow in the differential producer.

With a viscous liquid, the flow similarity will be determined by the shape and the Reynolds number. However, this relation itself may not be the only significant factor since temperature effects between the upstream and downstream connections are introduced by friction, which will have an effect upon the velocity distribution at the downstream connection. Also, in any actual differential producer there will be some transfer of temperature along the walls of the differential producer. The temperature-distribution effects are generally negligible for usual metering differential producers, although they may become appreciable in dealing with friction loss in pipe lines and with centrifugal pumps. The additional properties of the liquid which might be expected to become significant are the specific heat and conductivity and their relation to the temperature (as well as the temperature distribution and the temperature of the liquid), also the conductivity of the walls and the temperature distribution along these walls.

With a gaseous fluid, which is expansible and also viscous, the flow distribution simultaneously depends upon the Reynolds number and the ratio of the velocity of the fluid to its acoustic velocity. There will also be secondary effects of the temperature upon the significant properties of the fluid which may again be expected to depend upon the specific heats, conductivity, density, viscosity, and their relation to the temperatures and pressures of the fluid. There also will be some effect, as with a liquid, of the temperature distribution along the walls of the differential producer.

The acoustic velocity depends primarily upon the density, ratio of specific heats, and absolute pressure of the fluid, and it may vary somewhat from the upstream to the downstream connection. However, it may be exactly determined at the upstream connection, while at the downstream connection it may be somewhat uncertain due to the effect of the shape of the differential producer and the various secondary factors listed above. At this downstream connection, for example, there is likely to be an abnormal distribution of temperature, and hence density, and consequently also of the acoustic velocity, across the section.

Exact flow similarity will be approached most closely when a minimum effect is produced by other dimensionless ratios (besides Reynolds' numbers and acoustic-velocity ratios) which may involve secondary properties of the fluids and also temperature considerations. For the closest approach to the Reynolds number and acoustic-velocity ratio it seems preferable to use the exactly determinable conditions at the upstream connection, in view of the foregoing discussion.

Practical Metering Calculations. Coefficients are used in order to obtain a clear—usually a magnified—picture of some phenomenon. As many factors as possible, which have their relations definitely known, make up the body of the formula, while the remaining theoretically indeterminate values make up the coefficient. The conventional coefficient C for the venturi tube, for instance, expresses chiefly the excess pressure due to frictional resistance when the departure of the coefficient from its theoretically approximate limit of unity is considered, as above stated. For the liquid equation for square-edged orifices, the conventional contraction coefficient C indicates approximately

the value of the ratio of the area of the *vena contracta* to the orifice area. The usual point of interest, however, is to note the amount of departure from the usual value of approximately 0.6 due to various factors.

It is thus seen that the most physically significant dimensional value for a venturi tube is the throat area, or for a square-edged orifice is the corresponding area of the *vena contracta*. Further, in exceptional cases the inlet diameter may be so large that the velocity of approach has no significance. In this case the coefficient obviously depends upon the Reynolds number, using the throat diameter instead of the inlet diameter.

The Reynolds number chiefly determines the flow similarity, especially the coefficient expressing the amount of contraction at the *vena contracta* for the orifice, although for gases the latter will be increased slightly by the effect of expansion of the fluid at that section, with the acoustic-velocity ratio. This expansion-increased *vena contracta* is the proper constriction for use in the criterion for determining where the upstream pressure alone controls the rate of flow, i.e., where Napier's law comes into action. This point is the most important factor in controlling the type of flow, and consequently in actual use of the ratio of the velocity of the fluid to the acoustic velocity, the clearest physical picture is obtained from the use of the velocity of the fluid where the section of maximum constriction occurs. This section is at the *vena contracta*.

For very high speeds exceeding that of sound, it has been shown that stationary pressure waves occur (R-10).

IV—OTHER FACTORS AFFECTING THE COEFFICIENT

The foregoing proofs of the applicability of the Reynolds number for determining coefficients for geometrically similar differential producers in closed channels show conclusively that the method is rigorously correct when the flow depends exclusively, as it actually does in general, upon the factors included in the number itself. It must be understood, however, that there are certain definitely predictable conditions which introduce additional factors. One familiar case which already has been discussed is the correction for compressibility of gaseous fluids. There are a number of other conditions which may affect the flow distribution and consequently the coefficient. The existence of these other conditions in no way undermines the correctness of this method of "flow similarity" as the use of Reynolds' numbers is coming to be generally known, but merely requires that the additional physical conditions be taken into consideration, either by correcting for them or by eliminating the troublesome conditions.

Although some of these conditions have been discussed above, it is well to group them together here so that the limitations may be more readily recognized as they occur and to lend confidence to the use of coefficients determined from Reynolds' numbers without correction when these disturbing conditions do not occur. The method of similarity, i.e., Reynolds' numbers, is the only rigorously correct general method for correlating coefficients for fluid meters, and it may be used with entire confidence.

The flow distribution and coefficient depend basically upon the Reynolds number. Even when the coefficient is "flat"—i.e., such changes of diameter of pipe, the velocity, density, and viscosity of the fluid as occur in ordinary fluid measurement, with a particular meter, have no appreciable effect—still the coefficient actually depends on the Reynolds number, and its value sets the limits over which the coefficient may be safely used as being "flat."

Summary of Other Factors Affecting the Coefficient

Other factors affecting the coefficient are the following:

1 Gravity, only in "open channels"
2 Plastic fluids, not having true viscosity
3 Shape, size, and roughness, defined for true geometrical similarity
4 Discontinuity of flow, absolute pressure too low
5 Non-homogeneous fluids
6 Gaseous fluids, effects of expansion, and Napier's law
7 Dirty fluids, effects in practice
8 Non-uniform rate of flow.

Gravity. The differential producer and the conduit in which it is located must be closed—not have any surface exposed to the flowing stream which is not guided by fixed solid walls; i.e., there must not be any "free" surface or boundary between liquid and gaseous fluid. Where a "free" surface, due to gravity, occurs, there may be wave effects produced which modify the flow distribution and affect the coefficient. In this case an additional factor is seen to affect the coefficient, this factor being the ratio of the wave height to the square of the fluid velocity divided by twice the acceleration of gravity.

For similarity of flow through geometrically similar open channels, there necessarily will be a simple proportion between the wave height and any dimension of the channel such as its depth (R-11), and consequently this known fixed linear dimension may be used instead of a variable fluid dimension. This additional term (Froude's ratio) for similarity may thus be written:

$$2gL'/V^2 \qquad [41]$$

where the following notation is used:

Symbol	Dimension	Unit	Quantity
g	L/T^2	Ft. per sec.^{2}	Acceleration of gravity
L'	L	Ft.	Depth or other dimension of the channel
V	L/T	Ft. per sec.	Velocity of flow in the channel

A simple additional test to apply to determine whether gravity affects the coefficient is to consider whether a change of position of the differential producer relative to the vertical will cause a difference in the flow-distribution. Gravity not only has magnitude but also has direction, being a true vector quantity; consequently, if it has any effect, this effect must be influenced by direction. It is established universally that the position of differential producers in closed conduits with reference to the vertical has no effect whatsoever upon the coefficient and consequently "gravity" cannot enter as an additional limitation to the Reynolds number so long as the fluid fills the conduit and is of substantially homogeneous density throughout.

Plastic Fluids. There are some liquids which are only semi-fluids, being known as plastic or non-Newtonian fluids. Plastic solids do not flow until a finite force acts, but although plastic liquids flow slowly under infinitely small forces, still the rate of shear is not exactly proportional to the pressure. The absolute viscosity as used in the Reynolds number is not strictly applicable, and consequently the measurement of a plastic liquid with a differential type of meter involves additional difficulties. A familiar example of a plastic solid is a "grease" which may be formed from a truly fluid oil by adding a sufficient quantity of soap, while a plastic liquid would be formed by adding a much smaller amount of soap to the oil so it still flows slowly with any finite differential but does not have a constant "viscosity."

Size, Shape, and Roughness. For strict geometrical similarity, the linear dimensions of the roughnesses of the surface must be proportional to the linear dimensions of the differential producers. For the commercial differential producer of usual size, such as a well-shaped venturi tube or nozzle or a sharp-edged

orifice with flange connections, the effect of having the same absolute roughness, instead of roughness proportional to the diameter, will only produce a slight effect upon the coefficient. For a venturi tube or venturi nozzle, the coefficient for a small diameter will be only slightly less than for a large-diameter meter since the additional roughness increases the differential slightly; while for a smaller sharp-edged orifice with flange connections the coefficient will rise only slightly since the additional roughness will slightly decrease the inward flow of the fluid which causes the *vena contracta*.

Strict similarity of shape of the differential producer also requires that the shape, including roughness, of the conduit which contains the differential producer must be similar and must have a proportional length of straight pipe upstream and downstream from the differential producer to insure that the coefficients obtained by calibration will apply to the meter in location for actual metering. The same number of diameters of straight pipe preceding the meter is more important generally than the length of straight pipe downstream of the meter, since the velocity distribution at the inlet ordinarily affects the differential pressure considerably.

Where the upstream shape is such that the fluid rotates in the pipe as it approaches a venturi tube, producing helical flow, the coefficient is experimentally found to decrease by an amount which depends upon the "pitch" of the helix and the amount of constriction of the tube. The loss across the ends of the tube correspondingly increases when the pitch of the helix decreases. The coefficient for swirling flow is nearly constant, being practically independent of the rate of flow.

For an orifice, the coefficient rises for small angles of twist and decreases for angles larger than approximately 45 deg.[9a]

Straightening vanes may be used to eliminate errors due to swirling flow.

The design of pressure holes also should be similar for various sizes.

Slight constrictions in penstock lines are frequently calibrated and used as differential producers. The change of velocity from the inlet to the throat is generally quite small and the friction effect of the walls of the penstock is relatively large so that the coefficient is likely to fall at low rates of flow much more than with a standard-design venturi tube. This constriction is usually located just above the scroll case containing the turbine gates, which are opened more or less as required by operating needs, and thus there is a veritable change of the shape of the differential producer; the velocity distribution across the constriction and hence the coefficient being related definitely to the gate opening. However, the total variation of head is usually small, and consequently the calibration will apply closely under the usual conditions of operation. Two methods of rating these penstock meters in commercial use are the "salt-velocity method" of Prof. C. M. Allen, and the "pressure-velocity method" of N. R. Gibson.

Discontinuity and Absolute Pressure. The fluid may fail to fill the conduit and discontinuous flow may develop when the centrifugal forces of the fluid streams exceed the force of the absolute pressure which tends to hold the fluid stream in contact with the solid guiding walls of the conduit or differential producer. Liquids are generally more sensitive to this limitation than gaseous fluids, because in a liquid there may be formed bubbles of gas, which had been absorbed and held in solution by the liquid under a higher pressure but which can separate out with a decrease in pressure. Also liquids which are near their boiling points, such as hot water from a condenser, cracked petroleum distillate, or liquid anhydrous ammonia which has not been supercooled, may flash into vapor and errors of considerable magnitude be produced. It should be observed, however, that only about one-half of the error due to change of density, for a given lowering of pressure, will occur with a differential-pressure meter as with a displacement-type meter, since for the differential meter the error in the rate varies as the square root of the density, and for the displacement meter varies directly as the density.

Where there is no dissolved gas in the liquid, the boiling point of the liquid determines the safe minimum pressure—obviously the liquid will vaporize if the vapor pressure is more than the pressure in the differential producer.

When a liquid leaves a condenser without being supercooled, i.e., being at its boiling point, and is stored at the same pressure in a receiver, a slight drop in pressure will cause some of the liquid to flash into vapor. In order to keep all of the fluid in the liquid state, it is necessary to supercool the liquid somewhat, using additional cooling surface, or to put it under higher pressure than exists at the surface in the receiver; in other words, to keep the outflow pipe, at all points, and differential producer below the hydraulic gradient, based on the liquid level in the receiver.

This rule also holds good for liquids containing dissolved gases, it being important not to have the absolute pressure drop below that at which equilibrium has been attained.

There is somewhat less tendency for the coefficient to be affected in the case of a venturi tube or nozzle having properly streamlined, solid guiding walls forming the constriction in which there is a minimum tendency for a free contraction to form at the throat, than for the square-edged orifice in which the constriction is formed by the inertia of the fluid where there is the likelihood of pockets of gas or "broomy" discharge occurring on the downstream side of the orifice. Also, with an orifice there is more probability of erroneous coefficients being caused by deposits of material, heavier or lighter than the fluid, in front of the orifice.

Further, where wet steam is being metered there is a greater tendency for the entrained liquid to be thrown out of the steam and to accumulate in front of the orifice, it being generally necessary to provide a small drain hole at the bottom of an orifice when installed in a horizontal main.

Where a "free" surface exists as with the flow over weirs, there are also the effects of still other factors besides those included in the Reynolds number such as surface tension, adhesion, the viscosity drag of the gas (usually air) around the falling stream, the effects of the gas entrainment, and finally the effect of "back water," which may cause the weir to be "submerged," also the nappe of the weir can cling or be free depending upon certain shape factors, involving the admission of air beneath the nappe, which may not be included in the design of the portion of the weir which is actually in the flowing stream.

Non-Homogeneous Fluids. Where wet steam is being accelerated in passing through a differential producer there is what is known as the "braking" effect of the drops of water in the body of the steam. The steam tends to speed up when it reaches the constriction, but the drops of water tend to continue in motion at the much lower velocity of the steam in the main pipe at the inlet. A similar braking action would be expected in measuring air conveying powdered coal or sawdust, even if the mean density could be satisfactorily obtained. An additional effect of "gravity" on the flow distribution occurs when the density of the fluid is not constant over a cross-section of the differential producer or conduit. For example, in a pipe line conveying hot oil under streamline-flow conditions, the central body of the stream will be hotter than the oil near the walls of the pipe and consequently will be lighter. This will cause it to rise or "nose up" from the center of the pipe and seriously affect

[9a] Personal communication from Prof. W. S. Pardoe, University of Pennsylvania Hydraulic Laboratory.

any differential meter. Also where flowing oil is being either heated or cooled while the flow is streamline, there will be abnormal velocity distribution since the velocity distribution across a section is highly sensitive to any change in the viscosity of the oil. These changes are quite severe for small changes of temperature with heavy petroleum oils. Furthermore, the effect of upstream fittings is more persistent when abnormal flow exists in the streamline régime, since a highly irregular stream of very hot oil sometimes will flow for many pipe diameters through a "bed" or channel of much colder, sticky, semi-solid oil.

Gaseous Fluids. For orifices, especially, there are sometimes considerable changes in the flow distribution with the changes of pressure that necessarily occur. There are also corresponding changes of fluid density which tend to enlarge the *vena contracta* and thus increase the coefficient. This effect must be determined experimentally, and is in addition to the effect which occurs in nozzles which can be computed from thermodynamic considerations. It is also entirely distinct from compressional-wave phenomena which only occur at high expansion ratios where the velocity approaches that of sound.

When the velocity through the constriction of the pressure-differential producer approaches the velocity of sound, the pressure downstream cannot "back up" against the rapidly moving stream and so the capacity of the differential producer depends only upon the upstream pressure, naturally rendering useless the usual formula, for the capacity, which depends normally upon the differential pressure. This critical point is usually where the pressure, at the point of maximum constriction, is about half the absolute inlet pressure. Interesting compressional waves occur downstream from these conditions. A familiar hydraulic analogy is the performance of the usual weir which has the discharge dependent upon the differential when the backwater is high enough so that the weir is "drowned," while the discharge depends only upon upstream head (pressure) when the backwater is low enough so that it cannot work back against the freely falling stream—this latter condition of flow being termed "shooting flow."

Fundamentally, gases act differently from liquids in that they change density appreciably with small pressure differences such as ordinarily occur in metering differential producers.

Dirty Fluids. For a limited number of fluids, such as blast-furnace gas, which are extremely liable to have deposits separate out, the sharp-edged orifice may be more easily cleaned and will not tend to be affected as much as the shaped nozzle. This is chiefly due to the fact that the maximum pressure drop does not occur at the plane of the orifice but at the *vena contracta*, and consequently the formation of deposits due to this cause at the orifice is minimized. Further, the venturi tube, which is the usual form of nozzle used for this service, has considerably less loss of head than the orifice producing the same differential, and it is the accepted practice to use a higher differential with the venturi tube and thus obtain the utmost range of measurement. This higher differential, while generally advantageous, increases the tendency for deposits to occur. Commercial designs of venturi tubes have been developed which facilitate cleaning the throat. Also, steam-jacketed venturi tubes are generally used for certain gases where naphthalene deposits would otherwise form; on the other hand, the sharp-edged orifice does not lend itself readily to this remedy. For large-ratio orifices, a very slight deposit affects the accuracy considerably.

Non-Uniform Rate of Flow. The coefficient depends upon an additional factor when the flow is not at a steady rate. Where the flow is pulsating in character, there may be vortex rings formed with sharp-edged orifices and some disturbance of the flow distribution, even with nozzles. This effect is independent of the effect of the average of the differentials being different from the average of the rates, and also of the errors caused by pressure waves occurring in the pressure pipes connecting the differential producer with the register. Furthermore, where the main pipe diameter is small, there will be impingement of the streams from the pressure pipes into the main stream which will seriously disturb the flow distribution, particularly in the streamline régime where the normally streamline flow may become turbulent in consequence of the disturbance at the pressure connections.

V—VENTURI-TUBE ANALYSIS

(This chapter, and the following, are merely illustrative material, offered as being of general interest)

Analysis for Liquids, Using Bernoulli's Theory. For liquids, a simple relation holds between the head H and the velocity V:

$$H = V^2/2g \quad \ldots\ldots [42]$$

$$V = \sqrt{2gH} \quad \ldots\ldots [43]$$

as is familiar to all hydraulic engineers.

Where fluid flows through a frictionless nozzle with no approach velocity, i.e., through an infinitely large inlet,

$$Q = AV \quad \ldots\ldots [44]$$

$$Q = A\sqrt{2gH} \quad \ldots\ldots [45]$$

where all the velocities V over area A must be equal and parallel. With an actual venturi tube in a pipe there will be some approach velocity, some friction from the inlet to throat, and the velocity distribution will not be uniform.

$Q = A_1V_1 = A_2V_2$ where the fluid is not compressible and there are no leaks between the inlet and throat, consequently

$$H = \alpha_2 V_2^2/2g - \alpha_1 V_1^2/2g + bV_1^2/2g$$

where α_1, α_2, and b are functions of the Reynolds number; α_1 and α_2 are factors for non-uniformity of velocity distribution and b depends upon the friction loss and change of turbulence between the inlet and throat.

For large D_2/D_1 ratio meters, α_1, α_2, and b may become relatively large, but for the usual ratios of standard commercial venturi tubes they are quite small. ($R = D_1/D_2$.)

$$\left.\begin{aligned} H &= \frac{\alpha_2 (D_1/D_2)^4 V_1^2 - \alpha_1 V_1^2 + bV_1^2}{2g} \\ &= \frac{(\alpha_2 R^4 - \alpha_1 + b)V_1^2}{2g} \end{aligned}\right\} \quad \ldots\ldots [46]$$

The theoretical head H_t is

$$H_t = (R^4 - 1)V_1^2/2g \quad \ldots\ldots [47]$$

whence

$$Q_1 = A_1V_1 = A_1\sqrt{2gH_t} = CA_1\sqrt{\frac{2gH_1}{R^4 - 1}} \quad \ldots\ldots [48]$$

Also

$$Q_1 = A_1\sqrt{\frac{2gH_1}{\alpha_2 R^4 - \alpha_1 + b}} = CA_1\sqrt{\frac{2gH_1}{R^4 - 1}} \quad \ldots\ldots [49]$$

and

$$C = \sqrt{\frac{R^4 - 1}{\alpha_2 R^4 - \alpha_1 + b}} \quad \ldots\ldots [50][95]$$

[95] Correspondence with Elliot E. Moody (Jun. A.S.M.E., Standard Oil Co., Bayway, N. J.) checks this method.

Referring to the basic formula [10], $Q_1 = kA_1\sqrt{gH_1}$, and the basic venturi formula

$$Q_1 = C\frac{A_1}{\sqrt{R^4-1}}\sqrt{2gH_1} \quad \ldots\ldots [51]$$

it will be found that the coefficient C or its components (α_1, α_2, and b) must be determined from actual experiment. The coefficient C depends on the shape and the Reynolds number, and the ratio $1/\sqrt{R^4-1}$ merely serves to make the coefficient more nearly dependent upon the variations from the ideal case. Consequently it is generally found to be most satisfactory to plot the coefficient C against the Reynolds number instead of attempting to make the above further analysis involving α_1, α_2, and b.

Analysis for Gaseous Fluids, Using St. Venant's Equation. If the fluid be a gas, its velocity at the throat of the venturi tube, due to its expansion, will be somewhat higher than it would be for a liquid. When the pressure drop is large in relation to the inlet pressure, the correction for expansion may be considerable. Assuming adiabatic expansion, i.e., frictionless flow, the gain in kinetic energy equals the potential energy liberated (per pound of fluid), or where $r = A_1/A_2$ and n = specific-heat ratio,

$$V_2^2/2 - V_1^2/2 = \frac{P_1}{W_1}\frac{n}{n-1}\left[1-(P_2/P_1)^{\frac{n-1}{n}}\right] \quad \ldots [52]$$

and for adiabatic expansion

$$W_2/W_1 = (P_2/P_1)^{1/n}, \quad \ldots\ldots [53]$$

and

$$Q_w = A_1V_1W_1 = A_2V_2W_2 \quad \ldots\ldots [54]$$

therefore

$$V_1 = V_2A_2W_2/A_1W_1 = V_2\,(1/r)(P_2/P_1)^{1/n} \quad \ldots\ldots [55]$$

and therefore

$$V_2^2\left[1-\frac{1}{r^2}(P_2/P_1)^{2/n}\right] = 2\,\frac{P_1}{W_1}\frac{n}{n-1}\left[1-(P_2/P_1)^{\frac{n-1}{n}}\right] \quad \ldots\ldots [56]$$

$$Q_w = A_2V_2W_2 = A_2V_2W_1\,(P_2/P_1)^{1/n} \quad \ldots\ldots [57]$$

$$Q_w = CA_2\left[\frac{2P_1W_1\,\frac{n}{n-1}\left(1-(P_2/P_1)^{\frac{n-1}{n}}\right)(P_2/P_1)^{2/n}}{1-\frac{1}{r^2}(P_2/P_1)^{2/n}}\right]^{1/2} \quad \ldots\ldots [58]$$

which is the general discharge formula required, where C is the discharge coefficient as determined by experiment.[10]

[10] It is obvious that if the coefficient C for a venturi tube is sufficiently low, the expansion will not be strictly adiabatic and some correction to the above formula will be necessary to take care of the different law of expansion.

For an orifice, the coefficient C must include both the friction coefficient C_r and the contraction coefficient C_c, also the proper value of the ratio of areas r to be used in this formula must be the ratio of the area of the pipe A_1 to the area at the *vena contracta* A_c, and

$$A_c = C_cA_2 \quad \ldots\ldots [59]$$

This method of treatment theoretically follows the actual physical phenomena most closely. However, it is customary to use the formula derived above, using the known orifice area A_2 in the ratio r. For more detailed discussion, see the section on orifice theory given later.

The coefficient C may include the effects of viscosity.

The actual volume Q_1 passing, measured at the upstream conditions P_1 and T_1 (and W_1), may be obtained from the relation

$$Q_1 = Q_w/W_1 \quad \ldots\ldots [60]$$

Since the full equation is inconvenient in actual use, it may be simplified (R-8) by combining it with the simpler formula obtained above, on the assumption that the fluid is incompressible, i.e., that W is constant.

To derive this formula, the loss of potential energy is equated to the gain of kinetic energy per pound of fluid between A_1 and A_2, thus:

$$\frac{P_1-P_2}{W_1} = \frac{V_2^2-V_1^2}{2} = \frac{V_2^2-\left(\frac{V_2}{r}\right)^2}{2} = \frac{V_2^2}{2}\left(1-\frac{1}{r^2}\right) \quad \ldots [61]$$

Therefore

$$V_2 = \left[(P_1-P_2)\,\frac{2r^2}{W_1(r^2-1)}\right]^{1/2} \quad \ldots\ldots [62]$$

Combining equations

$$Q_w = CA_2\left[(P_1-P_2)\,2W_1\,\frac{r^2}{r^2-1}\right]^{1/2} \quad \ldots\ldots [63]$$

This is the approximate equation corresponding to the adiabatic equation.

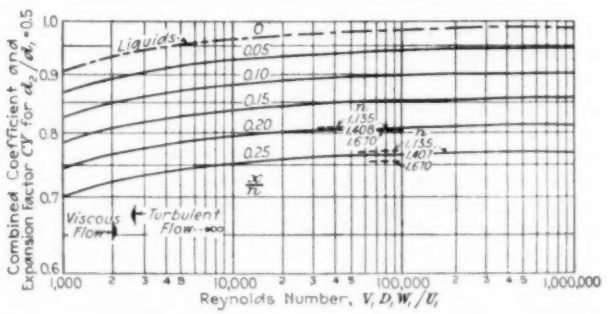


FIG. 7 RELATION BETWEEN COMBINED COEFFICIENT AND EXPANSION FACTOR CY AND REYNOLDS' NUMBER AND ACOUSTIC RATIO FOR A VENTURI TUBE

Combining equations where, as before,

$$P = P_1 - P_2 \text{ and } R = D_1/D_2 = \sqrt{r} \quad \ldots\ldots [64]$$

$$Q_1 = CY_1A_2\sqrt{\frac{R^4}{R^4-1}}\sqrt{2P/W_1} \quad \ldots\ldots [65]$$

or

$$Q_1 = CY_1\frac{1}{\sqrt{R^4-1}}A_1\sqrt{2P/W_1} \quad \ldots\ldots [66]$$

which are seen to be merely forms of the basic hydraulic equation with an added correction Y_1 for expansion.

$$Y_1 = \left[\frac{\left(\frac{1}{1-P_2/P_1}\right)\left(\frac{n}{n-1}\right)\left[1-(P_2/P_1)^{\frac{n-1}{n}}\right](P_2/P_1)^{2/n}}{\frac{R^4-(P_2/P_1)^{2/n}}{R^4-1}}\right]^{1/2} \quad \ldots\ldots [67]$$

If the throat conditions were used as a basis for the expansion factor Y_2, there would be considerably less correction for any given value of P_2/P_1. In other words, a higher differential can be used at a certain pressure with less expansion correction

than where the inlet conditions are used. However, it will be found in practice that the upstream conditions are actually more satisfactory for the expansion correction basis, since the upstream pressure P_1, which also affects the meter reading, is usually more nearly constant, and the combined effect of the density change and expansion correction is a minimum.

The combined coefficient and expansion correction factor must be a function both of the acoustic-velocity ratio and the Reynolds number when considered rigorously (see Fig. 7).[11]

VI—ORIFICE ANALYSIS

Orifice Considered as a Nozzle. As with the venturi tube, the theory for the orifice may be analyzed further to make clear the nature of the relations involved, and especially to permit the use of data obtained on one fluid for metering another fluid.

Expansion of the fluid affects the size of the *vena contracta* as well as the density of the fluid at the *vena contracta*. If the orifice is considered as a nozzle having its upstream and *vena contracta* corresponding to the inlet and throat connections, it is possible to obtain interesting information.

This method of consideration is correct for orifices having either "throat" connections, as above, or "flange" connections—which have nearly identical relations existing between quantity-rate of flow and differential pressure.[12]

The basic formula for gaseous fluids is

$$Q_1 = KA_1\sqrt{P/W_1} \quad \ldots\ldots\ldots\ldots\ldots [68]$$

as before, where K is the experimentally determined factor. This factor K includes the friction coefficient, contraction coefficient, and the combined correction for velocity of approach, including change of density from the inlet to the throat. (This factor depends both upon the Reynolds number and the acoustic-velocity ratio.)

Considering the orifice as a nozzle, it will be seen that the area of the *vena contracta* corresponds truly to the area of the throat at the nozzle. Experiments on orifices in glass pipes have established definitely the size of the *vena contracta* and consequently it has been possible (R-12*a*) to obtain the friction coefficient C_v as with a nozzle. This value has been found to be quite close to 0.97 for the usual sizes of orifice and ratios of orifice size to pipe size. The coefficient of contraction C_c has the expected value of approximately 0.61, depending somewhat upon the orifice ratio and the method of making pressure connections. The correction for compressibility and also the velocity of approach are exactly the same as would be expected with an ordinary nozzle. For purposes of determining this expansion Y_1 for the usual orifice ratios, it is merely necessary to divide the diameter of the orifice by $\sqrt{1.59}$ (= 1.26) in determining the ratio of diameters of the pipe and *vena contracta* for use in obtaining the adiabatic expansion factor Y_c.

The usual nozzle formula and coefficient may be used in determining the value of the compressibility factor and the *vena contracta*, assuming the friction coefficient C_v to be 0.97.

The "orifice" formula thus becomes

$$Q_1 = C_vC_cY_c\frac{1}{\sqrt{R_c^4 - 1}}A_1\sqrt{2gH_1} \quad \ldots\ldots\ldots [69]$$

the notation used being as follows:

Symbol	Dimension	Unit	Quantity
Q_1	L^3/T	cu. ft. per sec.	volume-discharge rate
C_v	....	0.97 (approx.)	friction coefficient
C_c	....	.....	contraction coefficient
Y_c	....	.....	adiabatic-expansion correction, (based on P_1) and diameter at *vena contracta*
$R_c = \frac{D_1}{D_c}$ ..... [70]			$\frac{\text{diameter at upstream (pipe)}}{\text{diameter at } vena\ contracta}$
D_c	L	ft.	diameter at *vena contracta* $D_c = \frac{D_2}{1.26}$ for computing Y_c
D_1	L	ft.	diameter of pipe
D_2	L	ft.	diameter of orifice
A_1	L^2	sq. ft.	area at upstream (pipe)
g	L/T^2	ft. per sec.^{2}	acceleration of gravity (32.174)
H_1	L	ft.	differential head of fluid itself at density W_1
W_1	M/L^3	lb. per cu. ft.	density at upstream conditions.

Consider an orifice having d_2/d_1 = nearly 0, measuring air. Then, for

P_2/P_1 =	1.0	0.90	0.80	0.70	0.60
x =	0.0	0.1	0.2	0.3	0.4
x/n =	0.0	0.071	0.142	0.213	0.284
$C_vC_cY_c$ =	0.597	0.584	0.569	0.552	0.533
$C_vC_cY_c/0.597$ =	1.000	0.978	0.953	0.924	0.893
Y_c =	1.000	0.944	0.887	0.825	0.758
C_c =	0.616	0.638	0.661	0.690	0.725
$C_c/0.616$ =	1.000	1.036	1.073	1.121	1.178

The increase of the *vena contracta* with the pressure differential is quite clear.

Consider another orifice where d_2/d_1 = 0.55 and d_c/d_1 = 0.55/1.26 = 0.437. Then, for

P_2/P_1 =	1.0	0.95	0.90	0.85	0.80
x =	0.0	0.05	0.10	0.15	0.20
x/n =	0.0	0.0355	0.071	0.107	0.142
$C_vC_cY_c$ =	0.608	0.601	0.593	0.585	0.576
$C_vC_cY_c/0.608$ =	1.000	0.988	0.975	0.962	0.948
Y_c =	1.000	0.9725	0.944	0.9135	0.883
C_c =	0.627	0.638	0.648	0.660	0.672
$C_c/0.627$ =	1.000	1.017	1.033	1.053	1.072

The increase of the *vena contracta* coefficient C_c for a gas referred to the corresponding coefficient for a liquid (same as gas at low differentials) 0.616 and 0.627, above, is seen to be substantially the same for both orifice ratios. Incidentally, the contraction coefficient is also a function of the Reynolds number and the acoustic ratio, and so is affected by a change in the ratio of specific heats as well as the pressure ratio.

Fig. 8 shows the results of this analysis graphically.

Vena Contracta and Reynolds' Number. Fig. 9 shows the combined effects of viscosity and expansion on the combined orifice coefficient and expansion factor. Although the expansion correction has been extrapolated equally at small and large Reynolds' numbers, the lines are dotted for streamline flow with

---

[11] These curves are based on the approximation that the corrections for viscosity and expansibility may be applied separately without appreciable error. Actually there will be a slight additional effect of expansion on the coefficient itself.

[12] However, this method of analysis is not so useful for orifices having "pipe" connections, 2½ pipe diameters upstream and 8 diameters downstream, since other factors affect the differential than in the case of a nozzle, especially the loss due to pipe friction and recovery of head following Borda's relation.

When "flange connections" are used having the orifice pressure connections close to the orifice itself, the pressures will correspond very closely with those at one pipe diameter upstream and at the *vena contracta*, and consequently the pressure corrections must be the same as though the pressure connections were as the pressures indicate, i.e., practically the same as with the "throat" connections.

For another type of connection ("pipe connections") sometimes used, which has the pressure connections 2½ pipe diameters upstream and 8 pipe diameters downstream, the use of the conditions at the upstream pressure connections is essential since the conditions at the downstream connection are practically indeterminate. In this case the pressure differential does not correspond with that at the point of maximum constriction, but is merely that due to an obstruction, and dissipation of energy occurs in which the expansion is not adiabatic, nor strictly anything else, as there is considerable effect of pipe friction included.

gases on account of the lack of direct data, also the increase of the *vena contracta* due to viscosity may have an effect upon its increase due to gaseous expansion.

The fluid in the center of the pipe passes through the orifice with considerable longitudinal acceleration, but without transverse deflection. However, the fluid near the wall of the pipe must be deflected radially inward toward the center to pass through the orifice opening, and in doing so acquires transverse momentum. Consequently, in the plane of the orifice itself the fluid near the edge of the orifice is moving toward the center as well as longitudinally at an angle Z which depends primarily upon the velocity distribution in the pipe for any given orifice ratio, i.e., the ratio of the velocity of the "obstructed" flow near the wall of the pipe to that in the center.

At large Reynolds' numbers with smooth pipe, the velocity

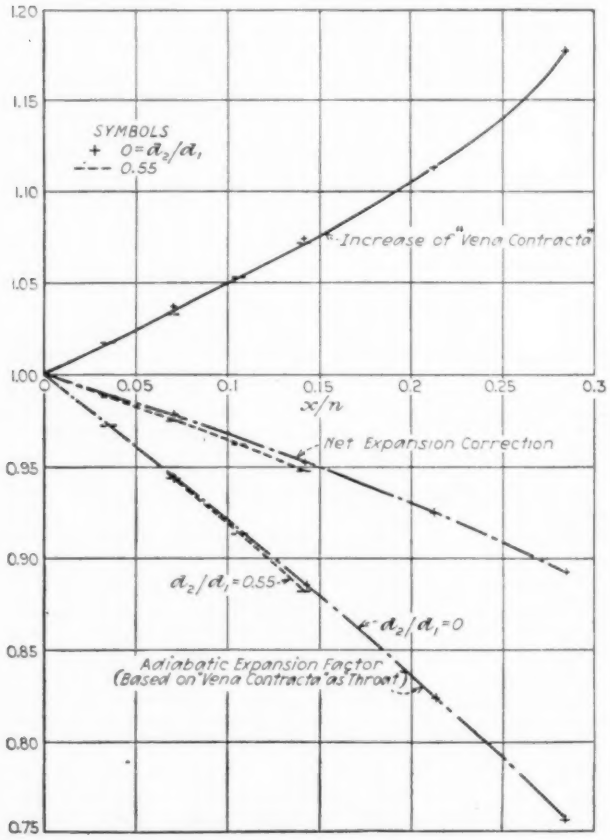


FIG. 8 VARIATION OF EXPANSION OF "VENA CONTRACTA," CONTRACTION COEFFICIENT C_c, AND ADIABATIC EXPANSION FACTOR Y_c FOR ACOUSTIC RATIO x/n FOR ORIFICES
(Data from U. S. Bur. of Stds. R.P.49.)

of the fluid near the walls of the pipe approaches the central velocity more nearly than at low Reynolds' number, the ratio of mean velocity to central velocity approaching a steady value of about 0.82 (see Appendix No. 3 on the "Flow of Fluids in Pipes"). (This is shown by the solid lines in Fig. 3.) The accompanying increased inward momentum causes the angle Z to be larger at high Reynolds' numbers than at low. With streamline flow the viscous friction of the fluid near the upstream face of the orifice itself tends to decrease the inward momentum and consequently the angle Z in addition to the above effect, until finally, at very low Reynolds' numbers the angle Z is negligibly small.

This type of flow at very low Reynolds' numbers is shown by the dot-dash lines in Fig. 3. For both high and low Reynolds' numbers the fluid between the stream and the pipe wall is practically stationary. The streamlines are *not* symmetrical on the upstream and downstream sides of the orifice, as is sometimes fallaciously assumed by mathematicians in theoretical analyses.

Downstream from the orifice, the stream contracts, when the angle Z is appreciable, until the section of maximum constriction or *vena contracta* is reached. There the velocity is nearly uni-

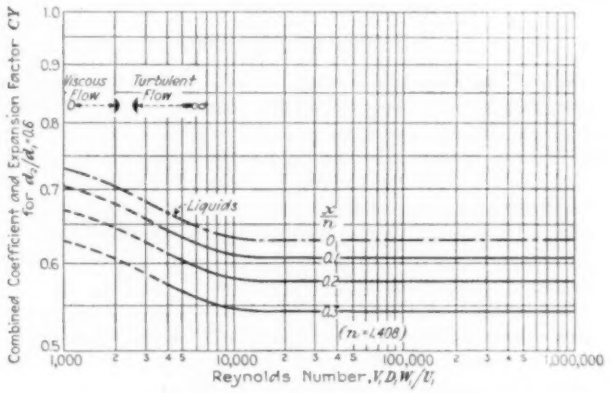


FIG. 9 RELATION BETWEEN COMBINED COEFFICIENT AND EXPANSION FACTOR CY AND REYNOLDS' NUMBER AND ACOUSTIC RATIO FOR AN ORIFICE

form and parallel and the ideal simple relation (that force equals mass times acceleration) is practically realized. The fluid between the stream and the wall of the pipe is in the "lee" of the orifice, and the only motion there is due to the relatively slight viscous drag of the stream. The motion of this fluid is generally sufficiently slow so that only slight effects occur due to its momentum.

The fluid in the stream at the downstream edge of the orifice at the plane of the orifice has practically the same pressure as at the *vena contracta*. However, the pressure in the center is in excess of that at the edge and, by the law of conservation of energy, the velocity in the center is less than that at the outside. It is this radial difference of pressure, and consequently density with gaseous fluids, which causes the *vena contracta* to increase for gases with the decrease of the downstream pressure (due to an increase of differential). With some shapes of differential producers, a serious deviation from "square-rootness" may occur with expansible fluids.

After the *vena contracta* is passed, the pressure increases and the stream fills the pipe as in the recovery cone of a venturi tube, but much less efficiently since more energy is lost due to eddy formation. The viscous drag of the quiet fluid in the "lee" of the orifice slows down the edge of the stream, turning it outward, and causing it to form the recovery "cone" as above.

The basic relations which govern the flow lines in the stream along the pipe are as follows:

1 The longitudinal forces, which are tangential with respect to the longitudinal profile of the boundary of the stream, depend primarily upon the acceleration from one section to another further downstream. This is primarily a function of the change of area of the sections.

2 The radial forces, with respect to the longitudinal profile of the boundary of the stream, depend upon the centrifugal law, which is that the radial force is proportional to the square of the velocity divided by the radius of the longitudinal profile of the boundary of the stream.

In other words, the longitudinal forces vary directly with the rate of change of the velocity with respect to time (depending

primarily upon the cross-sectional areas of the stream), while the radial forces vary directly with the square of the velocity divided by the radius of the longitudinal profile of the stream. Where the effects of viscosity are negligible, the ratio of these forces, and consequently the flow lines at the *vena contracta*, become highly stable, giving a constant or "flat" coefficient of contraction.

The overpowering effect of the large longitudinal acceleration generally wipes out the transverse momentum and thus produces streamline flow at the throat or *vena contracta*. The streamlines in the contracting section from the inlet to the *vena contracta* are very well defined for the usual ratios of orifice to pipe diameters, and the coefficient is stably defined by a single-valued function for the usual case where the approach is a straight

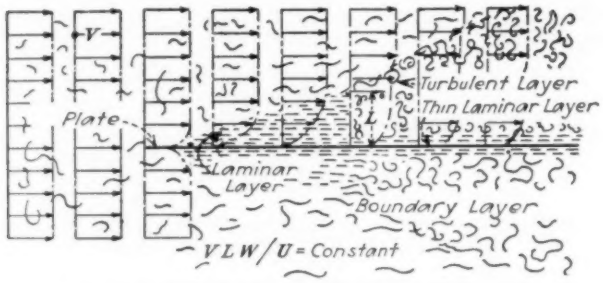


FIG. 10 TYPICAL BOUNDARY-LAYER PHENOMENA

length of smooth circular pipe with the velocity distribution and turbulence (flow similarity) definitely related to the Reynolds number.

Visual observation of flow of fluid in the glass pipes, using particles, dye streams, and "clouds" of dye, shows clearly that the flow near an orifice goes forward and inward at all points upstream of the orifice. This is contrary to the common assumption that there is a fixed eddy in the corner of the pipe and orifice just upstream of the orifice. It is surprising to notice how extremely slow and generally indeterminate the flow is in the "lee" of the orifice at moderate Reynolds' numbers for the observer who seeks order in physical phenomena.

The use of "streamlines" may seem a misnomer for describing highly turbulent flow through an orifice. For practical convenience, "dynamic streamlines" may be defined by stipulating that there must be zero net exchange of transverse momentum across such a line. So defined, dynamic streamlines may be useful in making the velocity-energy analysis necessary for the rational design of draft tubes, spillways, etc., in which the relation between quantity discharge and pressure differential must be estimated accurately. Here again, the best results may generally be obtained by taking full advantage of simple graphical methods, as opposed to complicated mathematical formulas.

The contraction for a parallel slot in an infinite plane has been theoretically determined for a non-viscous fluid. However, the mathematics is prohibitive in even the simple case of a circular orifice in an infinitely large thin plate. Consequently, since the mathematical difficulty of formal hydrodynamics is so great that even such simple physical cases are impracticable of solution, hydraulic engineers generally find it most profitable to make use of only enough of the theoretical mathematical analysis to obtain a clear picture of the physical phenomena involved, and then obtain the necessary information on the laws of flow from actual experiments and visual observation.

Coefficient for Low Pressure Ratios. When P_2/P_1 is low, say, below 0.25, the ratio P_1/Q_1 becomes constant (R-8), as in Napier's law; that is, the mass discharge becomes proportional to the upstream pressure as in the case of a shaped nozzle. If this constant is assumed to hold right down to the limiting value $P_1/P_2 = 0$, one may obtain, as follows, an equation which enables the curve relating the coefficient and expansion factor CY with P_2/P_1 to be extended down to zero P_2/P_1:

If

$$Q_w = P_1/E \quad \ldots\ldots [71]$$

where E is the experimentally determined constant value as above, then, using the basic hydraulic formula

$$Q_w = P_1/E = CYM\sqrt{2\,(P_1 - P_2)\,W_1} \quad \ldots\ldots [72]$$

or

$$CY = \frac{P_1}{EM\sqrt{2(P_1 - P_2)W_1}} = \frac{1}{EM\sqrt{2}} \frac{1}{\sqrt{\left(\frac{P_1}{P_1} - \frac{P_2}{P_1}\right)\frac{W_1}{P_1}}} \quad \ldots [73]$$

whence

$$CY = \frac{1}{EM\sqrt{2}} \frac{1}{\left(1 - \frac{P_2}{P_1}\right)\sqrt{1/R'T_1}} \quad \ldots\ldots [74]$$

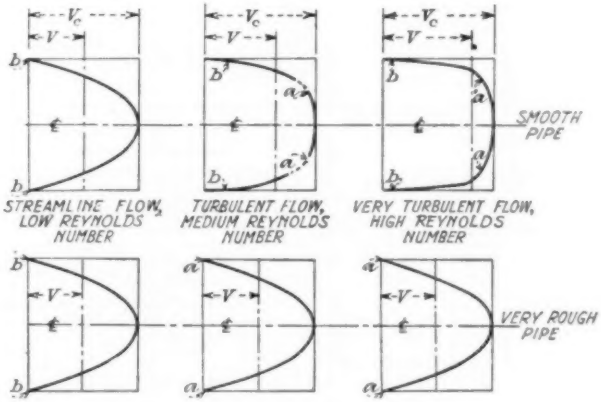


FIG. 11 DIAGRAM OF VELOCITY DISTRIBUTION FOR FLOW IN STRAIGHT CIRCULAR PIPES

where R' and T_1 are from the basic gas density relation $W_1 = P_1/R'T_1$.

$$CY = \frac{1}{EM}\sqrt{\frac{R'T_1}{2}} \frac{1}{\sqrt{1 - \frac{P_2}{P_1}}} \quad \ldots\ldots [75]$$

and in the limit where $P_2 = 0$,

$$CY = \frac{1}{EM}\sqrt{\frac{R'T_1}{2}} \quad \ldots\ldots [76]$$

Appendix No. 1

NOTATION SUMMARY

Symbol	Dimension	Unit	Quantity
...	M	lb.	mass
...	L	ft.	length
...	T	sec.	time
CY =	$f(R, R_a)$	.....	.....
C =	.....	.....	coefficient, correcting for shape and viscosity, as correlated with Reynolds' number R
Y	.....	.....	expansion factor, correlated with acoustic ratio R_a (from acoustic-velocity ratio V/V_a)

$R = VDW/U$	.....		Reynolds' number
$R_a = x/n$	.....		acoustic ratio
V	L/T	ft. per sec.	velocity of fluid
V_a	L/T	ft. per sec.	velocity of sound (acoustic velocity)
D	L	ft.	diameter
W	M/L^3	lb. per cu. ft.	density of fluid
U	M/LT	lb. per ft. sec.	viscosity of fluid
$x = P/P_1$	.....		differential-pressure ratio to upstream pressure
P	M/LT^2	lb. per ft. sec.^{2} (poundals per sq. ft.)	differential pressure

NOTE: Subscript generally as follows:
(1) indicates conditions at upstream connection, point 1
(2) at downstream connection, point 2
(c) at *vena contracta* for an orifice

P_1	M/LT^2	poundals per sq. ft.	absolute pressure at inlet
n	.....	.....	specific-heat ratio

Hydraulic Equation

$$Q_w = CY \frac{A_1}{\sqrt{(D_1/D_2)^4 - 1}} \sqrt{2PW_1} \quad \ldots\ldots [1]$$

Q_w	M/T	lb. per sec.	mass-discharge rate
A_1	L^2	sq. ft.	area of upstream section
$C = f_1(R_1)$	.....		coefficient, correcting for shape and viscosity
$Y = f_2(R_{a1})$	.....		expansion factor, correcting for shape and expansion

Straight-Line Expansion Formula

$y = 1 - Y$ [5]; $y = bx/n$ [6]			"straight-line" expansion correction
b			"straight-line" expansion constant depending upon shape

$H = V^2/2g = P/gW$

H	L	ft.	differential head of fluid acting against gravity
g	L/T^2	ft. per sec.^{2}	acceleration of gravity (32.174)

Basic General Formula

$$Q_1 = kD_1^2 \sqrt{P_1 W_1} \quad \ldots\ldots [9]$$

or

$$Q_1 = kD_1^2 \sqrt{gH_1} \quad \ldots\ldots [10]$$

Q_1	L^3/T	cu. ft. per sec.	volume-discharge rate (at upstream density)

Acoustic-Velocity Formula

$$V_a = \sqrt{nP_1/W_1} \quad \ldots\ldots [18]$$

Open-Channel Factor (Froude's Ratio)

$$2gL'/V^2 \quad \ldots\ldots [41]$$

L'	L	ft.	depth (or other dimension) of channel

Venturi-Meter-Coefficient Formula (From Bernoulli's Theory)

$$C = \sqrt{\frac{R^4 - 1}{\alpha_2 R^4 - \alpha_1 + b}} \quad \ldots\ldots [50]$$

where $R = D_1/D_2$

Venturi-Meter Expansion-Factor Formula (From St. Venant's Theory)

$$Y = \left[\frac{\frac{1}{1 - M} \frac{n}{n - 1} \left(1 - M^{(n-1)/n}\right) M^{2/n}}{\frac{R^4 - M^{2/n}}{R^4 - 1}} \right]^{1/2} \quad \ldots [67]$$

where $M = P_2/P_1$ and n = specific-heat ratio.

Orifice Formula (Based on *Vena Contracta* as Throat of Corresponding Nozzle)

$$Q_1 = C_v C_c Y_c \frac{A_1}{\sqrt{R_c^4 - 1}} \sqrt{2gH} \quad [69]$$

(subscript c indicates conditions at *vena contracta*)

C_v	= 0.97 approximately	friction coefficient
C_c	$= Ac/A_2$	contraction coefficient
Y_c		expansion factor, adiabatic expansion assumed from upstream 1 to *vena contracta* c
R_c	$= D_1/D_c$	

Orifice Formula (From Napier's Law)

$$Q_w = P_1/E = CY k_3 \sqrt{2(P_1 - P_2)W_1} \quad \ldots\ldots [72]$$

$$CY = \frac{1}{Ek_3} \sqrt{\frac{R'T_1}{2}} \quad \ldots\ldots [76]$$

where $P_2 = 0$, (R' from $W = P/R'T_1$)

Pipe-Friction Formula

$$H = K \frac{L}{D} \frac{V^2}{2g} \quad \ldots\ldots [80]$$

where $K = f(VDW/U)$

Appendix No. 2

NOTES ON VISCOSITY, AND AN INTRODUCTION TO THE LAW OF DIMENSIONAL HOMOGENEITY

VISCOSITY is a measure of the internal friction of a fluid. It appears as a force which tends to drag a surface in a moving stream of fluid along with the stream.

The absolute viscosity of a fluid is defined as the tangential dragging force exerted, per unit of area, on one plane by a fluid between two parallel planes (which are immersed in the fluid at a unit distance apart) when one plane moves parallel to the other with unit relative velocity, without rotation.

The dimensions of viscosity U are M/LT or $ML^{-1}T^{-1}$. In c.g.s. units this would be grams per centimeter second (gr./cm. sec.) and in absolute f.p.s. units would be pounds per foot second (lb./ft. sec.).[13]

In order to express this relation by formula, the flow must take place as follows:

1 The fluid must not "slip" at the surfaces, i.e., the fluid adhering to the plane must have zero velocity relative to the surface.

2 All the fluid motion must be streamline only, parallel to the direction of motion of the planes and without any eddying or turbulence.

3 The resistance to flow must be directly proportional to the rate of shear.

4 The resistance to flow must be proportional to the area of the surface.

True fluids meet these conditions at sufficiently low rates of shear. Viscosity is due to the interchange of momentum between molecules passing from one layer of fluid to an adjacent layer moving with a different velocity. The faster-moving molecules upon passing into the slower-moving layer of liquid increase its velocity in the direction of flow, and vice versa, the net tendency being to equalize velocities throughout the fluid, i.e., to resist shear.

This interchange of momentum obviously depends directly upon the difference in the velocities in adjacent layers—the rate of shear—and also with the area of the surface.

Where, as above, the following notation is employed:

Symbol	Dimension	Unit	Quantity
..	M	gram	mass
..	L	cm.	length
..	T	sec.	time
D	L	cm.	distance between two parallel planes
A	L^2	sq. cm.	area of each plane
V	L/T	cm. per sec.	tangential relative velocity
F	ML/T^2	gram-cm. per sec.^{2}	tangential force acting
U	M/LT	gram per cm-sec.	viscosity of the fluid
$F = UAV/D$		or in words,	

[13] The gram and pound are mass dimensions, as indicated by the use of M above.

the tangential force F acting is directly proportional to the viscosity U, area A, and rate of shear V/D.

$$U = FD/AV$$

and as the dimensions on each side of an equation must be equal, the dimensions of viscosity are

$$\left(\frac{ML}{T^2} \times L\right) \Big/ \left(L^2 \times \frac{L}{T}\right) = \frac{ML^2T}{L^3T^2} = \frac{M}{LT} \quad \text{[77]}$$

as above.

When in c.g.s. units, the unit for the viscosity is sometimes known as the "poise" after Poiseuille who first determined the absolute viscosities of various liquids; the corresponding unit in f.p.s. units has similarly been termed the "poisel." However, the fundamental units of gram per cm-sec. and lb. per ft-sec. must be adhered to, in order to prevent confusing engineers who do not have constant use for these values.

It is interesting to note here that the viscosity of water at about 68 deg. fahr. is almost exactly 0.010 gram per cm-sec., and thus the convenient value of this unit, termed the "centipoise," is in common use.

The kinematic viscosity ν is the ratio of the absolute viscosity U to the density, W, or is equal to U/W, where

U	M/LT	gr. per cm-sec.	viscosity
W	M/L^3	gr. per cu. cm.	density
ν	L^2/T	sq. cm. per sec.	kinematic viscosity

since for U/W,

$$\left(\frac{M}{LT}\right) \Big/ \left(\frac{M}{L^3}\right) = \frac{L^2}{T} \text{ [i.e., sq. cm. per sec.]} \quad \text{[78]}$$

The kinematic viscosity may be obtained from the number of seconds that it takes a standard quantity of oil to flow from a standard viscosimeter (such as 60 cu. cm. with the Saybolt Universal viscosimeter) and referring to a conversion graph or table.

Since the density of water at 68 deg. fahr. is close to unity, the kinematic viscosity in c.g.s. units of water is approximately 0.010/-1.00 = 0.010 sq. cm. per sec., and the specific kinematic viscosity based on water at 60 deg. fahr. (ν = 0.0113 sq. cm. per sec.) may be convenient in oil calculations where coefficients, based on water at 60 deg. fahr., are available, this being frequently the case since the specific gravity based on 60 deg. fahr. is in common use, and this places the units on a common basis.

It is fortunate that American petroleum oils generally have an approximately straight-line logarithmic relation between kinematic viscosity and temperature which permits the kinematic viscosity to be determined at any temperature (not too low) closely enough for commercial metering, by merely plotting upon logarithmic paper several kinematic viscosities obtained at several temperatures, say, from the Saybolt Universal viscosimeter, and drawing a line through them. Most oils at low temperatures form semi-solids. This type of oil can best be metered after raising its temperature until it becomes a true fluid, having a determinable viscosity.

Notes on Viscosity

Gases. The viscosity of a gas increases with the absolute temperature, but is practically independent of pressure. This naturally follows from the concepts that viscosity is due to the interchange of energy between molecules, and that the velocity of gas molecules increases with the absolute temperature.

For perfect gases, in which the particles repel each other as the nth power of the absolute temperature, the absolute viscosity U is

$$U = k\left(\frac{n+3}{2n-2}\right) \quad \text{[79]}$$

When $n = 5$, it will be seen that the viscosity is directly proportional to the absolute temperature.

Liquids. It is interesting to note that although the viscosity of gases increases with an increase of absolute temperature, still the viscosities of liquids generally decrease with an increase of absolute temperature.

Oils. For usual petroleum oils there is a slight increase of viscosity at very high pressures, still the viscosity may be considered as being practically independent of pressure, just as the density is considered practically constant—i.e., liquids are generally considered to be incompressible in ordinary hydraulics. The primary reason why the viscosity of oils decreases with an increase of temperature is that, when the temperature increases, the molecules are more widely separated and consequently there is less interchange of momentum between molecules. It is merely a fortunate physical coincidence that permits an approximately straight-line relationship between the logarithm of absolute viscosity (and the kinematic viscosity as well) and the logarithm of the temperature in degrees fahrenheit.

Appendix No. 3

FLOW OF FLUIDS IN PIPES

SINCE the usual fluid meters are installed in circular pipes, which are more or less rough, it is profitable to note the phenomena which ordinarily prevail, i.e., the relation between turbulence (flow similarity), the friction constant, and the Reynolds number. The pressure drop for actual fluids having finite viscosity, along a length of pipe, is a function of the shape factor (including roughness), and the Reynolds number, VDW/U.

The usual formula expressing the relation between pressure loss in a straight circular pipe and the friction factor, is conventionally derived by equating dimensional exponents. The formula is as follows:

$$H = K\frac{L}{D}\frac{V^2}{2g} \quad \text{[80]}$$

where $K = f(VDW/U)$.

Data illustrating the relation between the friction factor K and the Reynolds number (R-13) for smooth pipe appears on the graph, Fig. 1. The effect of roughness is practically negligible in the streamline régime, but has considerable effect in determining the critical velocities and the friction factor K in the turbulent régime. There is a definite but not highly stable régime between the streamline and turbulent which may well be termed the "superturbulent" régime, since the pressure loss varies approximately as the third power of the velocity there, while in the streamline régime it varies as the first (1.0) power of the velocity and in the turbulent régime usually varies as some power between the 1.75 and the 2.0 power of the velocity.

Conventional Derivation of Reynolds' Number

Assuming that the elasticity of the fluid does not appreciably affect the flow and that a steady flow state exists, the independent related quantities for flow through a differential producer of a certain shape must be as follows:

P	M/LT^2	pressure differential
D	L	diameter of main pipe
V	L/T	velocity of flow in pipe
W	M/L^3	density of fluid, and
U	M/LT	viscosity of fluid
Q	L^3/T	quantity-rate of flow, dependent upon D and V.

Consider any term in an expression for pressure differential; its dimensions must be those of pressure differential.

$$P = kV^wD^xW^yU^z \quad \text{[81]}$$

$$ML^{-1}T^{-2} = (LT^{-1})^w L^x (ML^{-3})^y (ML^{-1}T^{-1})^z \quad \text{[82]}$$

$$= M^{(y+z)} L^{(w+x-3y-z)} T^{(-w-z)}$$

Hence

for T, $2 = w + z$ [83]

for M, $1 = y + z$ [84]

for L, $-1 = w + x - 3y - z$ [85]

and therefore

$$w = 2 - z \quad \text{[86]}$$

$$y = 1 - z \quad \text{[87]}$$

$$x = -z \quad \text{[88]}$$

and each term is of the form

$$P = kV^{2-z}D^{-z}W^{1-z}U^z \quad \text{[89]}$$

$$P = kWV^2(VDW/U)^{-z} \text{ and } VDW/U = R \quad \text{[90]}$$

where the Reynolds number is R, however,

$$Q = 0.7854D^2V \quad \text{[91]}$$

and so

$$P = k_1WQ^2/D^4 \quad \text{[92]}$$

where $k_1 = f_1(R)$, and

$$Q = K_1D^2\sqrt{P/W} \quad \text{[93]}$$

where $K_1 = F(R)$.

K_1 and the corresponding values of the Reynolds number R may be determined from experiment. Thus the coefficient for a differential producer in a closed conduit is found to depend only upon the Reynolds number under the conditions stated above.

For one pipe-diameter length of straight pipe

$$P = K_1 W V^2 \quad [94]$$

where $K_1 = f_1(R)$, and for a length L of pipe (L/D pipe diameters long),

$$P = K_2 W V^2 \frac{L}{D} f_1(R) \quad [95]$$

In hydraulic work, this law is ordinarily expressed in head H of the liquid itself, or

$$H = K \frac{V^2 L}{2gD} \quad [96]$$

where $K = f(R)$.

Regarding the negative exponent for Reynolds' number, the departure of the coefficient from its "flat value," which is generally of primary interest, is due fundamentally to viscosity. The inverse of the Reynolds number is directly proportional to the viscosity, and so this inverse is generally more significant physically than the number itself. In the derivation of fluid-flow expressions, it will generally be found that the inverse of the number appears in the final expression. However, in view of its historical background and the familiarity of the Reynolds number, it is retained in this work.

PHYSICAL DESCRIPTION OF FLOW PHENOMENA

Flat Plate Moving Through Still Fluid (see Figs. 10 and 11). The modern idea of flow of fluids in pipes is based upon the recognition of the existence of a "boundary layer" next to any surface immersed in the fluid and is somewhat as follows: For simplicity, let us consider the flow past an infinitely thin flat plate mounted rigidly in a fluid moving uniformly in a direction parallel to the plate. There can be no finite slip at the wall, hence the fluid in immediate contact with the plate has zero velocity. As we pass outward into the body of the fluid perpendicularly to the plate, the velocity increases rapidly, but with an ever-decreasing rate of change of velocity, approaching asymptotically the velocity V of the undisturbed stream. Owing to the fact that the velocity V is approached asymptotically as we move outward from the plate, we cannot set an arbitrary limit to it except by defining its thickness as that distance at which the velocity has a value less than that of the stream ahead of the plate by some small fixed amount.

It is considered that this boundary layer is infinitely thin at the leading edge of the plate. It increases in thickness as we pass toward the rear edge of the plate, according to a definite law which is fairly well established. The flow in this layer is laminar as far as we have yet considered it, while the flow in the stream is turbulent. This turbulent flow in the undisturbed mass of fluid sets up forced vibrations in the laminar layer which, since it becomes thicker as we pass toward the rear of the plate, becomes more and more responsive to these impulses which are due to turbulence in the fluid outside. Finally, at a particular value of Reynolds' number (in which the linear dimension L is the thickness of the layer) the laminar flow in the boundary layer breaks down and becomes turbulent. The vortex sheet sometimes observed at the wall of a ship is one of the more obvious phenomena occurring in the turbulent part of the boundary layer.

As regards the relative turbulence in the boundary layer and outside of it, it is known that in the turbulent portion of the boundary layer, the turbulence is much greater than in the outside fluid. It is undoubtedly true that the turbulence (due to the forced vibration) in the laminar portion of the boundary layer is less than in the fluid outside but there is no definite data at hand on this subject.

Since the boundary layer increases in thickness as we pass from the leading edge, in the case of a plate or from the entrance of a pipe, it is obvious that ultimately a pipe will be entirely filled with the boundary layer.

(In what is said in the last two paragraphs it should be noted that the "turbulence" due to forced vibration in the laminar portion of the boundary layer, is a sort of oscillation and not "eddying" as in what is ordinarily termed turbulence.)

The fact that there is a thin laminar layer at the wall of a pipe is borne out quite clearly from heat-transfer phenomena: in which the resistance to heat transfer decreases markedly as the turbulence in the pipe increases. Roughness, which strongly affects the friction loss in turbulent flow, has no appreciable effect upon purely streamline flow and consequently this laminar layer must be quite thin for the effect of roughness to penetrate into the turbulent flow. Also actual experimental data with a microscopic pitot tube (R-14) showed that streamline flow exists immediately adjacent to the walls of a pipe in the turbulent régime. Fluid flowing in a large conduit at any reasonably high velocity must have turbulent flow existing throughout its body, since turbulence depends fundamentally upon the ratio of transverse to longitudinal momentum. Where the flow of the body of the fluid is parallel to a surface, the transverse momentum is necessarily suppressed at the surface itself, producing a thin lamina of streamline flow at the surface. The fundamental difference between streamline and turbulent flow follows from the fact that there is no transverse momentum, other than on a molecular scale, found in the streamline régime. This fact lends added weight to the modern conception of the existence of the "boundary layer" of laminar flow where a thin sheet of fluid flows in streamlines parallel to the surface.

There is a fundamental difference in the law of flow between the movement of a flat surface through a body of "still" fluid, parallel to the surface, and the movement of relatively unconfined "flowing" fluid past the same surface. The resistance to the motion of a vessel through quiet water is affected appreciably by the viscosity of the water. Tests on very large conduits containing highly turbulent flow (with large Reynolds' numbers) on the other hand, have the resistance follow practically the square of the velocity—in which case viscosity naturally has practically negligible effect. The viscous frictional resistance for a ship is due somewhat to the viscous drag of previously still water in the ship's direction of motion. The large conduit, on the contrary, is practically completely filled with the highly turbulent flow, and the viscous effect of laminar flow at the walls reaches a minimum.

The erroneous concept of how fluid flows in pipes, which was accepted popularly by hydraulic engineers, follows: For turbulent flow, it was assumed that the adhesion of the fluid next to the wall of the pipe broke (R-15), and that there was highly turbulent flow next to the wall of the pipe with substantially streamline flow in the center portion of the pipe. The fact that almost none of the usual closed conduits, with which hydraulic engineers are familiar, are transparent has permitted such a faulty, but apparently reasonable, conception to exist for many years among engineers who are well versed in making precise calculation of the pressure loss for various velocities, which being measurable quantities are consequently observable.

Velocity Distribution (Flow Similarity) and Reynolds' Number. For turbulent flow there is a fictitious "mechanical viscosity" which appears for any fluid in the center portion of a pipe, see Fig. 11. Parabolic velocity distribution exists in the center of the pipe from a to a. This viscosity depends only upon the Reynolds number. It is due to the fact that transverse momentum is transferred uniformly throughout the conduit in highly turbulent flow just as transverse momentum on a molecular scale is transferred uniformly throughout the body of the fluid in the streamline régime with laminar flow, which gives a parabolic velocity distribution from b to b.

[For a parabolic velocity distribution in a circular pipe, the "velocity head"

$$H_{v1} = V^2/g \quad [97]$$

and the mean velocity $V = 0.5\ V_c$ where V_c is the central velocity. With a venturi-tube throat for turbulent flow in the pipe, the velocity distribution is practically constant, and so the velocity head at the throat

$$H_{v2} = V^2/2g \quad [99]$$

and $V = V_c$; for turbulent flow in smooth pipe, $V/V_c = 0.5$ to 0.82; and so the velocity head $H_v = V^2/g$ to approximately $V^2/2g$. See Fig. 1 for the relation of V_1/V_c for smooth pipes data by Stanton.]

The velocity distribution is found experimentally to depend only on the Reynolds number for an extremely rough pipe. Parabolic velocity (a to a) distribution extends entirely to the wall of the pipe, whereas for ordinary relatively smooth pipe there is a boundary effect which extends inwardly from the walls of the pipe (to a) with the streamline laminar flow layer extending inwardly from the walls of the pipe to b, distances which diminish as the turbulence, as measured by the Reynolds number, increases.

The parabolic distribution of flow has become popularly associated with streamline flow whenever the value of the Reynolds number is less than Reynolds' critical-velocity value, which of course is true for smooth cylindrical pipes. However, at a venturi throat, or orifice *vena contracta*, there has been a powerful longitudinal acceleration which has practically wiped out all transverse momentum, thus producing true streamline flow. However, the velocity distribution is practically uniform across the throat or *vena contracta* under these conditions (absolutely *not* parabolic).

It has been observed that, with turbine nozzles, streamline flow

exists at much higher velocities than would be possible for flow in cylindrical pipes. Water issuing at high velocity from fire nozzles or Pelton turbine nozzles has the "glassy" appearance indicative of streamline flow. It has been erroneously assumed that since the flow in such a nozzle or venturi tube is streamline, there must be a parabolic velocity distribution. This conception has led to faulty estimates (R-16) of the value of the coefficient for the venturi tube. This important point is discussed here to emphasize the fact that the Reynolds number does absolutely not indicate the same velocity distribution across the throat of a venturi tube or *vena contracta* as it does in straight circular pipe, which constitutes another reason for using the physically significant Reynolds number with pipe diameter instead of throat diameter.

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Similarity: Limitations in Its Application to Fluid Flow

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A brief resume of the development of the art for the purpose of bringing about a better understanding of the effects of hydraulic similarity as related to the geometrical similarity of the various devices. Geometrical similarity is of great importance, and devices which are geometrically similar will make the operation more uniform. Geometrical similarity is a matter of construction. Hydraulic similarity is a matter of experiment. Large sizes may or may not show the same results as the smaller ones.

THREE stages are apparent in the usual analysis of physical phenomena. The first stage is reached when preliminary observations or discoveries give sufficient data to form a hypothesis and to warrant further investigation. The second stage is reached when the hypothesis has been applied to a number of cases, and the results warrant the formation of a theory or a workable formula. The third stage is when the theory is shown to hold under all known conditions. Then it is proved and becomes a law.

Physical laws are seldom directly workable. They are true only for abstract or ideal conditions. They are limited to the literal fulfilment of the "ideal" requirements, embodied in the statement of the given law. In other words, we have no conception, except in the abstract, of a case where a given "law" could be applied independently without the interference of other similar or dissimilar laws. In reality, physical relations of quantity are subject only to working hypotheses or theories expressed in given formulas or mathematical equations. These formulas usually are made up from corresponding sets of assumptions based upon some visible or apparent influences and expectations.

On the other hand, any set of well-defined assumptions establishes a "mathematical" relation which in the working process of determining numerical quantities has the same status as any other given set of assumptions applicable to the same case. The difference, then, between the two sets is only a mathematical quantity or ratio. By means of this ratio one set may be converted into the other. Either set may be a law or both sets may be only assumptions; the mathematical relation between the two remains the same.

Let us apply this "law" to the case at hand, the flow of fluid through similar pipes and orifices.

Hydraulic-Flow Equations

There has been a good deal of mathematical genius displayed in the development of fluid-flow equations. A multitude of formulas have been advanced, both rational, based on general assumptions, and empirical, based on a limited amount of experimental data. For our demonstration it will be sufficient to take three or four rational forms that have been thus far accepted as working bases.

The first equation was created by Torricelli (1)[2] in 1643 when he discovered that the velocity of a water jet issuing from an opening in the side of a reservoir was "nearly" the same as that which would be acquired by a heavy body falling from the vertical height measured from the surface of the water in the reservoir to the center of the opening where the jet leaves the reservoir.

This discovery gave rise to the equation

$$V = C\sqrt{2gh} \qquad [1]$$

which is based upon the law of falling bodies. If we use the letter p to denote the equivalent drop of pressure, then, since $gh = p/\gamma$ we have from [1],

$$V = C\sqrt{2p/\gamma} \qquad [2]$$

and

$$p = \frac{1}{2C^2} V^2\gamma \qquad [3]$$

As the science of hydrodynamics advanced after Torricelli's time, the early experimenters, like Edme Mariotte (2) and even Sir Isaac Newton (3), could not adequately explain the "contraction" of the jet after it leaves the opening in the reservoir. The first plausible explanation was advanced by the celebrated Danielis Bernoulli (5) in his famous "Hydrodynamica," published in Latin in 1738. Bernoulli discarded Newton's theory of moving and stationary parts in the reservoir or the so-called "waterfall" theory, and showed that the whole mass of the fluid moves toward the opening and that the resulting jet is determined by the state of equilibrium in the nature of that motion. Bernoulli's explanation of that phenomenon is claimed to be one of the most beautiful and the most artistic products of mathematics. Bernoulli's theory has brought to light other forces that come into play for establishing the state of equilibrium in the flowing stream.

It is of interest to note that the early experimenters have limited their activities to sharp-edged orifices and jets issuing from large tanks, and the fact that the orifice coefficient increases at low heads was observed by Bossut (6) before 1775. He explained that fact by the slight increase of contraction at higher heads due to the larger tangential force toward the center.

At the same time other experimenters [Couplet (4) being the first] were wondering how to express the drop of pressure in conduits where the velocity neither increased nor diminished, and how to reconcile that drop of pressure with the law of Torricelli.

It appears that around the year 1775 M. Chézy (7), director of the French school "des Ponts et Chausseés," was in charge of the Ivette canal construction and was endeavoring to obtain a formula for the relation between the slope of an aqueduct, the dimensions of its transverse section, and the amount of water it could supply in a given time.

Chézy assumed that when water moves in a given channel, the accelerating force of the slope is counterbalanced by the

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[1] Vice-President and Consulting Engineer, Republic Flow Meters Co. Mem. A.S.M.E.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

[2] A numeral after a name refers to the index of cited references at the end of the paper.

resistance the water overcomes in sliding along the walls of the channel. Then the resistance must be proportional to the surface in contact. Chézy also assumed that the resistance is proportional to the square of the mean velocity in the channel. With these assumptions he formed an equation of two terms, the first being the product of the slope by the cross-section of the channel (the accelerating force Ah/L); the second, the product of the wetted perimeter by the square of the velocity (the retarding force BV^2).

The assumption was that the term Ah/L must be proportional to BV^2. From his experiments on open channels Chézy obtained values for a coefficient that balanced the two terms. The work is on record in the manuscripts of the school "des Ponts et Chausseés."

The final form of the Chézy formula was

$$V = n\sqrt{rs} \quad \ldots\ldots [4]$$

in which V is the mean velocity of the fluid; r the mean hydraulic radius A/B, or ratio of the area to the wetted perimeter of the conduit; and s the hydraulic slope or the ratio h/L. For circular pipes $r = D/4$, so that

$$V = n\sqrt{\frac{D}{4} \times \frac{h}{L}} \quad \ldots\ldots [5]$$

Some of the writers, like Hamilton Smith (21), continued to make the coefficient n include the value of $\sqrt{2g}$, which they considered constant. This was really a disadvantage because the coefficient had different values for different systems of units. For our purpose we shall adopt the same form but not include the value of $2g$ in the coefficient. Thus

$$V = n\sqrt{\frac{D}{4L}}\sqrt{2gh} \quad \ldots\ldots [6]$$

and again replacing gh by p/γ, we have

$$V = n\sqrt{\frac{D}{4L}}\sqrt{2\frac{p}{\gamma}} \quad \ldots\ldots [7]$$

and

$$p = \left(\frac{1}{2n^2}\right)\left(\frac{4L}{D}\right)V^2\gamma \quad \ldots\ldots [8]$$

Note the similarity of Equations [7] and [8] and [2] and [3]. What does this mean? Is $n\sqrt{D/4L}$ of Equation [7] actually equal to C of Equation [2]? It is if we refer the two to the same experiment.

Let us illustrate by specific examples from the early experimenters. Couplet (4) in 1732, introducing his experiments on the flow of water in pipes, makes the following statement:

> We shall see by following our experiments that a conduit which according to the rules [of Mariotte] would have furnished 61 $^{11}/_{25}$ flowing inches of water, furnished only 2 inches and 63 lines because the conduit was extremely long, even though it was open mouthed as illustrated on the 4 inch pipe, Figure 1. . . . If the pipe were constructed according to the rules [of Mariotte] it would have been five times too small [in diameter] and would have supplied $^1/_{25}$ of 2 flowing inches and 63 lines.

Bossut (6) in 1775, the next experimenter, after giving in detail his and also M. Couplet's data, prepared a long table showing the relation between the delivery through the given length of pipe and the delivery through a short tube of the same diameter. So in both cases the scientists of that time tried to relate the flow in the pipe to the corresponding flow through a short tube or to Torricelli's equation.

In Bossut's table the ratio of pipe flow is actually a value for C in Equation [2]. His value of C for short tubes was approximately 0.815, and the given ratios varied from 3.18 in a short pipe to 28.5 in a long pipe. This ratio in itself has no meaning, but from the value of this ratio a solution may be obtained for n of Equation [7]. I found that in Bossut's experiments on 1-in., $1^1/_2$-in., and 2-in. diameter pipes n varied from a minimum of 11 to a maximum of 13, while in Couplet's experiments the value of n is from 4 to 5 in the 4-in. pipe, 8 to 9 in the 6-in. pipe, 9 to 11 in the 5-in. pipe, and 9 on one 18-in. pipe, and 13 on the other. This inequality caused Chevalier Dubuat (8) to retain only the data on the 5-in. pipe and one of the 18-in. for his demonstration.

What the similarity of the equations really means is this: Given a certain set of experimental data where a pressure difference and a velocity are involved, one may choose to assume conditions leading to Equation [2] and solve for C, or he may choose to assume conditions leading to Equation [7] and solve for n. As soon as a solution is obtained for either C or n, the shift from one set of assumptions to the other is subject only to the mathematical relation of C to n as obtained by reducing Equations [2] and [7] to one common denominator, or by shifting from Torricelli's assumptions to those of Chézy.

Chézy's formula was later modified to shift the coefficient from velocity to friction. The equation for friction drop in pipes then read:

$$f\frac{V^2}{2g} = \frac{h}{L}\frac{D}{4} \quad \ldots\ldots [9]$$

$$V = \sqrt{\frac{D}{4fL}}\sqrt{2gh} \quad \ldots\ldots [10]$$

$$V = \sqrt{\frac{D}{4fL}}\sqrt{2\frac{p}{\gamma}} \quad \ldots\ldots [11]$$

and

$$p = \left(\frac{f}{2}\right)\left(\frac{4L}{D}\right)V^2\gamma \quad \ldots\ldots [12]$$

The similarity of Equations [11] and [12] to [7] and [8] and to [2] and [3] is apparent. From these equations we have the following relations established between the three coefficients, provided we shift from one set of assumptions to the other for the same set of experimental data:

$$C = n\sqrt{\frac{D}{4L}} = \sqrt{\frac{D}{4fL}} \quad \ldots\ldots [13]$$

$$n = C\sqrt{\frac{4L}{D}} = \sqrt{\frac{1}{f}} \quad \ldots\ldots [14]$$

$$f = \frac{1}{C^2}\frac{D}{4L} = \frac{1}{n^2} \quad \ldots\ldots [15]$$

New Factors Appear

A contemporary of Chézy, the military engineer Chevalier Dubuat (8), was the first to observe a variation in the results with different sizes of conduit. Dubuat conducted a large number of laboratory experiments in 1781, 1782, and 1783 on both narrow and large tubes. His conclusions and the corresponding assumptions were more specific than those of Chézy. To quote from his introduction:

> It is evident that when water flows uniformly in any bed the accelerating force that maintains the flow is equal to the sum of the resistances which it forms, either by its own viscosity or by the friction of the bed. This law appeared to me as old as the rivers and it should form the key to hydrodynamics. It presents itself in an illuminating way, and it is in other respects the basic foundation of uniform motion.

Dubuat was also the first to use the term "viscosity" in this respect, and to vary the temperature of the water in his experiments. His empirical equation, however, contained only the elements of the Chézy formula and a logarithmic function to take care of the variation in the size of the tube. He included no function for temperature changes affecting the flow of the fluid.

Professor Gerstner (10), of the Prague University, experimented directly on the effect of temperature on the fluidity of liquids. His results, as published in the year 1800, showed that the temperature has a greater influence on small diameters and low velocities, and there is a remarkable reference in his article to the effect of heat on the flow of blood through the veins and arteries of the human body.

It is characteristic of that epoch that hydraulic investigations were a part and parcel of natural science, and that learned men in the field of physics and physiology took an active interest in hydraulics. In 1808 the physiologist, Thomas Young, M.D. (12), published an article under the title "Hydraulic Investigations Subservient to an Intended Croonian Lecture on the Motion of the Blood." The Croonian lecture by Dr. Young, published in 1809, was of a purely physiological nature, but the hydraulic investigations in 1808 disclosed a very able physicist and excellent mathematician.

Dr. Young, however, did not investigate the effect of temperature. Instead he adopted the formula advanced by Pierre Simon Girard (11), who, following the assumptions of Chézy and Dubuat and the principles advanced by Coulomb (9), concluded that the resistance to the flow of fluid in conduits was composed of two terms, one involving the first power and the other the square of the velocity. The equation, as given by Young, was in the form of

$$h = a\frac{L}{D}V^2 + 2b\frac{L}{D}V \quad \ldots\ldots [16]$$

Dr. Young rearranged the experiments of Bossut, Dubuat, and Gerstner in accordance with his formula (Equation [16]). Supplemented by his own tests he arrived at empirical values for a and b. These values were made functions of diameter in various powers.

Girard, at that time Engineer-in-Chief "des Ponts et Chausseés," was engaged in experimental work to establish the validity of his principle that the resistance to flow was a function of both the first and also the second power of the velocity. In 1813 Girard (13) submitted to the school "des Ponts et Chausseés" a classical piece of work on the subject of the flow of fluids in capillary tubes and the influence of temperature. From these experiments he showed that while in general the resistance to flow was a function of two terms, one involving the first power and the other the square of the velocity, yet in small, narrow tubes the value of the term involving the square of the velocity was insignificant and the function was reduced to the term involving only the first power of the velocity. Girard's general equation for circular pipes was

$$aV + bV^2 = g\frac{D}{4}\frac{h}{L} \quad \ldots\ldots [17]$$

and for narrow tubes it was reduced to

$$aV = g\frac{D}{4}\frac{h}{L} \quad \ldots\ldots [18]$$

Girard's coefficient a varied with the temperature and also with the diameter of the tube. It is noteworthy that his article contained a number of "viscosity" tables for various mixtures at different temperatures.

Following Girard, Navier (14), Poisson (15), and Stokes (17) have advanced very elaborate mathematical derivations showing that Girard's equation was correct for small, narrow tubes. Then Poiseuille's (16) experiments, published in 1842, virtually showed that the coefficient a in Girard's formula (Equation [18]) is proportional to diameter times fluidity. Poiseuille's equation read

$$V = kD^2\frac{h}{L} \quad \ldots\ldots [19]$$

Soon after, in 1854, Marcel Brillouin (18) showed by differentiation similar to Navier and Stokes, with a slight modification in the assumptions, that in small, narrow tubes the equation for velocity should be

$$V = \frac{1}{32}\frac{p}{L}\frac{D^2}{\mu} \quad \ldots\ldots [20]$$

and

$$p = 32\frac{\mu LV}{D^2} \quad \ldots\ldots [21]$$

This apparent theoretical contradiction in the derivations is cited for the purpose of demonstrating the fact that a theoretical equation is merely a filing case or container for an adopted set of assumptions. It is certain that the integral extensions of Navier, Poisson, and Stokes are mathematically correct the same as are those of Brillouin. The difference lies in the respective sets of assumptions. It will be found that the first three scientists following Girard assumed an additional resistance to the flow in proportion to the surface of contact, while the fourth one, following Poiseuille, did not include that resistance in his assumption. Navier expressed himself, and with great certainty, that the velocity in small tubes was independent of the thing called cohesion or viscosity. In his opinion the velocity depended almost entirely upon the adherence that exists between the fluid and the wall of the tube.

The Common Denominator

Coming back to Brillouin's equation, multiply both sides of Equation [20] by $V\gamma$; then

$$V^2\gamma = \frac{V\gamma}{32}\frac{pD^2}{L\mu} \quad \ldots\ldots [22]$$

$$V^2 = \frac{1}{16}\frac{D}{4L}\frac{VD\gamma}{\mu}\frac{2p}{\gamma} \quad \ldots\ldots [23]$$

$$V = \left[\frac{1}{4}\sqrt{\frac{D}{4L}}\sqrt{\frac{VD\gamma}{\mu}}\right]\sqrt{\frac{2p}{\gamma}} \quad \ldots\ldots [24]$$

In a similar manner, Equation [21] will read

$$p = 8\left(\frac{4L}{D}\right)\left(\frac{\mu}{VD\gamma}\right)V^2\gamma \quad \ldots\ldots [25]$$

There are, in the following, two sets of keynote equations placed in consecutive order to more clearly demonstrate their similarity. The common denominator or the meaning of the corresponding sets of assumptions is apparent.

First Set: Velocity Coefficient

$$V = [C]\sqrt{2\,p/\gamma} \quad \ldots\ldots [2]$$

$$V = \left[n\sqrt{\frac{D}{4L}}\right]\sqrt{2\,p/\gamma} \quad \ldots\ldots [7]$$

$$V = \left[1/\sqrt{f}\sqrt{\frac{D}{4L}}\right]\sqrt{2\,p/\gamma} \quad [11]$$

$$V = \left[\frac{1}{4}\sqrt{\frac{VD\gamma}{\mu}}\sqrt{\frac{D}{4L}}\right]\sqrt{2\,p/\gamma} \quad [24]$$

Second Set: Friction Coefficient

$$p = \left[\frac{1}{2C^2}\right]V^2\gamma \quad [3]$$

$$p = \left[\frac{1}{2n^2}\frac{4L}{D}\right]V^2\gamma \quad [8]$$

$$p = \left[\frac{1}{2}f\frac{4L}{D}\right]V^2\gamma \quad [12]$$

$$p = \left[8\frac{\mu}{VD\gamma}\frac{4L}{D}\right]V^2\gamma \quad [25]$$

A glance at the outline leads to the following analysis:

1 Equations [2] and [3] are universal, subject only to the law of falling bodies. All other influences are included in the coefficients.

2 Equations [7] and [8], the same as [11] and [12], are subject to the law of falling bodies and also to the resistance factor which is expressed in pipe diameters. This factor may be expressed as an orifice ratio, or any other restriction in the flowing stream. All other influences are included in the coefficient n when solving for velocity, or f when solving for pressure drop.

3 Equations [24] and [25] are subject to the law of falling bodies modified by the product $VD\gamma/\mu$ and to the resistance factor expressed in pipe diameters. This takes care of all cases below the critical velocity of the flow. The product $VD\gamma/\mu$ replaces the coefficients of the other cases.

4 The equations given can each be solved only for one unknown quantity, which is usually either V or p, or the coefficient of the given equation.

5 The shift from one equation to the other in no way affects the condition of the flow in the pipe. This seemingly very simple axiom is not generally understood.

6 Any given experimental solution for the coefficient in one of the equations invariably offers a solution for the corresponding coefficient in the other equation, when applied to the same conditions of flow.

Reynolds' Number

We are now coming to the most interesting question in our discussion. Since the hydraulic equations, as given in the previous chapter, were all available in various forms before the time of Osborne Reynolds, and since there are no additional equations of any consequence even now, one may ask: (1) What was Osborne Reynolds' discovery? and (2) What is the basis of the modern theory of fluid flow?

It is not difficult to answer the first question. The old Spaniards claimed that Columbus did not discover America. They said that Columbus was going west to reach India, and America being such a large continent, loomed up in his way and he couldn't pass it. The same is perhaps true of Reynolds' discovery. His experiments were made for the single purpose of determining the dividing line between viscous flow, following the law of Poiseuille, and turbulent flow subject to the less certain but available frictional resistance formulas. What actually happened is this: as long as the flow was viscous and laminar, the expression $VD\gamma/\mu$ was a definite entity or a single product. It did not matter how he varied each one separately, it was the product that determined the behavior of the flow according to the law of Poiseuille.

Reynolds then investigated what happened to the product after the critical point was reached. His actual statements, based on observations, were that the line of demarcation was not well defined, and that there was a certain period of unstable condition in which the flow might be either viscous or turbulent. After full turbulence was reached, Reynolds stated, neither the product nor any of the factors seemed to have any influence on the behavior of the flow.

It is more difficult to answer the second question, What is the basis of the modern theory of fluid flow? As the matter stood at the close of the last century, there were two independent formulas for the determination of frictional resistance in pipes. The engineers adopted the modification of the Chézy formula, while the physiologists, and later the chemists, took possession of the Poiseuille equation as derived by Brillouin.

It was known, however, that in the engineer's formula the coefficient f was not a constant. Darcy (19), in 1857, made f a function of the diameter. Weisbach and others made f a function of the square root of the velocity. Fanning (22) computed extensive tables giving numerical values for f varying with both diameter and velocity. Stanton and Pannell (24) were the first to plot experimental data to the parameter of the Reynolds number. This enabled them to represent the two modes of flow on one chart and to draw very useful conclusions.

Reynolds' experiments gave the hypothetical stage of the analysis. Stanton and Pannell's work justified the formation of the theory that hydraulic similarity should influence the coefficient of flow. Now the tendency is to establish this theory in the third stage; that is, to make it a law.

Briefly, the modern theory states that the analysis of dimensional equations makes it certain that the coefficient of friction and the coefficient of discharge for any flow through similar devices must be functions of the Reynolds number $VD\gamma/\mu$ as a single product and not otherwise.

Lord Rayleigh (23) showed the dimensional analysis and Dr. Buckingham (25) developed the π-theorem to prove this theory. In reality the π-theorem or the dimensional analysis is only another way of arriving at the mathematical relation between the respective sets of assumptions. After a study of the references cited I concluded, first, there is no proof, and second, no proof is necessary. If one believes the modern theory, he can believe it on the simple ground that the product is the determining factor in viscous flow, and therefore it should have some effect in the other regions. One may say by looking at the two sets of equations, as previously outlined: "Here it is, simply push a button on C and the Reynolds number comes out."

On the other hand, if one considers the complete analysis of the equations and the assumptions upon which they are based, and if he follows carefully the experiments that brought about this theory and especially those of Osborne Reynolds, he may agree with the author that the reason the product $VD\gamma/\mu$ comes out by pushing a button on C is because it does not belong there any more. The product has been disbanded, or at least may have been disbanded, and each one of the items may or may not have an influence on the behavior of the turbulent flow. The influence of one may or may not be the same as the influence of the other.

The main difficulty lies in the tendency to go so far as to make a law of this hypothesis or assumption. The outline of the equations has shown that there is no proof that the number retains its influence as a product in the case of turbulent flow because the relation is only one of mathematics and not of physics. The fact that Hodgson's (26) points for various fluids lie on the same curve does not prove this relation. The

fact that Stanton and Pannell's points do not all lie on the same curve does not disprove it. The points will fall nearly on the same curve for various fluids because the parameter contains (the) velocity times (the) density or the momentum of the moving particles, and the momentum of a moving particle has the same numerical value for the same differential pressure irrespective of the nature of that particle.

The Reynolds number is a very desirable parameter for recording experimental data on fluid flow. This parameter becomes a necessity when it is desired to plot on one chart part of viscous and part of turbulent flow, and also for the purpose of determining the region of demarcation between the two kinds of flow. It is imperative, however, to come to a complete understanding of the limitations to similarity that are subject to confusion by the application of this parameter.

Perhaps the thought can be expressed by drawing a comparison between "geometrical" similarity of construction and "hydraulic" similarity embodying the effect of geometrically similar construction. I asked Dr. Föttinger, the hydraulic engineer of the Technische Hochschule in Charlottenburg, how close experiments on geometrically similar models hold true on similar ships and airplanes, and he told me that the actual ship acts quite differently from the model. Thus the references made in this respect to hydraulic similarity as applied to construction of models are obviously misleading. The ship or the airplane is expected to perform in a "similar" manner to the model, but the performance is not expected to be exactly the same. That is, the experimental coefficient may or may not be the same in both cases. If it is within a few per cent, the model served the purpose well. In the case of flow measurement, the 1-in. pipe may be considered the model and the 10-in. pipe the final product. The action of both may be similar, but they may or may not be identical. The "experimental coefficient" may be different.

My objection, therefore, is not against the theory itself but against the method of its application. There is an abundance of theory on the subject, but there is a woeful lack of reliable experimental data. If this new theory will be confined to the use of the function as a parameter, made subject to experimental data, there will be no harm done. If, as the tendency grows now, the theory is set in the ranks of a law and its advocates will insist that experiments made on a "model" pipe can be safely applied to all other sizes in the region of turbulent flow, there will be a good deal of harm done because it will reduce the amount of experimental data that would have been otherwise accumulated.

The present argument on similarity may be compared with the case of M. Couplet, the first experimenter on pipe friction, and whose work was published in 1732. M. Couplet was surprised to find that the rule of Mariotte for the flow of water through short nozzles did not apply to the long pipes of the water works at Versailles. It is perhaps just as inadequate to expect that the laws of similarity, based on definite simplified assumptions of perfectly similar conditions, should apply directly to a measuring device in ordinary commercial pipe. Then there is no argument, and we agree that similarity is not fulfilled in the ordinary installations for fluid measurement.

On the other hand, considering the position of Couplet at that time, would he be expected, when additions were to be made to the water works of Versailles, to assign a diameter to the main such as would be derived from Mariotte's law or to order one that his experiments would indicate was adequate for the proper supply? The same should be true to a certain extent with the case at hand. If one finds that the experimental coefficient of the measuring device, constructed according to the present practice of measurement, shows a different value for different sizes of pipes, should he be advised to use the same coefficient for all sizes of pipes on the ground that, if all the installations in the various pipes were perfectly similar, they would all result in the same coefficient?

Here is the difference. The laws of similarity, or the statement that the flow in the pipe *must* be the same for the same value of the Reynolds number, may be physically correct, the same as the law of falling bodies or the relation between pressure and velocity. But in this latter relation when the results of a given test deviate from the law, we say that "there is the experimental coefficient," and we use that coefficient for practical application of the given law. In the case of similarity, as the argument would have it, if we find that the results of a given experiment deviate from the theoretical expectations, we are expected to say that "the reason is that the installation was not strictly similar."

Effect of Pipe Diameter

In 1922 the author (25*b*) presented to the Society the results of experiments on the flow of air through orifice plates in various sizes of pipes. These results showed that the coefficient of discharge of the orifice plate increases for the same orifice ratio as the diameter of the pipe decreases, irrespective of any other changes in the condition of flow. As an explanation of this phenomenon he stated in Par. 92 of the paper that—

The jet issuing from the orifice does not contract to the same extent in the smaller sizes of pipes. This is probably due to the fact that the portion of the flow which is restricted by the orifice has a proportionally lower velocity due to the increased skin friction in smaller sizes of pipes.

Mr. Hodgson, in discussing that paper, stated that "all the tests could have been carried out on one diameter of pipe had the $VD\gamma/\mu$ relation been used. . . . Had the orifices been similar, or had the edges been beveled off on the downstream side so that the ratio of thickness of the edge to diameter of pipe was kept constant, only one set of tests would have been necessary to determine the coefficients of discharge for orifices of any area ratio for any diameter of pipe."

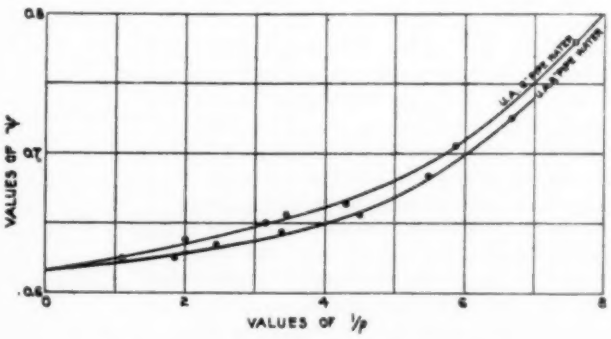


Fig. 1 Flow of Water Through Pipes
(Gaskell, Figs. 5 and 6.)

This criticism led the author to investigate the validity of the statement. From the recorded literature and from his own observations it appeared that the orifice-plate thickness up to $^1/_8$ in. would have no effect on the flow when the spacing was measured from the front face of the plate.

It so happened that, having used for the tests a blower of a given capacity, it was possible to obtain about four times higher velocities in the 3-in. pipe than in the 12-in. with the same ratio orifice. The kinematic viscosity was the same in all the tests. Then, if the supposition were correct, we should have found the same coefficient for the 12-in. pipe at the corre-

spondingly reduced speed as for the 3-in. pipe at the higher speed. Moreover, the increase in velocity in any of the pipes did not seem to change the coefficient; then the same should be

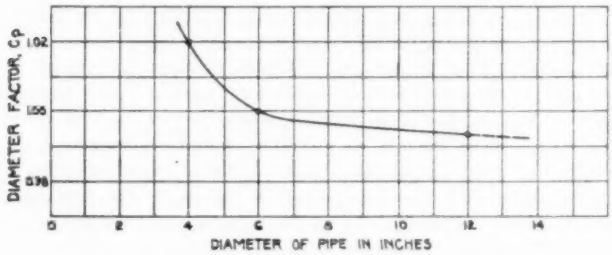


FIG. 2 FLOW OF WATER THROUGH PIPES
(Davis & Jordan, Fig. 10.)

true for the change in diameter. The actual tests, however, have shown the coefficient of the 12-in. pipe to be from 3 to 5 per cent lower than the coefficient of the 3-in. pipe for the corresponding orifice ratios.

The early experimenters have limited their tests to thin-plate orifices discharging from large containers. In such cases the effect of velocity distribution is practically unchangeable. There is, however, a good deal of data by recent experimenters which demonstrate beyond doubt that the orifice coefficient varies for different sizes of pipes at the same numerical value of the Reynolds number. Whatever the reason is for this phenomenon, the fact remains the same and should be complied with in practical measurement.

1 Holbrook Gaskell (23*a*), experimenting on the flow of water through a 6-in. and an 8-in. pipe, showed that the 6-in. pipe had higher coefficients than the 8-in. pipe for the same diameter ratio of the orifice. Fig. 1 is a reproduction of Figs. 5 and 6 in his paper. Mr. Gaskell employed orifice plates designed by John L. Hodgson for his tests.

2 Davis and Jordan (25*a*) have gone a step farther. They have demonstrated the variation of the orifice discharge coefficient with the pipe diameter in their tests on 4-in., 6-in., and 8-in. pipes, and have prepared a curve (Fig. 10) giving the "Diameter Factor" derived. (See Fig. 2.)

3 J. M. Spitzglass (25*b*) in the experiments referred to has shown the variation of the orifice coefficient with the size of pipe for diameters from 3 in. to 12 in. The curves showing that variation are reproduced in Fig. 3.

4 The Verein Deutscher Ingenieure (26*b*), in their publication "Regeln für Leistungsversuche, etc.," show on page 16 a table of orifice coefficients varying with pipe sizes from 100 mm. to 600 mm., based on the results of experiments by Jakob and Kretschmer. A corresponding set of curves, on page 49 of the same pamphlet, show the systematic variation of the coefficient with size of diameter, and also the relative change of the coefficient with the variation of pipe size for given orifice ratios. This curve is reproduced in Fig. 4.

5 Dr. Walter Pflaum (27) experimented on three sizes of pipes, 50-mm., 80-mm., and 100-mm. His Fig. 24 shows the variation of the coefficient with the pipe size. In his case, the sizes being very small and the taps close to the orifice section, each pipe shows a different shape of curve. The difference is very pronounced between the 50-mm. and the 80-mm. pipe. The curve is reproduced in Fig. 5.

6 Dr. R. Witte (28) experimented on the flow of oil, water, air, and steam in sizes of pipes from 50 mm. to 500 mm. On page 6 of his paper, Fig. 21 shows five distinctive curves plotted for five pipe sizes, 50 mm. to 500 mm. The curve is reproduced in Fig. 6. A paragraph in italics on the same page points out the fact that while in the case of his round nozzles the effect of diameter variation was not noticeable, the sharp-edged orifice showed a definite decrease in the value of the coefficient with the increase of the pipe diameter.

7 Jakob and Kretschmer (30) have gone into this subject to a larger extent than any of the other experimenters. They have tested orifices of various ratios in sizes of pipes varying from 125 mm. to 1000 mm., and the pipes were not all of the same absolute wall roughness. Fig. 7 is a reproduction in part of Fig. 72 of their bulletin, showing the variation of the coefficient for the given constant value of the Reynolds number. Other sets are

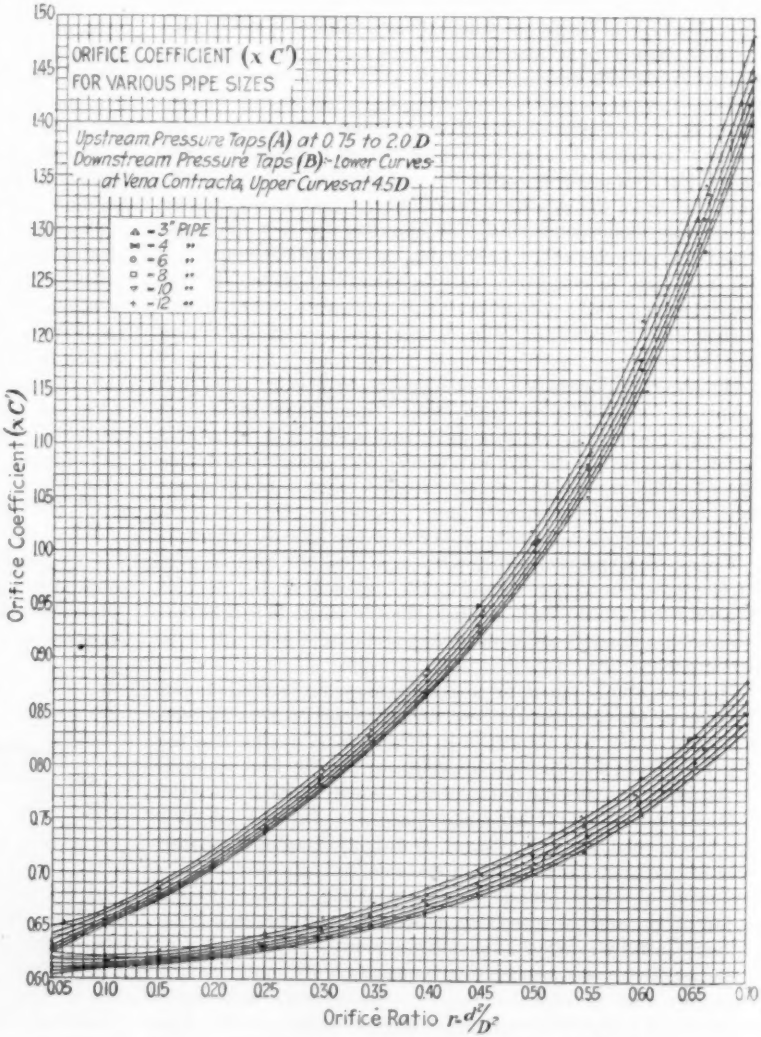


FIG. 3 ORIFICE COEFFICIENTS FOR VARIOUS PIPE SIZES

shown of the same form for different constant values of the Reynolds number.

A very interesting discussion is given by these authors on pages 20 to 27, under the heading "Effect of Pipe Diameter on the

Value of the Orifice Coefficient." The authors give two reasons for the variation, one depending upon the relative smoothness of the pipe, and the other upon the ratio of the velocity in the pipe to the velocity of sound.

The first effect is demonstrated in Fig. 8, which is a reproduction of Fig. 73 in their bulletin. The reasoning of the authors is that when two pipes, one large and the other small, are of the same absolute wall roughness, the relative roughness will be smaller in the larger pipe. The result will be that the larger pipe will have a flatter curve of velocity distribution than the smaller one. Accordingly a larger portion of the fluid will be retained by the orifice wall and diverted toward the center, thus resulting in a larger contraction for the same orifice ratio. The authors show by differentiation that the orifice in the smaller pipe will pass more fluid per unit of area for the same differential pressure than the orifice in the larger pipe.

The second reason given by the authors has reference to the work of Odquist (26*a*), in which it was shown that even in the case of smooth pipes, or such that have the same relative smoothness, there would still be a variation of the coefficient with the variation of the absolute size of pipe. The authors think that their tests have shown that the main cause of the variation is the relative roughness of the pipe; still the equations derived by Odquist show that the coefficient is a function also of the

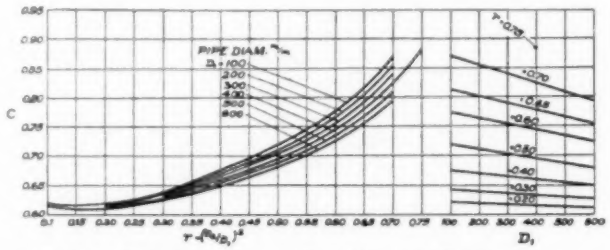


FIG. 4 *C* AS A FUNCTION OF PIPE SIZE AND ORIFICE RATIO
(Jakob & Kretschmer, V.D.I. Bulletin, Fig. 54. *C* is defined from $Q = 0.25\,\pi\,C_3^2 D\sqrt{2gh}$.)

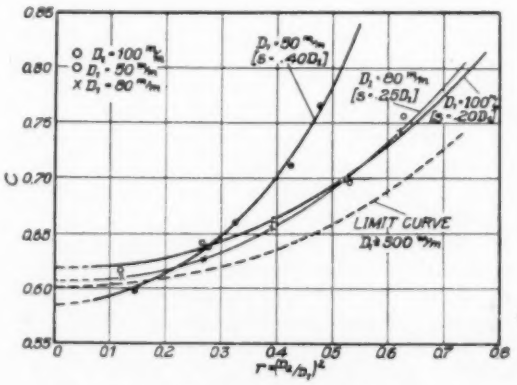


FIG. 5 FLOW OF WATER IN PIPES
(Pflaum, Fig. 24.)

absolute diameter of the pipe, as well as of the two dimensionless ratios, the Reynolds number and the acoustic velocity of the flow. All agree that the extent of this function can be determined only by experiment. There is, however, a further aspect to be considered.

It is by no means established that in the case of turbulent flow two pipes of the same relative roughness and different in size will have the same symmetrical curve of velocity distribution. Here again the basis of the mathematics is all a matter of assumption. One assumes that the retardation at the wall is transmitted through the medium of the fluid in a symmetrical way from the wall to the center. This is equivalent to assuming a symmetrical curve of velocity distribution for all sizes. If, however, one realizes that in the case of turbulent flow the effect of the wall, or what is termed skin friction, penetrates only a certain linear distance from the wall in each case, then even in pipes of the same relative roughness the velocity distribution

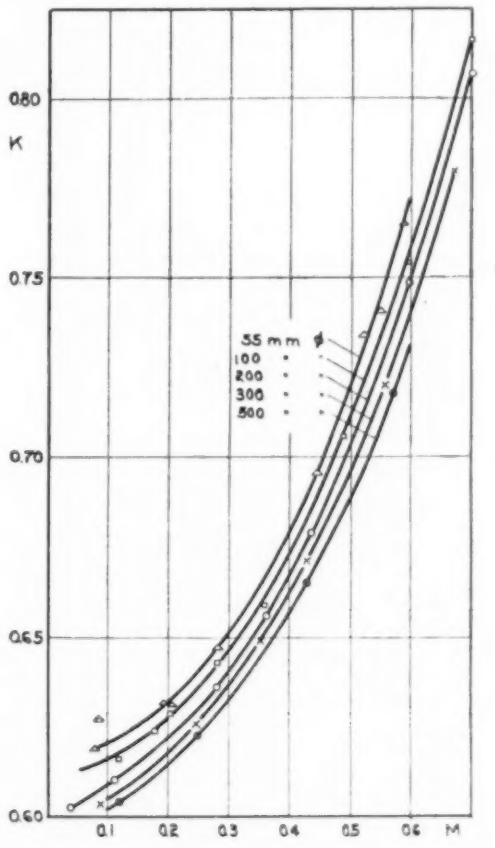


FIG. 6 FLOW OF VARIOUS FLUIDS IN PIPES
(Witte, Fig. 21.)

should be flatter in the larger pipe than in the smaller. Hence the reason for the variation of the coefficient with the absolute size of the pipe. How much this variation is, can be determined only by experiment.

THE REGION OF INSTABILITY

There is another tendency traceable to this theory which will be harmful to future developments. Hydraulic similarity is given the right of way in our recent publications of experimental work. However, it is used mainly for the establishment of the theory, and little attempt is made to follow hydraulic similarity in actual practice. A striking example of this nature is the lack of care used in the placing of pressure taps on the low-pressure side of an orifice plate. It is not appreciated that the jet issuing through the orifice has parallel lines of flow only at the *vena contracta*. Still even in experimental work connections are often made only at points where the flow is not parallel, where turbulence begins and no similarity of action should be expected.

As stated, Osborne Reynolds (20) made his experiments for the purpose of determining the dividing line between the two types of motion, streamline flow on one side and turbulent flow on the other. However, Reynolds' discovery showed that it was

not a line but an area or a region of instability between the two types, in which the flow changed from one condition to the other at the slightest disturbance in the stream.

Lord Rayleigh (23), immediately following Reynolds' work, set out to find a theory "to explain the complete change in the laws of flow which supervenes when the tubes are of larger diameter and the velocities not very small." While introducing the

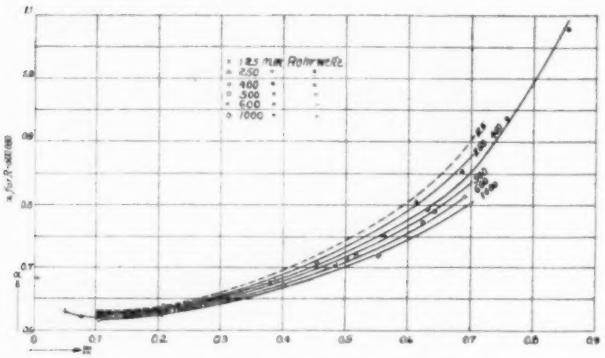


FIG. 7 FLOW OF WATER IN PIPES
(Jakob & Kretschmer, Fig. 72.)

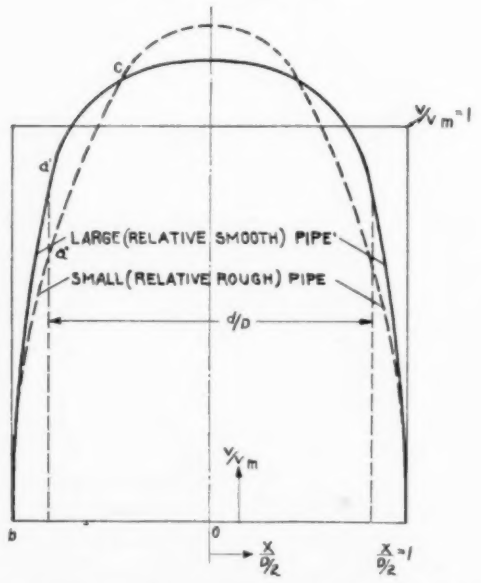


FIG. 8 EFFECT OF PIPE DIAMETER ON VALUE OF ORIFICE COEFFICIENT
(Jakob & Kretschmer, Fig. 73.)

subjects Rayleigh refers to Sir G. Stokes' theory of viscous flow and to Reynolds' application of the theory of dynamical similarity and to the experiments which showed "that the change in the law of resistance occurs when $VD\gamma/\mu$ has a certain value." To this Rayleigh adds: "The conclusion is perhaps most easily reached by applying the method of dimensions to the expression," etc. The application as given by Rayleigh shows clearly that the dimensional theory is only a method by which the physical balance of any equation is proved to comply with the corresponding set of assumptions. In the following derivations the parts "quoted" are copied from Rayleigh's text.

Let $P = p/L$, the ratio of the pressure drop to the length of the tube. "The dimensions of this ratio are those of a force divided by a volume. If it is assumed that this ratio may be expressed in terms of ν (equal to μ/γ, D, γ, and V in the form"

$$D^x \nu^y \gamma^z V^n$$

The dimensions of the quantities are:

$$P = ML^{-2}T^{-2}$$
$$D = L$$
$$\nu = L^2T^{-1}$$
$$\gamma = ML^{-3}$$
$$V = LT^{-1}$$

Therefore

$$ML^{-2}T^{-2} = L^x(L^2T^{-1})^y (ML^{-3})^z (LT^{-1})^n \quad \ldots\ldots (26)$$

Removing parentheses gives

$$ML^{-2}T^{-2} = M^zL^{x+2y-3z+n}T^{-y-n} \quad \ldots\ldots (27)$$

Equate the exponents; then, from M, $1 = z$; from L, $-2 = x + 2y - 3z + n$; from T, $-2 = -y - n$

Hence $x = n - 3, y = 2 - n, z = 1 \quad \ldots\ldots [28]$

and $P \sim D^{-3}\nu^2\gamma(DV/\nu)^n \quad \ldots\ldots [29]$

"Since n is here indeterminate, all that can be inferred from dynamical similarity is that

$$P = D^{-3}\nu^2\gamma f(DV/\nu) \quad \ldots\ldots [30]$$

where f is an arbitrary function."

"For capillary tubes and moderate velocities P varies as the first power of V, so that in Equation [29] $n = 1$. In this case

$$P = kD^{-2}\nu\gamma V \quad \ldots\ldots [31]$$

k being an arbitrary constant. When, on the other hand, DV/ν is great, experiment shows that $n = 2$ nearly. If this law be exact, Equation [29] gives

$$P = k'D^{-1}V^2\gamma \quad \ldots\ldots [32]$$

independent of ν. The second power of the velocity and independence of viscosity are thus inseparably connected."

It is well to note that while Rayleigh refers to Stokes' theory of viscous flow his solution of dimensions gives Poiseuille's equation and not Girard's formula as derived by Stokes. This emphasizes the importance of checking the physical balance of an equation. Equation [31] is identical in form with Equation [21], which is dimensionally correct. Equation [18] (Girard's formula) is dimensionally wrong, having an extra length in the second term, and the coefficient a must be considered as a physical constant. Equation [32] is identical in form with Equation (12), which is dimensionally correct.

After showing the method of dimensions, Rayleigh turns his attention to the formation of eddies with the reduction of viscosity. "From the description and drawings given by Reynolds it is natural to suppose that in the absence of viscosity the stratified motion would be unstable . . ." Rayleigh experimented for that purpose and found that stratified or linear flow is just as stable in the absence of viscosity. (The stream from a nozzle is a good example.) Rayleigh further makes the following significant statement with reference to the region of instability in a stratified or linear flow:

> We are thus confronted with a difficulty. For if the investigation in question can be applied to a fluid of infinitely small viscosity, how are we to explain the observed instability which occurs with moderate viscosities? It seems very unlikely that the first effect of increasing viscosity should be to introduce an instability not previously existent, while, as observations show, a large viscosity makes for stability.

This citation of physical phenomena is to my mind the keynote of the situation. The dimensional analysis shown by Rayleigh

and the π theorem developed by Dr. Buckingham do not apply to physical phenomena outside the given specified assumptions.

The Stanton and Pannell curve on fluid friction which was published 24 years later is an illustration of similar phenomena showing the area of instability between two types of motion. A reproduction of this curve is shown in Fig. 9. In that figure

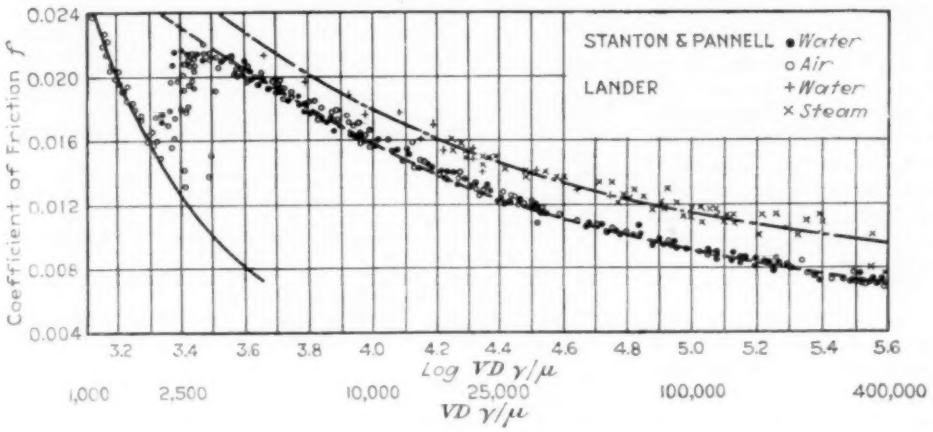


FIG. 9 COEFFICIENT OF FRICTION f AS A FUNCTION OF $VD\gamma/\mu$
(From Grunwald and Engel, Z.V.D.I., May, 1928.)

the friction coefficient, in the Chézy formula, is plotted against the log of the Reynolds number. Note at the left the stable condition of the viscous flow, and at the right the equally stable extension of the turbulent flow. Lord Rayleigh would probably say again, Why should the first effect of increasing the Reynolds number introduce an instability, while a further increase restores the stability? And further, why should this instability escape notice in the theoretical equations which cover the behavior of the fluid throughout the entire range?

With this idea in mind the author looked for a similar exposition of instability in the flow through a restricted opening. He realized that it was not the turbulent flow that was unstable, but it was the change from streamline to turbulent flow that produced the instability. One could comprehend why the results of measurement become uncertain when the low-pressure orifice tap was moved from the *vena contracta*, because the pressure came from a section where the flow was changing from streamline to turbulence and the pressure conditions were unstable. This

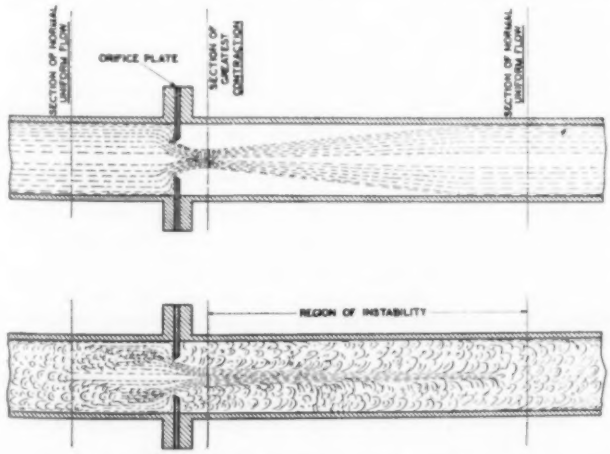


FIG. 10 REGION OF INSTABILITY IN FLOW SECTION
(Above, incorrect presentation of flow condition; Below, correct presentation of flow condition.)

phenomena is shown diagrammatically in Fig. 10. The upper section shows the customary presentation of the orifice or venturi section, which on the face of it is incorrect. The lower section shows the changes that are probably taking place in the section. The turbulence is changed into stratified or linear motion at the approach to the orifice, reaching full stratification at the *vena contracta*. Beyond this point the lines break into eddies, reaching the condition of turbulence again at several diameters below the section of the orifice. In Osborne Reynolds' language, the upper section shows the way an army marches on a parade; the similarity is perfect. The lower section shows the way an army marches in action, under the fire of the enemy.

SURVEY OF RECENT PUBLICATIONS

Six articles of special merit have appeared in the recent literature on the subject of fluid flow. Three of them were published in Germany, one in England, and two in the United States. Each gives full consideration to the laws of similarity.

The German authors, Messrs. Pflaum (27), Witte (28), and Jakob and Kretschmer (30), have used the Reynolds' function as a parameter with the understanding that there are other influences that may affect the coefficient. That is, they have shown that while it is advantageous to plot the coefficient on the parameter of similarity, the coefficient does not necessarily have to be on the same curve in all cases. They have shown that the coefficient of an orifice plate varies with the size of the pipe when plotted to the scale of the Reynolds' function. In other words, they have considered similarity as a theory and not as a law. A theory needs an experimental coefficient, and their experiments have shown that the coefficient, in its practical application, varies with the size of the pipe.

Mr. Hodgson (29), of England, stressed the validity of the viscous-flow criterion to a larger extent. In fact the title of his paper is "The Laws of Similarity." Mr. Hodgson believes that a coefficient obtained on a small size of pipe will be duplicated on a larger size if both the hydraulic and geometric similarity are maintained. He points out, however, that: "If in practice the pipes vary in roughness, the coefficients of discharge for otherwise geometrically similar orifices will not be the same, especially for the smaller pipes where a given change in the roughness produces the greatest effect."

Messrs. Bean, Buckingham, and Murphy (31), of the Bureau of Standards, have tested three sizes of pipe with a large number of orifice plates. They have combined the results of the 6-in. and 8-in. pipes into an empirical equation for the orifice coefficients. They have rejected the data for the 4-in. pipe because it showed consistently higher values for the coefficient than the other two pipes. They have gone extensively into the theory to show that the "absolute" size of the pipe could not have any effect on the coefficient. In other words, they believe that geometrically similar shapes must produce the same hydraulic similarity irrespective of size. They have arrived at this conclusion with the understanding that it applies to perfect similarity, which, as they have demonstrated themselves, is seldom obtained in practice.

Mr. Smith (32), in his contribution, has gone a step farther.

He has brought the modern theory of fluid flow or the laws of similarity to the front in a very forcible and convincing manner. His work is very valuable as a guide for determining the coefficient of discharge in and near the region of viscous flow. His contention is that the coefficient is still a function of the Reynolds number in turbulent flow, where according to his indications, the function becomes a constant and the curve remains "flat" for all values of the criterion. This means that after a certain turbulence is reached, the coefficient is independent of the charge or variation of either one or all of the factors of the criterion. However, this is not borne out in practice, since the coefficient does not have the same value for the various pipe sizes although it appears to be independent of the variations in the other quantities of the Reynolds number—velocity, viscosity, and density. This difference of the coefficient with size is due first to the practical difficulty of obtaining exact geometrical similarity with the various sizes of pipe and second to the effect of the boundary layer which may vary with the absolute size of the pipe.

Conclusion

Looking back on the historical development of fluid-flow measurement, one is greatly impressed with the effort exhibited first in one and then in the other branch of this development. It started from Torricelli's discovery, in 1643, of the velocity of the water jet. The main study then was the flow of water through orifices and jets issuing from large tanks. In the course of a century the attention of scientists was turned toward the investigation of friction drop of pressure in pipes. Almost all the theories on fluid flow of the succeeding two centuries have as their object the motion of fluids in tubes, pipes, and channels.

The practical application of fluid-flow measurement is nearly all a product of the twentieth century. In the two preceding centuries the profession was interested mainly in how to convey and deliver fluids through given conduits. The chief interest at present is how to measure the fluid with a reasonable degree of accuracy.

In both cases the development of the subject was greatly advanced by the analytical work of the various scientists in their respective times. In all cases it was experimental discoveries that formed the basis of scientific analysis.

It is hoped that this brief résumé of the development of the art will bring about a better understanding of the effects of hydraulic similarity as related to the geometrical similarity of the various devices. Geometrical similarity is of great importance, and devices which are geometrically similar will make the operation more uniform. Geometrical similarity is a matter of construction. Hydraulic similarity is a matter of experiment. Large sizes may or may not show the same results as the smaller ones.

Notation

A = area of conduit
a = first constant in Girard's formula
B = wetted perimeter
b = second constant in Girard's formula
C = coefficient of Torricelli's equation
D = pipe or tube diameter
f = coefficient of friction
g = acceleration of gravity
h = differential head
k = constant in Poiseuille's formula
L = length of pipe
n = constant in Chézy's formula
p = differential pressure
$r = A/B$ = mean hydraulic radius
$s = h/L$ = hydraulic slope
V = mean velocity
γ = density
μ = viscosity
ν = kinematic viscosity.

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Discussion of Papers on Fluid Flow[1]

LOUIS GESS.[2] In the opening paragraph of the paper by Messrs. Beitler and Bucher the authors make a statement regarding the accuracy demanded by the user of fluid meters today. It would have been well, the writer believes, to have made a statement that in order to obtain these accuracies the customer must pay more attention to the recommendations of the Fluid Meter Committee and other authorities. The results obtained by these tests would never have been possible if the investigators had not taken all the precautions which they did.

No mention is made by the authors as to the leakage tests on the condenser used when making the tests with steam.

G. D. CONLEE.[3] The paper by Messrs. Beitler and Bucher is a real addition to the volume of data on orifice coefficients. It is really surprising how few reliable data of this sort are being published. It is the writer's hope that the promise made will be fulfilled and that these data will be very much extended.

The advocates of similarity tell us that data obtained on one size may be applied to all other sizes of similar construction. This may be theoretically correct, but by all means give us actual test data on reasonable-sized commercial pipes on which to base the extension of values into the territory where actual experiments are out of the question.

A point of great interest in orifice design, or rather installation, is the location of the downstream, or low-pressure, tap. Practical experience has shown that this tap must be located at the *vena contracta*, or point of minimum pressure, and if the tap is not so located the results cannot be exactly predicted.

The accompanying Fig. 1 is a plot showing the location of the *vena contracta* with reference to pipe diameter and orifice ratio. The curved line was obtained from the tests made by Mr. Spitzglass and reported on in his "Orifice Coefficients." The straighter line is a location defined as 2.5 times the height of the orifice rim, that is, 1.25 $(D - d)$, where D is the pipe diameter and d the orifice diameter. The small circles show the various points obtained from this paper.

It is to be noted that the two are very close together, the difference being less than the possible effect of the pressure connections used. They were of $^3/_4$-in. brass pipe which probably had an internal diameter of about $^1/_2$ in. which corresponds to nearly 0.1 on the scale of abscissas.

From these curves it is obvious that the greater the orifice ratio, the nearer to the orifice is the *vena contracta*. One thing not shown in the plot, but generally accepted, is that the higher the orifice ratio, the more critical the tap location becomes. As a consequence, if it were possible to keep all orifice ratios below, say, 60 per cent, there would be very little trouble due to improper tap location. This is a practice adhered to in the measurement of natural gas. It is for this reason that the Bureau of Standards' tests in paper No. 49 were not satisfactory for higher-value orifice ratios. (See p. 589.)

In the use of flow meters for the measurement of steam and water, due to the high velocities experienced, it is necessary to use high ratio orifices, and the writer believes that carelessness in locating the low-pressure tap has had a great deal to do with errors in metering and results which were not comparative.

The data on compressibility are very interesting and give some direct information on this subject. With the very high steam velocities used, up to 20,000 ft. per min. in some cases, the orifice ratios and differentials are both high and the effect of compressibility has caused the writer a lot of thought.

In the second edition of "Fluid Meters," page 58, is a table of "Values of Factor Y" based on condition p_1. Y is the theoretical factor to be used for correcting the hydraulic equation for the effect of compressibility. These values are carried out to $R = 2.0$, or $d/D = 0.50$, a value for orifice ratio which is of little use in present-day steam measurement.

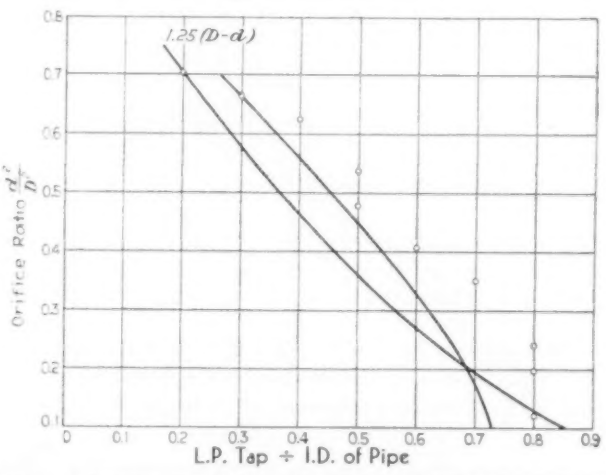


FIG. 1 LOCATION OF "VENA CONTRACTA" WITH REFERENCE TO PIPE DIAMETER AND ORIFICE RATIO

These values have been extended to cover higher ratios and greater pressure drops. We have not had the opportunity of making sufficient tests to verify the extent of this correction. However, in a number of cases we have been able to check the output steam from boilers against an input water meter. Without the application of the compressibility correction, the steam meter reads higher than the water meter. When this *theoretical* correction is made for compressibility, there has been in every case a reasonable spread between the water and the steam. Fortunately it has been possible to develop a device which would automatically correct for this condition, and we have had this device in use for the last year.

R. E. SPRENKLE.[4] Throughout the series of tests by Messrs. Beitler and Bucher on the six-inch-size pipe, using orifice and nozzles as the differential-pressure-producing media, the authors of this paper used exceptionally great care in watching all the fine points pertaining to this calibration work in an effort to eliminate as many unknown factors as possible. Attention is called to the method of reading static pressures on the inlet side of the orifice, by means of a calibrated mercury manometer instead of the usual pressure gage. All readings of this U-tube were corrected for all changes in barometer, temperature, relative

[1] "The Flow of Fluids Through Orifices in Six-Inch Pipes," by S. R. Beitler and Paul Bucher, pp. 77–87.
"Quantity-Rate Fluid Meters," by Ed S. Smith, Jr., pp. 89–109.
"Similarity: Limitations in Its Application to Fluid Flow," by J. M. Spitzglass, pp. 111–121.

[2] Engineering Department, The Brown Instrument Co., Philadelphia, Pa. Jun. A.S.M.E.

[3] Chief Engineer, Republic Flow Meters Co., Chicago, Ill. Mem. A.S.M.E.

[4] Mechanical Engineer, Bailey Meter Co., Cleveland, Ohio. Assoc-Mem. A.S.M.E.

humidity, etc. An earnest endeavor also was made to hold the steam conditions absolutely constant for the duration of the run. This included a long warming-up period in order to be sure every condition was stabilized absolutely so as to prevent any energy losses due to any change in the steam conditions.

Mention was made in the appendix to this paper, of the correction applied to the two sides of the water manometer for the unbalanced air column. This is a refinement which, to our knowledge, has not been duplicated in any previous work. An idea of the magnitude of this correction is gained from Fig. 2, which shows the relation between the unbalanced air column on one leg of the U-tube and the necessary correction to be applied to the manometer readings to correct for this unbalanced head. It will be noted that at air pressures of 75 lb. abs. a correction of approximately 0.3 per cent is necessary, which is an appreciable amount in this type of calibration work.

The radiator and reservoir connections to the steam pipe, as shown in Fig. 14 of the paper, were so designed as to maintain equal water columns on the reading manometer. This pre-

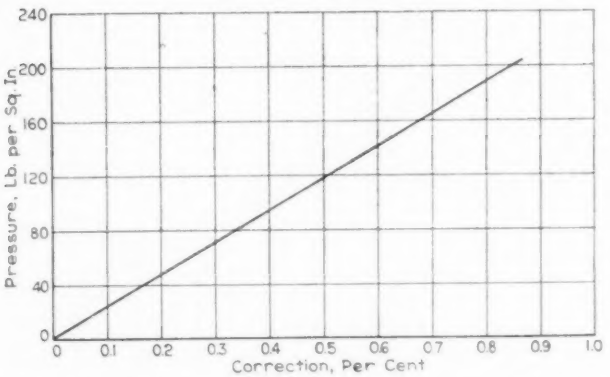


FIG. 2 CURVE SHOWING CORRECTION FOR PRESSURE OF AIR ABOVE WATER COLUMN

supposes an equal water level in both connections, this water level starting with the offset gate valve. While there is no reason to suspect that equal water legs did not exist in this series of tests, yet since it was not possible to see, visually, these water levels in the gate valves, some doubt was expressed as to whether this was the best type of apparatus to use in accurate laboratory calibration methods. The scheme of steam-flow calibration hook-up, as shown here in Fig. 3, was then devised and has now been in operation for some time. This scheme does away with the radiator connection as shown in Fig. 14 of the paper, and in its place is substituted essentially a simple water manometer connected through condensate chambers directly to the steam line.

Provision for cooling these manometer connections is made by introducing at the top of each manometer leg high-pressure air at regular intervals. Before readings are taken on the water manometer, the air is admitted to the manometer system by closing valves 1, opening valves 2, and opening the high-pressure-air valve supply. The rate of admission of air to each leg of the U-tube column is regulated by the needle valves 4. By this method the connecting piping between the manometer and the steam line is blown out thoroughly, thus cooling the tubes. Excess condensation is drained out of the condensate chambers whenever necessary, but experience with this device has shown it is not necessary to remove the condensate very often.

Corrections are also made in this system for the unbalanced air column on top of the manometer, as well as for the temperatures in the manometer itself.

SANFORD A. MOSS.[6] A great deal of matter in the three papers is concerned with attempts to measure fluid flow with accuracy with use of orifices in thin plates. There are cited as difficulties

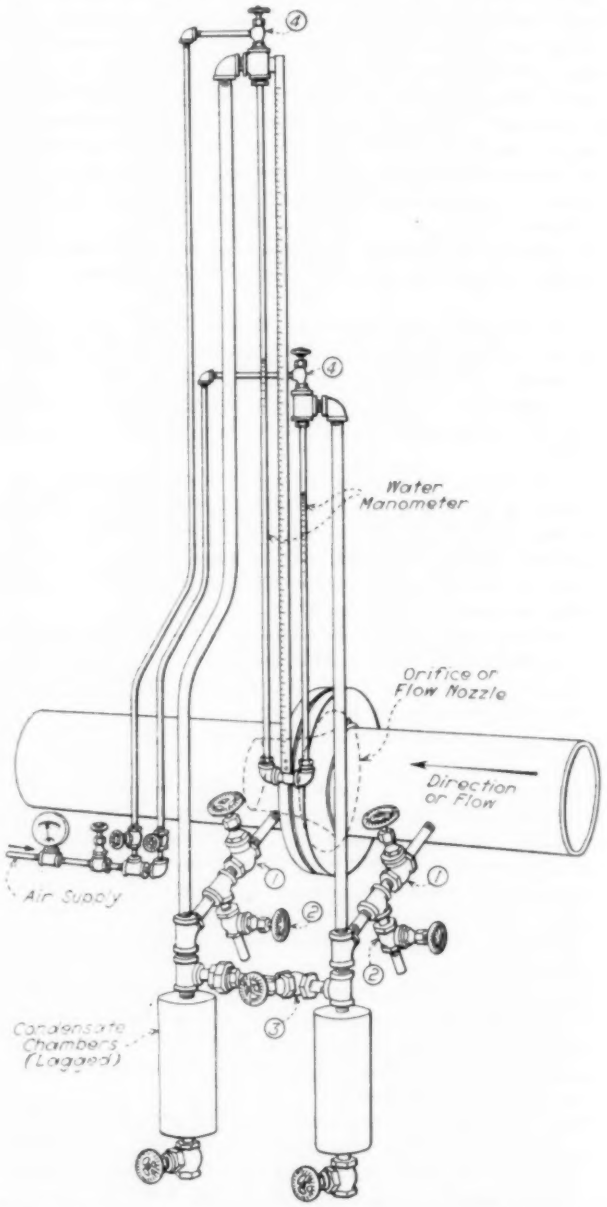


FIG. 3 METHOD OF MEASURING DIFFERENTIALS AS PRODUCED BY STEAM FLOW AT OHIO STATE UNIVERSITY

in this connection, the failure of the *vena contracta* to follow the dimensional laws, the difficulty of location of the downstream pressure taps, the failure of the coefficient with steam as found by Messrs. Beitler and Bucher to follow the theoretical expansibility laws, the disagreement of the coefficients of these gentlemen with those of Judd and Spitzglass, and the raggedness of their plotted observations in spite of their obvious care in making the experiments.

Of course a conclusion which can be drawn is that when accurate flow measurement is wanted, one should not use an

[6] Mechanical Engineer, Thomson Research Laboratory, General Electric Co., Lynn, Mass. Mem. A.S.M.E.

orifice in a thin plate at all, but should use a venturi meter, or nozzle with well-rounded approach. This is illustrated in the paper of Messrs. Beitler and Bucher, although it is preferable to use a smaller nozzle diameter in proportion to the pipe size. Such nozzles have been presented in a number of A.S.M.E. papers and were used as primary standards for calibration of orifices in thin plates in the Bureau of Standards paper by Bean, Buckingham, and Murphy cited. With such nozzles all questions of dimensional similarity are concerned only with the very small differences between 0.96 and 1.00.

It is to be noted that the complicated factors involving the ratio of pipe to nozzle diameter can be avoided by use of an impact tube instead of a static hole for the initial pressure. This may be before or after the nozzle.

The factor Y and the acoustic ratio x/n as used in the papers are very convenient ways for presenting results for all kinds of fluids in a single formula. However, if one deals with compressible fluids only, there is a slight extra complication in finding the results for an incompressible fluid, and then multiplying by a factor Y which is a function of x/n. It is somewhat simpler in dealing with gas only, to use a single formula, expressly arranged for the compressibility of the problems at hand.

Louis Gess.[6] Referring to Chapter I of Mr. Smith's paper, which is offered as a basis for the revision of Part I of the A.S.M.E. Fluid Meter Report, undoubtedly a revision of that report is necessary. Since it is desired that it be widely read by the users of fluid meters, the revision should be carefully done. A large number of these readers will be only interested in the subject generally, and not in theoretical discussions. The writer would suggest, then, that the first portion be written primarily for the engineer who wishes to know something about the subject, and has only a limited amount of time in which to do so, and that the second portion be added for the benefit of those who wish to know something of the theory of fluid meters. If the report is handled in this manner it will be equally satisfactory to both classes of readers.

R. E. Sprenkle.[7] The writer has enjoyed very much studying Mr. Smith's paper and its application to commercial metering, and believes that the data it presents will prove of great value to the art of metering.

Smith's Fig. 2 showing the relation between the expansion factor Y and the acoustic ratio x/n is very interesting and represents a decided advance over the information as published in the second edition of Part I of the A.S.M.E. Fluid Meters Report. This curve is especially interesting in view of the research work on steam measurement as being carried on by the Ohio State University at the present time on three- and six-inch steam lines. In this research all orifices used have throat connections, that is, the inlet connection is made one diameter on the inlet side of the orifice and the outlet connection at the *vena contracta*. Results of this study of the adiabatic expansion of the gaseous flow through a 50 per cent ratio concentric orifice are shown on the writer's Fig. 4. By plotting these data against the acoustic ratio as was done in Mr. Smith's Fig. 2, it is indeed gratifying to note that the curve expressing this relationship coincides almost exactly with the 40 per cent ratio orifice curve as shown in Mr. Smith's Fig. 2. Mr. Smith's curve shows a slight curvature, whereas the data which are shown on the writer's Fig. 4 plotted up to be substantially a straight line. However, the difference is so slight that it can be disregarded for all practical purposes. It is presumed from the data given on the figure that air was used as the flowing medium, since the value of n, or the ratio of the specific heats of the gas for all the solid lines, is given as 1.408, which is that of air.

Under the caption entitled "Standard Differential Producers" definitions are given for venturi tubes, venturi nozzles, venturi nozzle tubes, flow nozzles, and orifices with flange, throat, and pipe-line connections. This group should be enlarged to include other primary devices, such as various forms of pitot tubes, eccentric and segmental orifices, orifice segments, and venturi segments. Better yet, it would be advisable, in the writer's opinion, to omit this classification from this proposed revision of the A.S.M.E. Report, for descriptions of all the standard types of differential pressure-producing devices are covered in detail in Part 3, to be published very shortly, and any description of these primary devices in Part I would therefore be mere repetition.

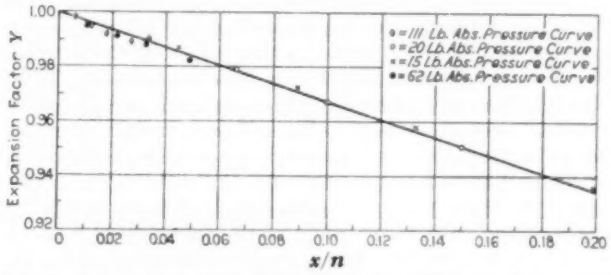


Fig. 4 Relation Between the Expansion Factor Y and the Acoustic Ratio x/n—Superheated Steam for 50 Per Cent Ratio Orifice

(r = ratio of differential pressure to absolute static pressure; n = ratio of specific heats—taken at 1.30 for superheated steam.)

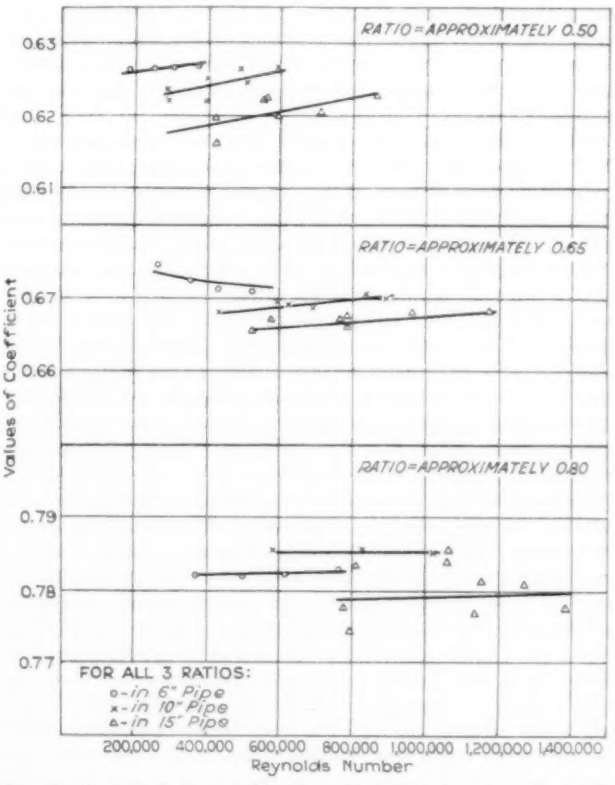


Fig. 5 Coefficients of Various Primary Devices Plotted Against Reynolds' Numbers

In the first sentence at the top of page 96, Mr. Smith states that the coefficient must therefore be a function of the Reynolds

[6] Engineering Department, The Brown Instrument Co., Philadelphia, Pa. Jun. A.S.M.E.

[7] Mechanical Engineer, Bailey Meter Co., Cleveland, Ohio. Assoc-Mem. A.S.M.E.

number only. Further attention is called to the fact that an identical coefficient will be obtained when complete geometric similarity of the shape of the differential producer and exact similarity of the flow through the producer are obtained. These statements, as the writer understands them, would indicate that the same ratio orifice, assuming that all points of design and conditions of flow are similar, would have the same absolute coefficient value regardless of pipe size. Fig. 5 shows graphically that such a relationship does not hold true, as shown by data obtained from hundreds of experiments conducted using steam and water as the flowing medium. The variation in coefficient for pipe size is quite appreciable, as will be noted from the curves, and is not covered by the Reynolds number. While it appears from data obtained at the Bailey Meter Company laboratory and also the Ohio State University laboratory, as well as from the data in Mr. Smith's paper, that the Reynolds number does serve a real need in ascertaining the correct coefficient of discharge of a fluid having entirely different properties as to pressure, temperature, and viscosity from that of a tested fluid, yet it does not hold that the Reynolds number includes the variation in coefficient due to change in the pipe diameter and that such data can only be obtained by experimental research.

In the second sentence at the top of page 100, Mr. Smith makes the statement that for a venturi tube or venturi nozzle the coefficient for a small diameter will be only slightly less than for larger diameters, since the additional friction increases the differential slightly. It is of interest that the coefficient of small-diameter flow nozzles is slightly larger than those for the large diameters. This holds true both as to variation in pipe size and in diameter ratio.

In discussing the application of orifices in the metering of dirty fluids or gases, the statement is made that there is more probability of erroneous coefficient being caused by deposits of material in front of the orifice, than would be the case where primary devices such as venturi tubes or flow nozzles are used. Further, statement is made that where wet steam is being metered there is a greater tendency for the entrained liquid to be thrown out of the steam and accumulate in front of the orifice, which might affect the coefficient. Attention is called to the fact that orifices can be made with the hole flush with the bottom of the pipe or so made that the only portion of the rim which projects inside the pipe is the segment which can be placed on top of pipe, either of which methods will eliminate the filling up of the pipe immediately in front of the orifice.

On the other hand, the orifice is a decidedly superior type of primary device in the use of metering dirty liquids or gases. On account of its relatively sharp edge, accumulations of extraneous matter will not form along the orifice edge, except possibly under the very worst conditions, and then not nearly as fast as in the case of a flow nozzle or a venturi tube. The use of pitot tubes or nozzle plugs is even more to be avoided than that of flow nozzles or tubes in the measurement of this type of fluid, on account of the rapidity with which the impact and static holes of the tubes become fouled in service. Mention is further made that venturi tubes are now provided with steam jackets for scavenging the throats of these tubes for naphthalene or tarry deposits. Attention is called to Fig. 6, in which the same idea is applied to an orifice installation.

It is noted that in Equation [69] Mr. Smith has included separate coefficients for friction and contraction, and further, that he has expressed the ratio R_c as referring to the diameter at the *vena contracta*. Such a formula for orifice design is extremely difficult to use. The writer sees no object in separating the various coefficients of discharge of an orifice into their component parts of friction, contraction, and other unknowns, nor does he deem it advisable to calculate the area ratio on the basis of the diameter of the *vena contracta*. A much simpler formula and one which can be applied quite readily in commercial design can be expressed as follows:

$$Q = CY \frac{1}{\sqrt{1 - r^4}} A\sqrt{2P/W}$$

in which C represents the coefficient of discharge, Y the expansion factor, r the ratio of the orifice hole to the diameter of the pipe on the inlet side, A the orifice area in square feet, P the differential pressure in poundals per square foot, and W the density of the fluid under the conditions at the inlet at the orifice. If gravitational units are not objectionable, as indeed they should not be in practically any condition, the above formula can be written as follows:

$$Q = CY \frac{1}{\sqrt{1 - r^4}} A\sqrt{2gH}$$

In this case g is the gravitational effect having a value of 32.174 and H is the differential head of the fluid at the density of the flowing conditions on the inlet side of the orifice. Such an equation reduces the unknown coefficient to one term C, and modifies this coefficient by the expansion factor Y in the case of compressible gases.

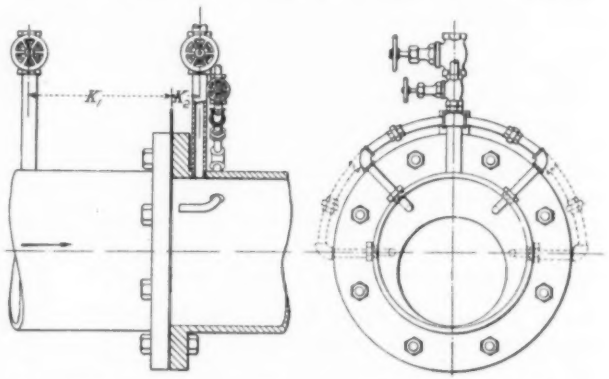


FIG. 6 STEAM JETS FOR CLEANING ORIFICES IN GAS LINES

A. J. NICHOLAS.[8] The writer hopes that he may be pardoned in digressing a little from the main topic of the papers by mentioning the work of the pioneers who initiated the steps that now bring us to our present concept of fluid flow and its measurement.

Stokes[9] in 1850 showed mathematically the factors that enter into fluid friction. Reynolds[10] in 1883 experimentally proved the critical points of fluid flow, showing where viscous and turbulent phenomena occurred. Lord Rayleigh[11] in 1892 proved Reynolds' experimental work to be mathematically correct. Stanton,[12] Bairstow, and Pannell in 1912 experimentally substantiated the mathematical concepts of Stokes and Rayleigh for flow in smooth and rough pipes. Stanton in 1917, while discussing Hodgson's[13] paper, suggested the plotting of meter coefficients on the basis of Reynolds' number. This Hodgson subsequently attempted in Figs. 32, 33, and 35 of his paper. These figures are undoubtedly the first published experimental coefficients plotted on the basis of dynamical similarity. A check of V_1D_1/U, which was used as the basis of plotting, shows it to

[8] Engineering Research Division, Hudson Motor Car Company, Detroit, Mich. Assoc-Mem. A.S.M.E.

[9] Stokes, Math. and Phys. Papers, vol. 111, p. 17 (1850).

[10] Reynolds, Phil. Trans. Roy. Soc., p. 935 (1883).

[11] Rayleigh, Phil. Mag., 5th ser., vol. 34, p. 59 (1892).

[12] Stanton, *Engrg.*, p. 437 (March 29, 1912).

[13] Hodgson, Min. Proc. Inst. C. E., vol. 204, part 2 (1916–1917).

be 10^2 too high, but the value of the work is nevertheless unimpaired if the multiplying factor of V_1D_1/U is 10^4 instead of 10^6 as given. Hodgson[14] realized this error shortly after the proofs had been passed and the paper had been published. Pannell[15] in 1919 anticipated a similar method of plotting coefficients for orifices, which unfortunately never materialized due to the untimely death which befell him shortly after as a passenger on a disastrous flight of a British dirigible. In discussing the Stanton orifice meter in his paper, for which geometrically similar orifices of 0.25 to 1.25 in. were available, Pannell stated, "The author hopes at some future date to publish these calibration figures and show that the results for the different orifices may be compared on a basis deduced from the principle of dynamical similarity." Hodgson[16,17] subsequently published orifice, nozzle, and venturi calibrations on this basis in 1921 and 1922. Since then this means has become more known and popular as a method of coordinating fluid flow.

With reference to Fig. 1 of Mr. Smith's paper, an omission of one of the important metering elements is noted, namely, the shaped nozzle. Such data are available from Hodgson's[18] paper for a complete range of Reynolds' number.

Hodgson,[19] in plotting coefficients as in Fig. 3 for similar types of orifices, has a straight line for ratios of 0.09 to 0.7, consisting of seven orifice ratios. Mr. Smith shows only three points, which seems to give a hump and a droop to the curve.

The significance of Equation [13] may be further amplified on the basis of Newton's second law. It is obvious that since the force is proportional to U and the mass of fluid per unit volume accelerated by this force is proportional to W, the accelerating effect will depend on the ratio U/W, i.e., force/mass, is proportional to acceleration. It is interesting to note that Maxwell was first to name this ratio U/W as the "kinematic coefficient" of viscosity, and the reason for it is obvious.

Mr. Smith states that gaseous flow at critical-pressure condition is generally known as Napier's relation. This relation is not generally known as such. Osborne Reynolds[20] was first to explain this phenomenon and showed that the velocity of the fluid leaving the metering element is equal to that with which sound is propagated in the fluid, so that any reduction in the back pressure cannot be communicated back against the stream.

It seems that Mr. Smith's Equation [25] is incorrect, which may be verified by the substitution of the fundamental units of mass, length, and time. These show that the equation as given is unbalanced by J. Equation [26], which follows, is correct, but cannot of course be arrived at from the preceding equation. Equation [25] is intended to be a form similar to Poiseuille's formula, which may readily be deduced by the method of dimensions in the form of $P_m = K_1 \frac{UV_m}{D_m}\left(\frac{VD_mW}{U}\right)^n$ so that $P_m = \frac{KUV_m}{D_m}$ from which Equation [26] naturally follows. It is interesting to note that for orifices and nozzles of short length, the discharge $Q_m = \frac{KD^3{}_mP_m}{U}$ where $K = f(R)$. Hodgson[21] deduces a modified equation for viscous flow on this basis, embodying the velocity of approach factors as in Mr. Smith's Equation [1]. The writer believes such an equation should be incorporated in the Committee's revised report, even though its field of usefulness for commercial metering purposes should be limited. Its usefulness in other fields of measurement cannot be overemphasized, such as, for example, research fields and the writer's own automotive work, where fluids are accurately metered in small quantities of varying viscosities in small-sized conduits.

In the writer's opinion the author's physical interpretation of Reynolds' number, flow similarity, and coefficients is unconvincing. A physical interpretation is no more convincing than a mathematical derivation properly diagnosed.

In connection with acoustic ratios the writer notes that in Mr. Smith's Fig. 5 relations are determined for steam with $n = 1.135$. This may be true for venturi tubes, but for orifices and nozzles it is more than likely that the expansion factor $n = 1.3$, based on the theory of supersaturation. Callendar[22] showed the discrepancy in flow through nozzles or orifices based on Zeuner's value of $n = 1.135$ and supersaturation theory with $n = 1.3$. There seems to be no doubt that quantity rate for nozzles or orifices should be based on $n = 1.3$, rather than 1.135. In the case of venturi tubes the value of $n = 1.135$ may be correct. Has Mr. Smith based the computations of Fig. 5 for the venturi on the assumption that Zeuner's value is correct, or was the value of n actually deduced by experiment with venturi tubes?

In conclusion the writer congratulates Mr. Smith on presenting a paper of great merit. It is a decided step toward standardization of all fluid meters on a common basis. In the writer's opinion it is an outstanding contribution on fluid meters.

STEPHEN E. SLOCUM.[23] The problem discussed in Mr. Spitzglass' paper concerns flow through pipes and orifices. This is of such a limited and special nature that the true significance of the hydraulic principles involved is not always apparent, and it may therefore add something to the discussion to consider the problem from a somewhat more general standpoint.

First of all, it seems necessary to confine the discussion to laminar or streamline flow, since it has been shown by experiment that turbulent flow in pipes follows quite different laws. So far as rational analysis is concerned, it must, for the present at least, be limited to laminar flow since the canonical laws of hydraulics are based on this state of motion.

Assuming laminar flow, geometrical similarity as defined in this paper is not necessarily equivalent to dynamical similarity, which is the proper basis for comparison. The writer is here considering the flow of a viscous fluid past a solid boundary, which in general may be either the surface of a solid immersed in the fluid or, as in the present case, the inside of a pipe wall. When the boundaries of two objects are identical in form but differ in scale or dimensions, then if l denotes the ratio of two homologous dimensions, such for instance as pipe diameters, in order for the conditions governing flow to be comparable, the following relations must be fulfilled:

Speed of advance (or relative velocity).....	$V = v\sqrt{l}$
Speed of revolution.....	$N = n \div \sqrt{l}$
Unit pressure.....	$P = pl^3$
Torque.....	$T = tl^4$
Power.....	$H = hl^{7/2}$

Solids geometrically similar in form, therefore, do not insure hydraulic similarity of flow, irrespective of size; or, stated conversely, dynamical similarity is a function of the relative scale of geometrically similar forms.

It is well known, however, that there are often apparent discrepancies in comparative results even when the laws of dynami-

[14] Personal communication with the writer.
[15] Pannell, *Engrg.*, p. 261 (Feb. 28, 1919).
[16] Hodgson, Min. Proc. Inst. C. E., vol. 212, part 2, p. 272 (1920–1921).
[17] Hodgson, Trans. Inst. Naval Architects, vol. LXIV (1922).
[18] Hodgson, loc., cit. 9.
[19] Hodgson, loc., cit. 9.
[20] Reynolds, *Phil. Mag.*, vol. II, p. 311 (March, 1886).
[21] Hodgson, Inst. C. E. Paper No. 31, p. 22 (1925).
[22] Callendar, Inst. M. E. (1915).
[23] Consulting Engineer, Ardmore, Pa.

cal similarity are satisfied. For example, in predicting the performance of a full-sized ship from model tests, the power required to drive the ship at a speed corresponding to that of the model, namely $V = v\sqrt{l}$, is often as much as 20 per cent greater than the corresponding power derived from the relation $P = pl^{7/2}$. This fact has long been recognized, but has recently been presented again as a result of experiments carried out in the Washington model tank as part of the present research program of the United States Shipping Board.[24]

The cause of such apparent discrepancy is probably to be looked for in the laws governing vortex formation arising from flow of a viscous fluid past a solid boundary. A viscous fluid, such as water or air, differs from an ideal fluid by the property that its retardation in the neighborhood of a solid boundary produces a steep velocity gradient which results in the formation of a vortex layer or sheet along the boundary. In the free fluid, viscosity effects are negligible, but in the neighborhood of a solid boundary they are of the same order of magnitude as inertia effects.

So-called frictional resistance in fluid flow along a boundary by no means resembles what we know as sliding friction. It is created rather by building up a vortex sheet along the boundary, which separates this boundary from laminar flow in the fluid itself. For example, in the case of a ship moving through the water, vortex sheets are formed along both sides of the vessel, which peel off from the stern on each side and persist behind the vessel, forming actual boundaries between the wake stream and the undisturbed water lying outside. It has been shown by Karman, Rehbach, and others that the resistance of an immersed body like a ship is a function of this vortex formation, depending jointly on the distance apart of the vortex sheets, the spacing or pitch of the vortex filaments in each sheet, and the strength of these vortex filaments. Very possibly the laws of vortex formation, on which resistance depends, are not directly related to the laws of dynamical similarity mentioned above, which would explain in whole or in part the discrepancy existing between model and full-scale results as now interpreted.

Rayleigh's law, VL/K = constant, where K denotes the kinematic viscosity (or Reynolds' number, as the left member is called), is certainly well established as the proper basis for the comparison of fluid flow. It is not so clear, however, that it applies without modification to the boundary layer, or vortex sheet, adjacent to a pipe wall. As mentioned above, the viscosity effects in the neighborhood of a boundary are of the same order of magnitude as inertia effects, and as the size of pipe decreases, these necessarily become of greater relative importance in their effect on discharge.

A valuable feature of the Reynolds number is that it makes it possible to apply the extensive data obtained in aeronautics directly to hydraulic problems. Thus if the subscript w in the following relation applies to water, and the subscript a to air, Rayleigh's law becomes

$$V_w L_w = \frac{1}{13} V_a L_a$$

By means of this numerical relation one is thus enabled to apply many of the results derived from wind-tunnel tests directly to marine practice. In particular, one can pick out from the innumerable airfoil shapes that have been tested those which are most suitable for blade sections of marine propellers or of hydraulic turbines.

The exceptions cited in Mr. Spitzglass' paper, therefore, do not discredit existing hydraulic laws, but reveal their limitations and indicate the need for further research in the domain of vortex phenomena, which has proved such a fertile field for investigation in the hands of scientists like Prandtl.

DR. RUDOLF WITTE.[25] As long as the effect of friction on flow measurement cannot be mastered by mathematics, that effect must be determined experimentally and must be presented by an empirical coefficient. For the comparison of individual measurements we have, for the time being, only the mechanics of similarity at our disposal. These teach us that a result obtained under a given research condition is applicable also to other conditions when there are definite relations between the dimensions of the magnitudes involved. They also teach us how the research data can be presented (namely, as a function of the so-determined dimensionless characteristics) so that the results may be extended beyond the scope of the original data.

For flow in closed pipes the fundamental hydraulic equations of Euler-Stokes are applicable:

$$\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial t} = -\frac{1}{\rho}\frac{\partial p}{\partial x} + \frac{\mu}{\rho}\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial u^2} + \frac{\partial^2 u}{\partial t^2}\right)$$

p = pressure, ρ = density, μ = viscosity

These equations apply equally for liquids and gases as long as the change in density with the change in pressure and temperature can be ignored—also for adiabatic phenomena where the expansion is not significant. We must also bear in mind that such possible outside energy influences as weight, capillarity, and compressibility are not considered in the relation, and these must not have any influence. The forces which appear under such simplified assumptions in the Euler-Stokes fundamental equations are of various types (inertia forces, friction forces, pressure drops): if in general the lengths are changed in a definite proportion f_l, if the velocities are changed in another definite proportion f_w, and—transferring to another fluid—if the density and viscosity are changed in a similar way in the proportion of f_ρ and f_μ, respectively (that is, all the fundamental dimensions which are involved), then each one of the forces assumes a dimensionless factor which can be constructed from the four scale relations:

$$\frac{f_\rho f_w^2}{f_l}, \quad \frac{f_\rho}{f_l}, \quad \frac{f_w f_w}{f_l^2}$$

By equating these various factors we arrive at the limiting conditions under which the equations remain true; that is, under which the forces remain in equilibrium:

$$f_w = \frac{f_w}{f_\rho f_l}, \text{ or } \frac{l_1 w_1 \rho_1}{\mu_1} = \frac{l_2 w_2 \rho_2}{\mu_2}$$

We then find that the limiting condition is the so-called Reynolds characteristic, according to which the relation of diameter times velocity, divided by kinematic viscosity, must remain constant. When the measuring installation is changed so that it is geometrically similar, then the velocity must be changed in inverse proportion to the lengths, or if the fluid is changed, the velocity must be changed directly with the viscosity. Following this and *by strictly adhering to the given assumptions* one must absolutely demand the same dimensionless coefficient for equal values of Reynolds' function in the case of restricted openings. In practice, however, we cannot strictly adhere to these assumptions, and only experiment can determine the practical deviations from the general rule.

From this standpoint the writer has carried through, in his publication of 1928, an examination of the application of the laws of similarity, with the result that for one and the same restricted opening in the same pipe the law of similarity holds true within

[24] *Marine Engineering and Shipping Age*, March, 1930, p. 145.

[25] Ludwigshafen am Rhein, Germany.

one per cent—for water, oil, steam, and air—and that in the case of the orifice plates when the pipe diameter is changed the law of similarity does not hold—as Spitzglass has shown before. From this deviation of the law of similarity the writer came to the conclusion that in the respective orifice coefficients there exists another unknown influence, and he undertook the task of discovering this influence and thereby explain the dependence of the coefficient upon the size of the diameter. In another work, which has been in the hands of the V.D.I. since the middle of September and which will be published in January, 1930, he has accomplished this with the results that when strictly adhering to the geometrical similarity of the orifice plate and the pipe wall surface, the laws of similarity apply exactly for various pipe diameters, because he has obtained *for pipe diameters from 500 to 1000 mm. exactly the same coefficients* for the same orifice ratios. In the case of ordinary rough pipes the requirements for geometrical similarity are not fulfilled, and in fact this has a considerable influence on diameters under 300 mm. Then it is also possible that in the case of rough pipes and small diameters the limits are not the same for liquids as for gases. (With reference to the limiting film on which the surface forces are acting.) For this the writer has conducted further very careful calibration tests with air and then even in the small diameter of 50 mm. he found the laws of similarity correct. It is understood that according to the above limitations on the derivation of the Reynolds characteristics the coefficient for water can be applied to air only if we neglect small changes in density. For larger expansions (differentials in gas measurement) there is nothing definite that the law of similarity can show about the coefficient, and in fact the coefficient is different at larger differentials.

In many cases the Reynolds function becomes a *necessity* for presenting measurement data. The presentation of data on the basis of velocity is at least not entirely correct—because when the temperature changes (change in viscosity) there is at the same velocity a change in the coefficient far into the region of turbulent flow. For rough pipes this theory is not yet completed. The application of the law of similarity for rough pipes is yet to be established.

The writer believes, therefore, that the various laws of similarity are actual laws which hold for the assumptions for which they are derived. Any deviation in the experiments with models must be attributed to the insufficient fulfilment of similarity requirements. In studying these deviations—as for instance the diameter influence on the orifice coefficient—we learn new influences which must be corrected for in the methods of fluid measurement.

Dr. Max Jakob.[26] I have noted with much interest the exceedingly fundamental outline of historical development in Mr. Spitzglass' paper. In addition I have the following to say: I have not at hand the titles of Odquist and Härlin's works to which we (Jakob and Kretschmer) referred in the recent bulletin. I call your attention especially to Equations 22–24 in our bulletin.

As regards the principle of similarity, it appears to me to be in the following state: This principle is correct; it must, however, be correctly applied, that is, it is necessary to take into consideration not only the similarity of the Reynolds number but also all other requirements of similarity. For instance, there is involved, through the roughness of the pipe, at least one more quotient of two lengths; namely, the mean height of the obstruction and of the diameter. Inasmuch as the obstructions are generally not geometrically similar, it is absolutely necessary to look for other parameters in order to take care of the roughness. Therefore, in practice it is very hard to compare the roughness of various pipes, and in this I see the fundamental reason for the differences in the results of various observers. It is necessary, for example, with gases, as Odquist and others have shown, to include another parameter consisting of two ratios, one the ratio of the mean velocity to the velocity of sound in the medium, and the other the ratio of the specific heat at constant pressure to the specific heat at constant volume. The difference in the observations on water and air, for instance, depend on these items which are not fully explained as yet. In a word, there is no doubt about the principle of similarity, but under the circumstances it is very doubtful whether similarity can be fully applied and taken into consideration.

We have in preparation a new and most interesting contribution from Dr. Witte which will be published in the first issue of the magazine *Technische Mechanik und Thermodynamik*, of which I am the editor. Witte's work deals with additional measurements at the I. G. Farbenindustrie on orifice plates and nozzles, and in this work the question of similarity is advanced forward a considerable step. The contribution will be published in the January and February (1930) issues of the new magazine.

Alton C. Chick.[27] The three papers presented in this symposium bring our attention very forcibly to the present method of analyzing fluid flow in closed conduits, namely, correlating it with the dimensionless ratio involving a linear dimension characterizing the shape, the velocity of flow, and the kinematic viscosity of the fluid, generally termed the Reynolds number.

The method used by Mr. Smith in showing how the Reynolds number is related to the flow distribution, by means of the nozzle and capillary tube in parallel, is very interesting. It gives the basis for the relationship and helps one to obtain a clearer understanding of the physical meaning of the Reynolds number itself.

Mr. Smith has succeeded in placing the venturi meter and the orifice meter on the same practical working basis by means of the Reynolds number and the acoustic ratio—two dimensionless ratios, and has given the results of a considerable number of published experiments to illustrate his method.

His development of the acoustic ratio lays down a new and very simple operator for use in the metering of gaseous fluids that promises to play as important a rôle as has the Reynolds number in metering viscous fluids. He has presented a simple straight-line correction formula for the effect of expansion, which ties his theory with present data. This has greatly simplified the mathematics involved in the commercial metering of fluids.

In appendixes Mr. Smith touches upon the basic theory of fluid flow, giving a clear discussion of viscosity and the law of dimensional homogeneity as applied to fluid flow.

Messrs. Beitler and Bucher have presented the results of practical tests of orifices in six-inch pipes which check with values of correction factors and coefficients obtained by Mr. Smith, thus further establishing the usefulness of this method of analysis.

Mr. Spitzglass has presented an interesting historical discussion of the development of some of our present hydraulic equations, but there appears to be a difference of opinion concerning the application of the principle of similarity to experiments with fluids, using small-scale structures. Mr. Spitzglass erroneously gives one to understand that the Reynolds number is not a criterion of similarity in the régime of turbulent flow, and that there is an effect or size (which in the case of a pipe, would be the diameter) which is unaccounted for when different sizes of similar models and prototypes are tested.

Actually in the case of closed conduits the use of the Reynolds number implies and requires that the model and prototype be geometrically similar in all respects including the roughness of the interior walls, and that as has been shown by several authori-

[26] Berlin, Germany. From a letter received by Mr. Spitzglass.

[27] Assistant Engineer to John R. Freeman, Providence, R. I. Jun. A.S.M.E.

ties,[28] the Reynolds number is the *only* criterion necessary for complete similarity. In the case of open channels, where the surface is exposed to the atmosphere, other factors are involved such as gravity and surface tension which bring into account other dimensionless ratios such as Froude's number, usually expressed as Lg/v^2, etc.[29] In the case of a closed pipe, if the interior roughness of a large pipe is not similar to that of a smaller-scale pipe which in other respects is geometrically similar, then it has been shown[29] by dimensional reasoning that if we could express the size of the irregularities by a simple linear quantity such as ϵ, then the coefficient would be not only a function of the Reynolds number but also a function of ϵ/d, where d is the diameter of the pipe.

We should not be led to condemn the validity of such a relationship as the Reynolds number simply because it may be impracticable commercially to construct structures of different sizes that will have their interior surfaces *exactly* geometrically similar. This apparent scale effect, the writer believes, is due only to the lack of precision in making the model and prototype truly similar, and this has been adequately discussed in the present paper by Mr. Smith. In his 1923 paper on the oil venturi meter, it was shown for the first time, the writer believes, that there was a slight scale effect on the coefficient, when plotted against the Reynolds number, resulting from the difference in velocity distribution in the pipe at the inlet and a difference in the loss caused by friction between the inlet and the throat, due to the roughness not being geometrically similar in the various sizes of pipes and throats.

When we do have geometrical similarity between the roughnesses in model and prototype then the coefficients will be identical, as Mr. Smith points out. This has been established by many experiments in European hydraulic laboratories which have been performed to test the validity of the laws of similarity.[30] Experiments by B. F. Groat,[31] one of the first in this country to make use of the laws of similarity in testing models of hydraulic structures, with models of ice-diversion structures for the St. Lawrence River, caused him to conclude that "there was nothing in the results of the experiments to indicate that they (the models) did not perform *exactly* as their prototypes."

Finally, the writer wishes to emphasize the necessity of having the approach and exit conditions at the model geometrically similar to the approach and exit conditions for the prototype in order to insure similarity of flow in the structure itself. If these conditions are not properly duplicated then the velocity distribution will be different, and an *apparent* scale effect will be found in the results.

All over the world today, and especially in Europe, a great amount of experimentation is being done on small-scale models in hydraulic laboratories. The principle of similarity applied to fluid flow is constantly being tested for all types of structures, for closed conduits as well as for open channels. The present papers are a notable addition to our knowledge of this subject.

ED S. SMITH, JR. The paper by Messrs. Beitler and Bucher shows evidence of high-grade experimental work, the essential data of which are confirmatory of previously established facts. In their paragraph referring to their Fig. 20, it should be remembered that the A.S.M.E. compressibility factor applies to nozzles having fixed non-expansible throats, and not to cases of orifice flow where the *vena contracta* itself is highly expansible. This point is discussed in the writer's closure to his paper which is a part of this symposium.

For example, their Fig. 20 for steam and the accompanying discussion are confirmatory of the writer's Fig. 2, presenting data from U. S. Bur. of Stds. R.P. 49 for air. The actual expansion factor Y for orifices is definitely established within small limits. Mr. Spitzglass similarly published Fig. 26 on page 324 of the discussion of Hodgson's 1928 A.S.M.E. paper,[32] which is confirmatory of Beitler-Bucher's Fig. 20. The deviation of the actual Y from the theoretical value Y_t of the expansion factor for orifices is highly important, and the writer welcomes its frequent mention.

R. Bachmann, of Darmstadt, in 1912 brought out data on this point. Hodgson's 1917 equation $C'' = 0.914 - 0.306\ P_2 / P_1$ for the coefficient C'' for orifices for gas (to be used with the theoretical expansion factor Y_t) clearly brings out the point that the theoretical expansion factor Y_t is excessive. There has been considerable progress since 1912 and the actual expansion factor Y for the usual concentric orifices having diameter ratios up to 0.6 (as well as for venturi tubes and nozzles) is at present known to within a few tenths of one per cent for any gaseous fluid with any usual differential pressure ratio.

The writer suggests the general use of the term "turbulence ratio" (symbol R) instead of the Reynolds number including cases where the orifice diameter is used instead of the pipe diameter. This term is highly descriptive and may well be abbreviated in actual use to the single word "turbulence."

The correction for the weight of air over a water column is also an established part of the writer's fluid-meter calibration and manufacture practice for precision fluid measurement. The writer has long dealt with fluid metering on the mass-flow basis instead of the weight-flow basis in which the values are affected by the weight of air. A complete sample computation involving air-weight correction factor would form a valuable addition to the appendix of the paper by Messrs. Beitler and Bucher.

Orifices of different ratios actually constitute different "shapes" and each shape necessarily requires calibration. In regard to the authors' statement in the third paragraph from the end of their paper, it is pointed out that the use of the Reynolds number in fluid flow requires that the shapes be geometrically similar. Data for wet steam would be an important extension of this work.

Mr. Spitzglass urges the need for caution in the application of the general method of similarity and the Reynolds number to fluid metering, especially in regard to its adoption as a law, for meters for various sizes of pipes.

In his 1923 A.S.M.E. paper,[33] "The Oil Venturi Meter," the writer published data for various sizes of venturi tubes having the same absolute roughness in which the coefficient was plotted against the Reynolds number, clearly showing the "scale" effect, the smaller venturi tubes having slightly lower coefficients than the larger. The correlation between coefficient and the Reynolds number for any given meter was definitely and unquestionably established not only for viscous streamline flow but also for turbulent flow. The scale effect was shown to be relatively small for the usual commercial sizes. This scale effect was clearly stated as being simply the slight departure of the shape, due to roughness, from exact geometrical similarity of otherwise similar venturi tubes. Since these venturi tubes had the same absolute roughness it is obvious that the smaller tubes were rela-

[28] Lindquist, Erik, "Am Modellregler eller Lekformighetssatser vid Vattenbyggnadstekniska Förösk" (The Laws of Similitude for Model Tests on Hydraulic Structures), *Teknisk Tidskrift*, Stockholm, 1925. Nos. 30, 34, and 43.

[29] See "Hydraulic Laboratory Practice," A.S.M.E., 1929, Appendix 15, also see "Die Grundlagen der Achnlichkeitsmechanik und ihre Verwertung bei Modellversuchen" (The Fundamentals of the Mechanics of Similitude and Its Use in Model Experiments), by M. Weber, *Jahrbuch der Schiffsbautechnsichen Gesellschaft*, 1919.

[30] See "Hydraulic Laboratory Practice," A.S.M.E., 1929.

[31] "Ice Diversion, Hydraulic Models and Hydraulic Similarity," by B. F. Groat, Trans. A.S.C.E., vol. 82 (1918), p. 1138.

[32] Trans. A.S.M.E., paper No. FSP-51-42.

[33] Trans. A.S.M.E., vol. 45 (1923), pp. 67–75.

tively rougher than the larger tubes, and indeed the effect of this slightly greater relative roughness was reflected in their slightly lowered coefficients.

The method of similarity, as used by the writer at that time, has come into increasing use, since it is an efficient method for correlating and using data: for any given size and shape of differential producer, the coefficient may be determined with any convenient fluid plotted against the Reynolds number, and then used with certainty for metering any other known fluid. Numerous experiments have established the fact that the deviations from exact geometrical similarity due to scale effect (relation between pipe size and relative roughness) are shown by the coefficient to be relatively slight—a mere "vernier" correction for the more usual differential producers. The use of the Reynolds number as a rigorous basis for correlating coefficients with turbulent and viscous flow has been established by the actual experimental work of Stanton and Pannell on friction loss in pipes, and by Hodgson, Daugherty, the writer, and numerous others, on various fluid meters.

The method of similarity (involving the use of the Reynolds number in determining flow) is not based upon any assumption. It is derived simply from the fact that certain variables are involved in fluid flow, and hence are known as "pertinent." From the simple obvious requirement that the dimensional units on each side of any true equation must be identical, a formula is obtained directly which merely has the pertinent variables in their proper relation, and the empirical dimensionless constant must be established by actual calibration. This formula is inevitably rigorous when it includes all of the pertinent variables, and no others.

For the flow of liquids in closed conduits, this method shows that the coefficients for differential producers of geometrically similar shapes depend only upon the Reynolds number when the pertinent variables are the velocity, density, and viscosity of the fluid and any corresponding linear dimension, such as diameter, of the conduit. Repeated indisputable calibrations with various liquids have established the fact that the coefficient is a function of the Reynolds number only, within the limits of experimental error. This proves that the Reynolds number includes all of the pertinent variables, and no others, affecting the coefficient, and directly confirms the fact that absolutely no assumptions are involved. This is not a mere matter of belief but of proof, i.e., *this method works*.

Roughness is an essential part of the shape of a differential producer; a basic element of the theory of similarity. The lack of an absolute measure of roughness has vexed hydraulicians up to the present, and even now constitutes the subject receiving the most consideration. The best expression of relative roughness which has yet been devised is probably simply the ratio of the height e of the rugosities to the pipe diameter d, thus e/d, which accurately takes account of the diameter effect for pipes of the same absolute roughness. However this ratio is not complete as a general basis since the shape of the rugosities and their pattern certainly affect the "shape" of the roughness.

First consider a situation where practically geometrically similar shapes produce substantially true hydraulic similarity, irrespective of size—in other words, the absolute size of the pipe does not appreciably affect the coefficient except as it enters the Reynolds number. Such a situation is not difficult to arrange, in fact it can be simply obtained, within commercially acceptable limits of error for the various sizes generally used, by using an orifice with flange taps and with velocity-equalizing "straightening vanes" similarly located upstream of the orifice, as Bean's recent experiments (referred to by Dr. Buckingham) clearly establish.

Now consider another situation involving orifices with the connections located one diameter upstream and at the *vena contracta*, as used by Spitzglass, where nominally "similar" orifices (of identical diameter ratios) in 4-in. relatively rough pipes have slightly higher coefficients for the same Reynolds number than in 8-in. relatively smooth pipes. Mr. Spitzglass maintains, the writer understands from his paper, (1) that hydraulic similarity does not (necessarily) exist in various sizes for equal Reynolds' numbers even though exact geometrical similarity does exist, and (2) that the coefficient is a function of the absolute size of the pipe (not of the shape, including the roughness). However, the writer would like to point out (1) that the absolute size of the pipe only enters in the Reynolds number and in the shape in so far as it affects the pipe's relative roughness, (2) that roughness is as truly a part of the pipe's shape as its general roundness, (3) that, consequently, small pipes are thus not exactly geometrically similar to large pipes having the same absolute roughness, and (4) that since the geometrical similarity is not exact in such small and large pipes, exact hydraulic similarity is not required by the law of similarity to exist with identical Reynolds' numbers.

Mr. Spitzglass seems to have overlooked the material on the "scale effect" in the closure of Mr. Hodgson's 1928 A.S.M.E. paper. Mr. Hodgson is in a position to have access to many full-scale calibrations of a variety of orifices upon numerous fluids.

Dye clouds passing through an orifice distinctly show the profiles shown in the writer's Fig. 3 or in Spitzglass' upper Fig. 10, which is labeled "incorrect." Spitzglass' lower Fig. 10, which is marked "correct," merely shows where turbulence exists. The writer suggests that a composite diagram, similar to his own Fig. 10, would be more satisfactory, showing the "dynamic streamlines" (mean flow direction), velocity-distribution traverses (mean velocity magnitude), with curls showing where eddies occur. At best any static diagram of transient flow phenomena is inadequate, slow-motion pictures such as Baron Shiba's A.S.M.E. film, showing the moving particles, are really required.

Finally, Mr. Spitzglass bases a "variation of coefficient with absolute size of the pipe" upon an assumed pipe-friction effect (incomplete boundary flow concept) which is not in accordance with the findings of Gibson[34] and others that for small and large smooth (practically geometrically similar) pipes the velocity distribution is the same for identical Reynolds' numbers.

Also, on Mr. Spitzglass' page 117, he seeks to base the variation of the coefficient with absolute pipe diameter upon Odquist's work, which was shown to be fallacious upon the writer's page 96.

Authors' Closures

S. R. Beitler and Paul Bucher. The authors are grateful to those who have discussed their paper and offered criticisms which may be useful in the continuation of this work. This continuation is proceeding at a very satisfactory rate, and it is hoped that the final results will be available for publication by the end of 1931. A much more complete paper covering the work done to date will appear as a Bulletin of the Engineering Experiment Station of the Ohio State University about the first of March, 1931.

For the information of Mr. Gess, condenser leakage tests were a matter of weekly routine in the laboratory.

Mr. Conlee's discussion on the location of the *vena contracta* is of much value. In the continuation of our work a change has been made in the method of taking the pressure and a smaller opening into the pipe is used. It was felt that this would give

[34] "The Principle of Dynamical Similarity, With Special Reference to Model Experiments," by A. H. Gibson, *Engineering*, March 14, 1924, p. 327. "In the smooth pipes, identical (velocity distribution) curves were obtained . . . when the Reynolds number was the same in each pipe, one being 4.93 and the other 7.40 cm. diameter."

more accurate results where the flat portion of the *vena contracta* was relatively short.

Dr. Moss's criticism of the orifice versus the flow nozzle is a question which is yet unsettled in the opinion of the authors. However, the fact that for commercial purposes it is necessary to use comparatively large-ratio nozzles, which Dr. Moss does not advocate, has some bearing in the matter. The additional data which have been collected but not as yet completely analyzed nor published cover both orifices and nozzles in all practical ratios and should help to settle this question.

Mr. Smith requests that a complete sample calculation involving the air-density correction to the differential head be added to the appendix. This request is denied at this time due to lack of time only and will be granted in the later publication already mentioned in this closure.

Ed S. Smith, Jr. The author's paper was presented in order to place the available information on fluid metering in shape to attract criticism so that the material would either be confirmed or rejected as a basis for the revision of the Fluid Meter Report.

Mr. Conlee's desire for data on meters on the different commercial pipe sizes is certainly shared by the "advocates of similarity." After all, similarity merely enables one to make the best use of data, since it permits the data for the various sizes to be placed upon a common rational basis.

Flange taps generally have been in better agreement with test results than have published data upon the *vena-contracta* taps, according to studies available to the author, and have the advantage that they are more flexible since it is not necessary to change them whenever an orifice is changed. Authorities fail to agree on the location of the *vena contracta:* while the Beitler-Bucher data agree closely with those of Johansen in the Hodgson paper,[35] the Spitzglass data would indicate that the *vena contracta* is 0.7 diameter from the orifice instead of 0.9 as Johansen shows for $d/D = 0.4$. Furthermore, at high differentials, Buckingham affirms that the *vena contracta* blows further downstream with gaseous fluids. The flange taps favor the use of the annulus which is ideal with concentric orifices in avoiding the effects of local pipe-surface irregularities and accidental eccentric installation of the orifices; of course, for the highest flows, segmental or eccentric orifices may be used with the simple flange taps with reliable results.

Although Mr. Conlee's use of the venturi expansion correction factor has been disproved for orifices, his use of it brings out the need for more published data upon high-ratio orifices with gaseous fluids. The acoustic ratio will permit the correlation of such data based upon various gaseous fluids so that this field will be most quickly covered.

The author concurs with Dr. Moss that the standard venturi tube or nozzle offers the highest accuracy which can be relied upon; however, for those who find it necessary to use orifices in metering gaseous fluids, the acoustic ratio was offered as a proper basis for the actual expansion factor Y, or for CY, which Dr. Moss suggests will prove more convenient. In correspondence with the author Dr. Moss independently confirms the validity of the acoustic ratio.

Mr. Gess confirms the aim of the paper as to its form, which was to place orifices and venturi meters upon a common basis with the simplest possible method of use, followed by adequate theoretical justification of this basis.

The author greatly appreciates Sprenkle's study and discussion, presenting the Beitler-Bucher orifice expansion-factor data for steam plotted upon the acoustic ratio x/n, which by its remarkably close agreement with the data on air from U. S. Bur. of Stds. R.P. 49, similarly plotted on the author's Fig. 2, strikingly confirms the use of this acoustic ratio.

Sprenkle's straight-line expansion factor for steam agrees with the data on air for similar taps, when plotted upon the acoustic-ratio basis—the average difference being only about one-tenth of one per cent. Bean's more recent data on natural gas definitely confirms the straight-line relation, and consequently the straight-line formula, when used with proper constants, as shown on the author's Fig. 4. The remarkably close agreement between the expansion factor Y data on steam, air, and gas by separate experimenters commends the work of all and also shows the validity of the acoustic ratio as the basis of correlation.

Ruppel,[36] in a recent paper on the effect of expansion on the *vena contracta* of orifices, presents an interesting computation of the expansion effect for orifices which checks very closely with both the air and steam data referred to above. The relation is seen to be very nearly a straight-line one, and the computation is along the lines of the author's Chapter VI, except that Ruppel goes further and computes the actual expansion of the *vena-contracta* diameter upon the assumption that its expansion also follows purely adiabatic laws. Ruppel computes values for both air and steam which are found to be practically

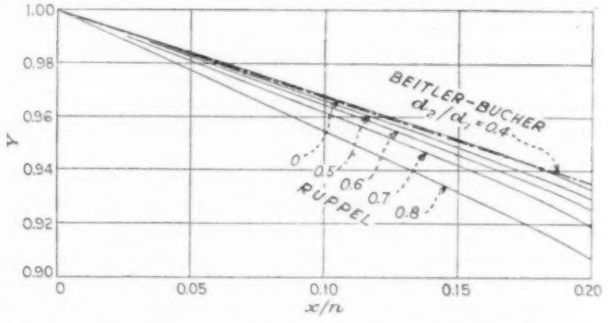


Fig. 2a For Throat Taps. Relation Between Expansion Factor Y and $1 - (P_2/P_1)^{1/n}$

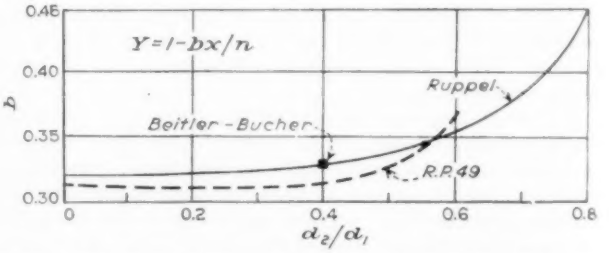


Fig. 4a For Throat Taps (Based on $x/n = 0.10$). Relation Between Expansion Constant b and Diameter Ratio d_2/d_1

identical when both are plotted against the author's acoustic ratio. His values and those of Messrs. Beitler and Bucher and U. S. Bur. of Stds. R.P. 49 are shown on the accompanying Figs. 2*a* and 4*a* to be almost unbelievably confirmatory of each other and of the validity of the acoustic ratio. Ruppel also uses the hydraulic formula (in a slightly modified form) and brings out the advantage that one may calibrate an orifice with water at corresponding Reynolds' numbers and then use the coefficient with air or steam with greater exactness than the coefficient can generally be obtained with equal ease on the gaseous fluid itself, confirming the author's method.

[35] Trans. A.S.M.E., paper No. FSP-51-42, p. 324.

[36] G. Ruppel, "Effect of the Expansion on the Contraction beyond Orifices." Berlin-Siemenstadt, Technische Mechanik und Thermodynamik, vol. 1 (1930), pp. 151–157.

In accurately metering hot fluids such as steam, while using a coefficient C obtained with a cold liquid such as water (and the proper expansion factor Y), it is necessary to consider the expansion due to temperature of the orifice plate or venturi throat, the area expansion in per cent at 500 deg. fahr. being as follows, other temperatures readily interpolated:

cast iron	0.48
cast steel	0.57
cast hard bronze	0.83
cast brass	0.88

Consideration of this expansion may serve to bring the experimental values for steam into closer agreement with those of air. This correction has long been used with venturi meters, see "The Venturi Meter as an Instrument for the Measurement of Liquids and Gases," by F. G. Gasche and R. S. Younglove, Illinois Steel Company, 1911.

The author's paper presents a method, not data, and this is clearly stated in the Introduction: "The data presented are purely for illustrating the use of this method." However, the author submits that his method shows data from reliable sources to be mutually confirmatory, and he welcomes the publication of such data. Sprenkle's Fig. 5 shows a lack of flatness of the coefficient at high Reynolds' numbers, which is foreign to the author's experience. Data plotted to such a large scale occasionally seems to indicate tendencies beyond the actual accuracy of the laboratory. Numerous coefficient graphs obtained by Pardoe at the University of Pennsylvania hydraulic laboratory, show that the coefficients are substantially flat at the Reynolds numbers covered by Sprenkle's data.

Sprenkle's Fig. 5 indicates that pipe roughness is really part of the shape of a differential producer, large pipes being generally relatively less rough than small pipes, and that it is consequently necessary to calibrate differential producers in the various commercial sizes of lines, since the Reynolds number only brings out flow similarity in geometrically similar shapes.

The author's experience on meters for wet steam and dirty fluids containing segmental orifices shows that while such orifices are generally excellent, they provide the basis for the author's remarks, which are confirmed by reference to Fig. 2551 of "Principles and Practice of Flow Meter Engineering," by the Foxboro Company. The author's first paragraph on "Dirty Fluids," page 101, comparing orifice and nozzle performance, seems adequately to cover Mr. Sprenkle's discussion on this point.

Sprenkle's formulas proposed for use with orifices are seen to be equivalent forms of the author's Formula [1], likewise intended for commercial use with both orifice and venturi meters. The author's Chapter VI presents an orifice analysis, like Chapter V for the venturi tube, which contains the heading that these two chapters are "merely illustrative material, offered as being of general interest." Thus, while the author's Equation [69] shows those interested that which occurs in flow through orifices, it was not recommended for commercial use. However, the great results which Ruppel has achieved through the extension of such an analysis more than justify him in its presentation.

In substance, Sprenkle's discussion seems to be generally confirmatory of the author's material.

Mr. Nicholas' historical summary could well be extended to include Helmholtz.[37]

As Mr. Nicholas observes, there is actually a "hump and a droop" of the coefficient in the author's Fig. 3, as confirmed by U. S. Bur. of Stds. R.P. 49 and Hodgson's Fig. 7 in his paper No. 31 of Mr. Nicholas' reference 21. The writer's "three" points represent the mean of a considerable number of tests with water in a 6-in. pipe.

Mr. Nicholas' physical interpretation of the kinematic viscosity as an acceleration is interesting but the dimensional units are L^2/T and not L/T^2, those of acceleration. In regard to the author's physical illustration of the Reynolds number as applied to flow similarity and coefficients, the physical reality seems more important than the mathematics, which is a mere tool that must be guided by the physical analysis. The history of hydrodynamics has repeatedly shown the limitations of purely mathematical analyses as opposed to those accompanied by insight into the actual physical situation.

Mr. Nicholas' criticism of the author's reference to the law of gaseous-fluid flow at the critical-pressure condition as Napier's relation seems unsustained in view of its use for the steam flow at this condition according to Marks' "Mechanical Engineers Handbook," and Hirshfeld and Barnard's "Heat-Power Engineering," both of which are representative of American practice.

The author prefers to use physically descriptive terms instead of proper names for engineering quantities as far as possible without departing seriously from current usage. If this critical-pressure relation should bear the name of Reynolds, as inferred by Nicholas, confusion with the well-known Reynolds number might result. However, the latter is being loosely used at present with the venturi-tube throat or orifice diameter instead of the pipe diameter. Hodgson prefers to call VDW/U the Rayleigh number, while B. F. Groat insists that it is properly the Stokes number. To avoid such confusion, the physically descriptive term turbulence (using the symbol R), which Nicholas used in a recent fluid-meter paper, is preferred by the writer, who proposed it in his 1923 paper on metering oil. In line with this reasoning, the author christened x/n the acoustic ratio.

In his discussion of the author's Equation [25], Mr. Nicholas failed to note that the author's J is a pure number, being the number of diameters the tube is long, and not the length in dimensional units.

Mr. Nicholas offers a suggestion that his viscous-flow equation, equivalent to the author's Equation [26], be included in the revision of the Fluid Meter Report. He suggests that it not only be corrected for velocity of approach as in the Hodgson reference, but it also be considered that the proper velocity-head with viscous flow is V^2/g instead of the usual $V^2/2g$.

As a matter of fact, the author nowhere in his paper has stated that the value of n is 1.135 for steam, although this value was formerly used generally for metering wet steam. All numerical values given throughout the paper are merely to illustrate the method, and the values of n from 1.135 to 1.67 were offered to cover any possible range of values. However, in answer to Mr. Nicholas' question, the proper value of n for superheated steam is unquestionably near 1.3 and this value also holds closely for wet steam, as established by the Callendar reference, while it is in substantially adiabatic rapid flow through venturi tubes, nozzles, or orifices. Although the performance of commercial steam meters, with n as 1.3, has been confirmed by weighed water tests, still more data are naturally to be desired. The remarkably close agreement of the Beitler-Bucher steam data with the U. S. Bur. of Stds. R.P. 49 air data is also confirmatory of the use of 1.3 for superheated steam. The use of various values of n for air recommended by various authorities is noted: 1.3947 for "normal" air, Moss; 1.40–1.402, Buckingham; 1.408 (the formerly generally accepted value), Hodgson.

Acknowledgment is gladly made to Mr. Hodgson for his correspondence containing the following simple derivation of the acoustic ratio as an approximation of the adiabatic-flow formula, which is an important independent verification of its validity:

"The adiabatic criterion for flow is

[37] See "Wissenschaftliche Abhandlungen," vol. I, p. 185.

$$\frac{(P_1/P_2)^{(n-1)/n} - 1}{(n-1)\,[1 - (d/D)^4\,(P_2/P_1)^{2/n}]} \quad \ldots\ldots [1]$$

By substituting

$$P_2/P_1 = 1 - h/P_1 = 1 - x \quad \ldots\ldots [2]$$

in [1] and expanding, the first term becomes

$$\frac{1}{1 - (d/D)^4}\,\frac{x}{n} \quad \ldots\ldots [3]$$

and when d/D approaches zero, or for any given shape d/D with geometrically similar meters, the approximate flow criterion is simply

$$x/n \quad \ldots\ldots [4]$$

which is the basis adopted by Mr. Smith, and termed the acoustic ratio.

"A further simple elastic-flow criterion, which will be found to be very suitable for correlating experimental data, may be deduced from the assumption that, for similar conditions for different fluids, the ratio of velocities V_1/V_2 must have the same value. This is equivalent to saying that the ratio of densities W_2/W_1 must be constant for each case compared; so that, if the expansion is adiabatic, the elastic-flow criterion may be written:

$$\left(\frac{P_2}{P_1}\right)^{1/n}$$

"Actually $(P_2/P_1)^{1/n}$ is the most accurate of the various approximate flow criteria listed.

"The criterion P_2/P_1 was used throughout the author's own paper because it was the simplest in the above list, and because it was found convenient to plot the various curves against P_2/P_1 as this term occurs in the general discharge formula."[38]

Comparison of Hodgson's $(P_2/P_1)^{1/n}$ with the author's acoustic ratio x/n shows that, within the usual commercial range of conditions, there is no appreciable difference in the results obtained from the use of these two operators. Only a few tenths of one per cent difference exists between expansion factors determined by each with the usual range of gases of $n = 1.2$ to 1.4 up to an acoustic ratio of $x/n = 0.3$ for concentric orifices of 0 to 0.75 diameter ratio and $x/n = 0.2$ for nozzles of 0 to 0.5 diameter ratio; and even when the range of gases includes monatomic gases so that $n = 1.2$ to 1.67, the corresponding acoustic ratios are 0.2 and 0.13, respectively, which still covers the usual commercial field satisfactorily.

Hodgson's $(P_2/P_1)^{1/n}$ is more nearly exact, is also dimensionally correct, and is based upon true similarity of flow following an isentropic expansion. Hodgson's concise paragraph, showing its derivation, clearly expresses the essence of flow similarity for expansible gases. However, this operator was not followed by the author since the simple acoustic ratio x/n is certainly more convenient and sufficiently accurate for all commercial metering needs.

Following a modified Maclaurin's expansion:
$(P_2/P_1)^{1/n} = (1-x)^{1/n} = 1 - x/n - 0.6\,(n-1)(x/n)^2$ (almost exactly) $= 1 - x/n$
closely enough for practical metering.

The acoustic ratio x/n may be satisfactorily modified as follows, if unusually high precision should be required:

$$1 - (P_2/P_1)^{1/n} = x/n + 0.6\,(n-1)(x/n)^2$$

and this $1 - (P_2/P_1)^{1/n}$ used as a basis as in Figs. 2*b* and 4*b* which show Ruppel's calculated values and the Beitler-Bucher steam data for orifices.

The acoustic ratio must not be used blindly for all types of expansion, since it is based upon the ratio of the velocity of the fluid to that of sound in the fluid, and can hold rigorously only when this ratio properly includes the pertinent variables. The velocity of a sound wave depends upon isentropic compressions and expansions and for the acoustic ratio to be theoretically correct it is necessary for the velocity of the fluid similarly to depend upon isentropic compressions and expansions, as is actually substantially the case with flow through nozzles and orifices with flange or throat taps. For pipe taps which use the loss

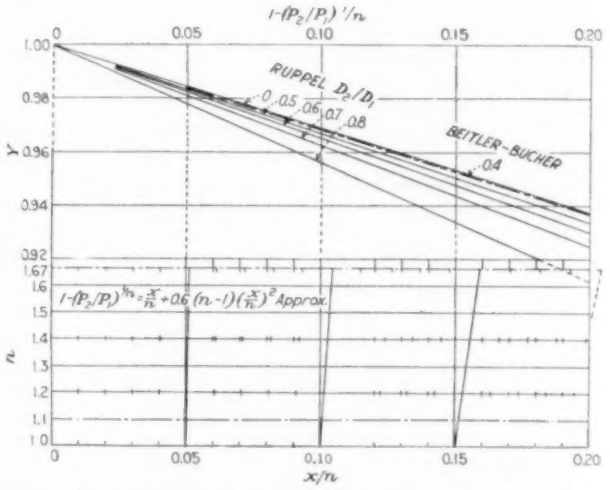


Fig. 2*b* For Throat Taps. Relation Between Expansion Factor Y and $1 - (P_2/P_1)^{1/n}$

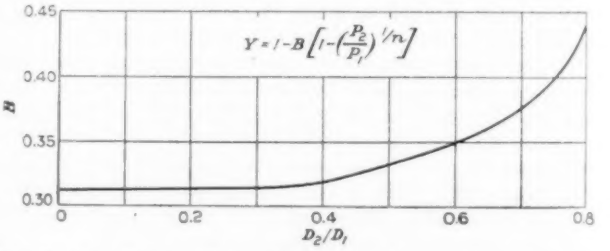


Fig. 4*b* For Throat Taps (Based on $x/n = 0.10$). Relation Between Expansion Constant B and Diameter Ratio D_2/D_1

across the orifice, there is a question as to whether this applies, since this differential is practically isothermal for a perfect gas, being substantially dissipative. For non-isentropic expansions, such as the isothermal, the expansion factor no longer depends strictly upon the acoustic ratio x/n but rather upon the separate ratios x of pressure and n of the specific heats as confirmed by Dr. Edgar Buckingham. However, for the pipe taps the author considers that the acoustic ratio x/n should still be significant and probably adequate since it controls the expansion of, and hence the actual diameter of, the *vena contracta* which, according to Borda's law, chiefly determines the overall loss. The author anticipates that an extension of Ruppel's calculations for this case would be closely checked by experimental data for pipe taps with various gases. Such data would be most welcome.

Much of the author's paper deals with an expansion correction for gaseous fluids. U. S. Bur. of Stds. R. P. 49 on orifices shows that the correction for orifices is small when calculations are based upon the mean pressure $(P_1 + P_2)/2$. Consequently,

[38] "The Laws of Similarity for Orifice and Nozzle Flows," by John L. Hodgson, Trans. A.S.M.E., paper No. FSP-51-42, formula [16], p. 313.

it may be of interest that the author some time ago conceived the idea of having the flow directly produce the proper pressure for use in the hydraulic equation. This entirely eliminates the necessity for any expansion factor in commercial metering (being indeed more accurate than the mean-pressure method above). With orifices, this simply involves the use of an impact-type tube for the pressure, while with venturi tubes the "square-root" pressure tap is taken at a slightly reduced section immediately downstream from the standard throat. For any commercial differentials, the deviation from "square-rootness" with such "square-root" orifices and venturi tubes is less than the experimental error of a tenth or so of one per cent, as determined in recent tests by the author.

The "square-root" gas meter has the important advantage that it can be compensated precisely for various pressures (really densities) since the instrument does not contain an expansion factor (such as Mr. Conlee mentions), which can only be exactly correct at the single pressure for which the meter is graduated, and must necessarily be inexact as the pressure departs therefrom.

P. J. Kiefer, professor of mechanical engineering at the U. S. Navy Postgraduate School, in private correspondence, agrees with the author upon the use of the Reynolds number as follows:

> Relative to the propriety or reliability of the use of the Reynolds number as a parameter, which practice is so vigorously attacked by Spitzglass and Pardoe, I agree with those gentlemen only to the extent that, like any other physical relation, it must be used with reasonable discretion. However, in view of the truly amazing results which have been obtained by means of the dimensional theory and the Reynolds number in correlating the flow of wide varieties of fluids through pipes and in predicting full-scale performance from model tests, I can go no further with them and am wholly in agreement with Smith's use of that number as one of his parameters.

Dr. Edgar Buckingham's oral discussion of similarity and the Reynolds number at the Symposium was appreciated by the author; his corroboration at this time being most helpful, also his generous correspondence with the author did much to eliminate errors and increase the usefulness of this paper.

Prof. R. L. Daugherty's clear remarks at the same time, confirming the use of the Reynolds number with any particular differential producer for a variety of fluids, was welcome as confirmation of the basis proposed by the author.

While Professor Pardoe and Mr. Spitzglass at the Symposium failed to agree completely with the author, he acknowledges the exceptional value of their remarks which have done a great service in forcing clearly worded expressions from the advocates of similarity that will aid immeasurably in assisting the general understanding of the subject.

Mr. Slocum, in his printed discussion, gives the relative velocity criterion for similarity as $V = v\sqrt{L/l}$, which the author has shown to enter where an open surface exists, as with a ship or an open conduit, and not to exist for a flow in a closed conduit of fixed shape. Slocum's remarks on the fundamental difference between flow in a conduit and the passage of a ship through open water agree with the author's paragraph in Appendix 3 under "Physical Description of Flow Phenomena." That eddy-flow similarity exists at identical Reynolds' numbers was early experimentally demonstrated, see Report of Advisory Committee for Aeronautics, 1911–12, p. 97.

Dr. Witte's confirmation of the use of the Reynolds number is invaluable as to the criterion for the similarity with liquids and low differential gas measurement; however, the author has extended the range of similarity into the high-differential régime of gaseous flow by the use of the acoustic ratio as an additional factor. The recent advance in this subject, as set forth in the author's paper, is clearly evidenced by Witte's comment that "for larger expansions (differentials in gas measurement) there is nothing definite that the law of similarity can show about the coefficient and in fact the coefficient is different at larger differentials."

Dr. Jakob's statement as to the work of Odquist and others also confirms the advance in the practical metering of gaseous fluids which the author's paper presents in the simple acoustic ratio.

Mr. Chick's invaluable assistance in the considerable work involved in preparing and checking the author's paper has not been adequately acknowledged. His remarks in the discussion clearly bring out the essential points of the Symposium, especially as regards correcting the erroneous concept of an "absolute" size (pipe diameter) effect, clearly showing it actually to be due to inexact reproduction of geometrical similarity in the various sizes.

Mr. Pigott, as chairman of the Fluid Meters Committee, has been exceptionally generous throughout in his assistance and guidance, which was instrumental in bringing this paper into the shape that would be most useful in the subsequent revision of the Fluid Meter Report—Part One.

J. M. Spitzglass. Mr. Slocum very aptly introduced the general principles of dynamical similarity in his discussion. Furthermore, he has shown that there *are* exceptions to the law of similarity in its general application and that there *are* limitations to be considered also in its particular application to fluid flow in closed conduits.

Dr. Witte, on the other hand, believes that ". . by adhering to the given assumptions one must absolutely demand the same dimensionless coefficient for equal values of Reynolds' function in the case of restricted openings."

Dr. Jakob remarks that ". . there is no doubt about the principle of similarity, but under the circumstances, it is very doubtful whether similarity can be fully applied and taken into consideration."

Mr. Chick insists that ". . in the case of closed conduits . . the Reynolds number is the *only* criterion for complete similarity." He further maintains that the author erroneously gives one to understand that the Reynolds number is not a criterion of similarity in the region of turbulent flow.

The author does not "condemn the validity of such a relationship as the Reynolds number." Its value as a parameter for correlating experimental data is well established. Thus far in the discussion he is the one who pointed out the only known case where dynamical similarity and particularly the Reynolds number applies directly without exception and that is the law of Poiseuille for capillary tube discharges. With all due respect to the Euler-Stokes fundamental equations, one finds that their assumptions do not completely cover the full relationship of dynamical similarity which is exhibited in the law of Poiseuille.

Beyond the region of laminar flow the characteristics of fluid motion become much more complex. When full turbulence is reached the Reynolds number loses its significance in the determination of the coefficient. It is paradoxical to state that the Reynolds number is the *only* criterion for similarity in the region of turbulent flow and at the same time assert that in that region the coefficient remains practically "flat" for all values of the Reynolds number. The fact is that the coefficient is not flat but varies with the size of the pipe. How much this variation is can be determined only by experiment.

HYD-52-8

The Kaplan Adjustable-Blade Turbine

By B. E. SMITH,[1] YORK, PA.

In this paper the author gives details of construction of the Kaplan adjustable-blade hydraulic turbine, and tells of the advantages obtained by adjusting the blades and the wicket gates simultaneously. This turbine differs from the ordinary wicket-gate turbine in the shape of the top plate and the construction of the runner and shaft. The top plate is shaped so as to form a transition space free from vanes between the wicket gates and the runner, in which the direction of flow of the water is changed from radial to axial. The runner hub carries the movable blades, and the operating connections for moving the blades are inside the single-piece hub casting.

THE FIRST known instance of an attempt to use an adjustable-blade turbine is evidenced by the United States Patent No. 67,994, issued to Ludlow in 1867. In this design only the blades were adjustable. There were no water-control gates on the turbine, and stationary guide vanes were used.

About the time the Ludlow patent was granted, there were also many patents issued showing axial-flow runners with adjustable gates of the cylinder, wicket, and other types. Apparently, however, the advantage to be obtained by adjusting the blades and the gates simultaneously to maintain a predetermined relationship was not realized, and no attempt was made to design or build such a turbine. The first application of this principle was made, about 1912, by Dr. Victor Kaplan, of Brünn, Czechoslovakia, who filed patent applications in Europe about 1913 and in the United States in 1914. Dr. Kaplan conceived the ideas and carried on the investigations and tests which led to the development of the modern high-specific-speed propeller-type runner. During these laboratory tests he discovered the marked advantages to be obtained by adjusting the blades and the wicket gates simultaneously. The historical development of the Kaplan turbine is fully set forth in the article written by himself, "Die Entwicklung des Kaplan-Laufrades," in the "Wasserkraft Jahrbuch," 1927–1928, pp. 414–424.

Due to the disturbance in business conditions caused by the war, apparently no attempt was made either in Europe or in this country to make use of the Kaplan designs in a practical way until about 1920. Shortly after this date, a European builder entered into a contract to furnish two turbines with automatically adjustable blades operating under a head of 20 ft. and developing 1100 hp. each. This installation was rapidly followed by others of all capacities and operating under various heads. One European builder alone has furnished 85 turbines of this type, developing a total of 187,490 hp. under heads of from 7 to 48 ft., and in sizes up to 40,000 hp in one unit. During this same period the same manufacturer has built only 19 fixed-blade propeller-type turbines with a total capacity of 79,000 hp. At the present time, European practice is to install turbines with automatically adjustable blades in practically every instance where the head conditions are suitable.

Propeller-type turbines were introduced in the United States about 1917, but it was not until 1923 that the first adjustable-blade turbine was installed in this country. This unit had to be shut down and the flume unwatered in order to make the adjustment of the runner vanes, which were bolted to the runner hub. In 1925 an adjustable-blade turbine was installed with an arrangement whereby the runner vanes could be adjusted from the turbine pit when the unit was shut down, without unwatering the flume. Since that time several units of this type have been installed in this country.

In 1927 an American manufacturer acquired the exclusive rights to the Kaplan United States patents for turbines which permit the automatic simultaneous adjustment of the runner vanes and wicket gates while the unit is in operation. The first automatic adjustable-blade turbine was sold in 1928, and this order was rapidly followed by other contracts. Since that time this company has sold 16 adjustable-blade units operating under heads of from 10 to 40 ft. and developing a total of 37,366 hp. Table 1 gives the details of these installations.



FIG. 1 CONSTRUCTION OF THE SINGLE-PIECE HUB CASTING

The adjustable-blade Kaplan turbine is constructed in the same manner as an ordinary wicket-gate turbine, except for the shape of the top plate and the construction of the runner and the shaft. The top plate is shaped so as to form a vane-free transition space between the wicket gates and the runner, in which the direction of flow of the water is changed from radial to axial. The construction of the runner hub, which carries the movable blades, requires special attention, due to the comparatively small space available. All of the operating connections for moving the blades must be contained inside the single-piece hub casting (see Fig. 1), and must be of such form that they will operate

[1] Secretary, S. Morgan Smith Co. Jun. A.S.M.E. Mr. Smith was graduated from Cornell University in 1239 with the degree of Mechanical Engineer, and was for the next two years associated with the Georgia Railway & Power Co., engaged in the design and field construction of hydroelectric plants. Since 1926 he has been connected with his present concern, engaged in the manufacture of hydraulic turbines.

NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

TABLE 1 SMITH-KAPLAN TURBINE INSTALLATIONS

Company, plant, and state	No. of units	Date purchased	Head, ft.	Speed, r.p.m.	Hp. per unit
Metropolitan Edison Co., York Haven, Pa.	1	7-19-28	26	200	2970
Central Power & Light Co., Devils River No. 9, Tex.	1	8-29-28	33	277	1900
Wisconsin Power & Light Co., Rockton, Ill.	1	10- 8-28	10½	100	750
Nova Scotia Power Comm., Tusket Falls, N. S.	3	11- 1-28	20	225	1052
Utilities Power & Light Co., Valentine, Neb.	1	4-22-29	29	600	320
Wheatsworth, Inc., Hamburg, N. J.	1	5-17-29	29	600	310
Metropolitan Edison Co., York Haven, Pa.	3	7- 8-29	26	200	2970
Connecticut River Development Co., McIndoes, Vt.	2	9-17-29	29	150	5120
Utilities Power & Light Co., Lebanon, Mo.	2	10- 7-29	40	327	2030
San Joaquin Light & Power Co., Merced Falls, Cal.	1	11-22-29	26	128.6	4750

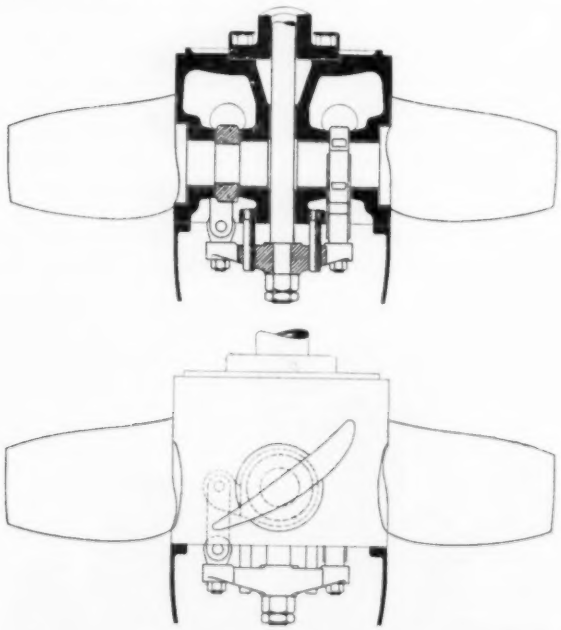


FIG. 2 PRINCIPLE OF OPERATION OF THE KAPLAN ADJUSTABLE-BLADE PROPELLER RUNNER

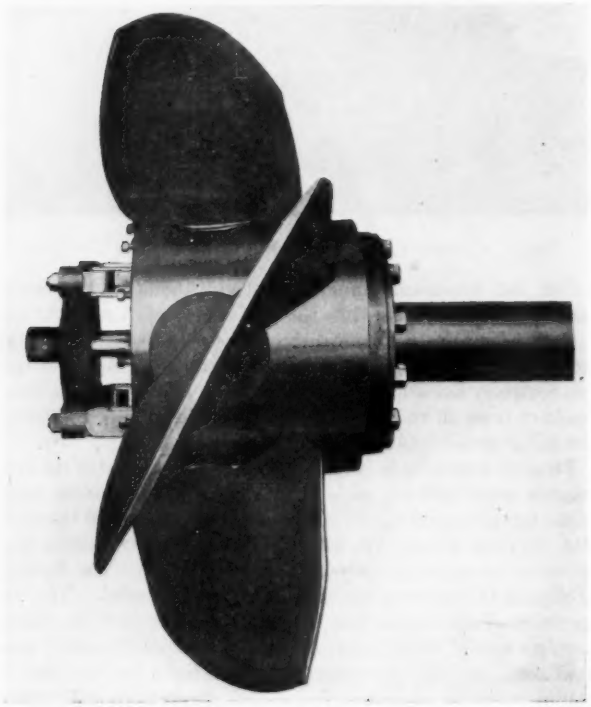


FIG. 3 RUNNER WITH BLADES IN THE OPEN POSITION

indefinitely under the most severe service requirements, without any adjustments. By very carefully utilizing every cubic inch of space, a design has been developed which has proved eminently satisfactory in the many plants already built and in operation. The stresses are held within conservative limits to guarantee freedom from excessive strains and deformation, and to reduce the possibility of wear to the absolute minimum. Careful inspection of the Kaplan turbines already in operation for several years shows no signs of wear.

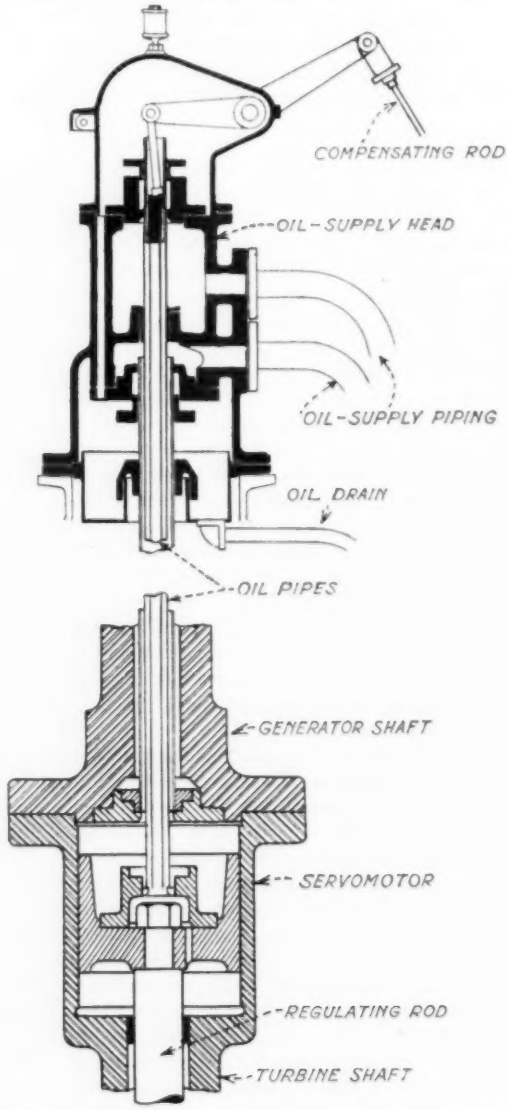


FIG. 4 SECTION THROUGH OIL-SUPPLY HEAD AND THE RUNNER-BLADE SERVOMOTOR

The hub is bored out and bushed with bronze to receive the trunnions of the runner blades. A crank is keyed to each trunnion and clamped to secure the blade against lateral displacement. These cranks, which stand normal to the shaft axis in their mid-position, are connected to a sliding cross by means of links and eyebolts. The cross is carried on a sliding regulating rod which

operates inside the hollow turbine shaft. This regulating rod is guided in bronze bushings and moves axially in the shaft to move the blades through the agency of the sliding cross, the links, and the cranks. The eyebolts and links are bushed with high-grade bronze. The cross slides on keys, which prevents any motion except in an axial direction.

Fig. 1 shows an end view of the Kaplan adjustable-blade runner. Fig. 2 shows a cross-section through the hub of the runner and an outside view of the hub with the lower section removed. Fig. 3 shows a runner with the blades in the open position.

In order to provide constant lubrication for the moving parts in the hub, it is filled with a heavy-bodied oil, which is supplied through the hollow turbine shaft through openings located at a convenient height for refilling. Each runner-blade trunnion is packed against leakage where it goes through the hub body, so that oil loss due to leakage is eliminated. This packing is arranged so that it, in combination with the inside oil pressure (partly static due to the head in the hollow shaft and partly in centrifugal head due to the rotation), prevents the seepage of water and sand or other impurities into the hub. A brass pipe plug is provided in the lower section for draining the oil if necessary.

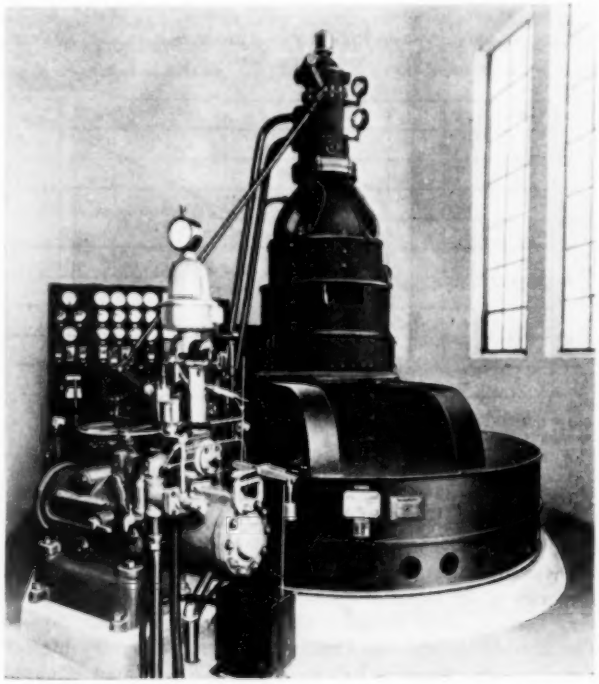


FIG. 5 GENERATOR WITH OIL-SUPPLY HEAD AND OIL PIPING IN PLACE

The runner vanes and hub are made of cast steel. Great care is used to finish the vanes to templets, so that they will have the correct form and offer the least resistance to the flow of water through the turbine.

The regulating rod in the hollow turbine shaft is moved axially by a servomotor arranged between the flanges of the turbine and generator shafts. The piston is connected directly to the upper end of the regulating rod. The servomotor is operated by oil supplied under pressure from above, through two pipes arranged concentrically within the hollow generator shaft. The inner pipe carries oil to the lower side of the piston and is rigidly connected to it, moving up and down with the piston. The governor-compensating mechanism is connected to the upper end of this pipe. The two pipes to the servomotor terminate in two distributing chambers arranged one on top of the other, and mounted on the upper end of the generator shaft. Special provision is made to prevent any leakage of oil over the generator. Fig. 4 shows a section through the oil-supply head and the runner-blade servomotor, and Fig. 5 shows a generator with the oil-supply head and oil piping in place.

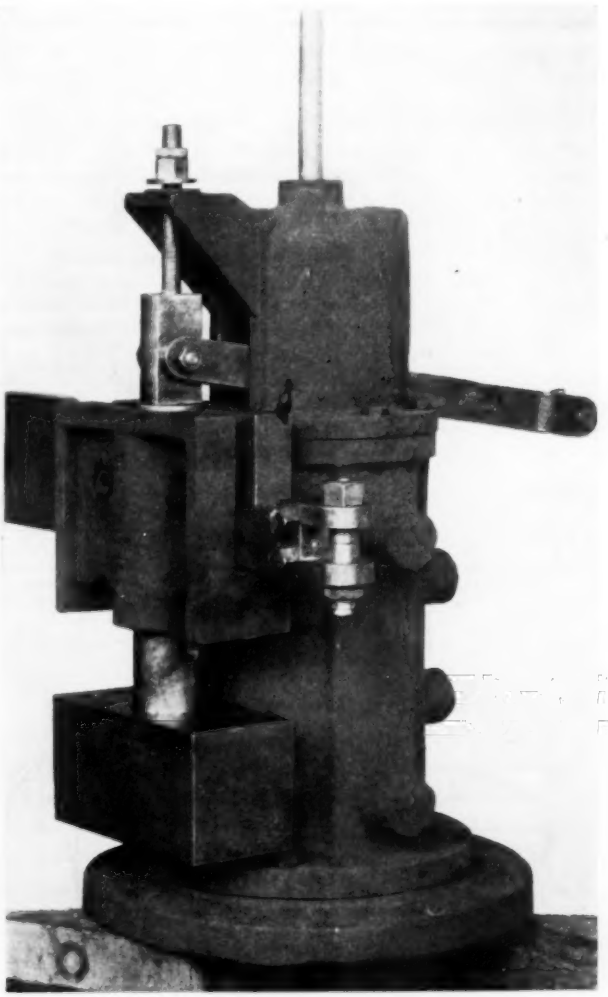


FIG. 6 DISTRIBUTING VALVE AND CAM

The governor itself is a simple speed governor, which operates the wicket gates in the usual manner, either through a vertical gate shaft or through servomotors and push-and-pull rods connected to the turbine-gate shifting ring. In addition, this governor has a second distributing valve controlling the flow of oil to the runner-vane servomotor in the turbine shaft. A cam is connected to the gate-operating mechanism and to the runner-blade compensating connections in such a way that the gates and runner blades always move in the definite relationship required to give the maximum efficiency for every variation in load. Fig. 6 shows this distributing valve with the cam in place.

The simultaneous adjusting of the runner blades and wicket gates maintains the best conditions of flow through the turbine throughout the entire range of output, thus giving exceptionally high part-load efficiencies and consequently very flat efficiency-horsepower curves. Since the blade and gate angles and openings always have the correct relationship to each other, the automatic

adjustable-blade turbine always operates with maximum smoothness, and with minimum vibration and pitting.

Fig. 7 shows a comparison of the efficiency-load curves of a Kaplan automatic adjustable-blade turbine with those of fixed-blade turbines at various heads and with different runner-blade angles. The dotted curves show the performance of a fixed-blade runner at various heads, and the dash-and-dot lines show the results obtained at 26 ft. head when three different fixed-blade runners are used, one in which the blades are inclined at

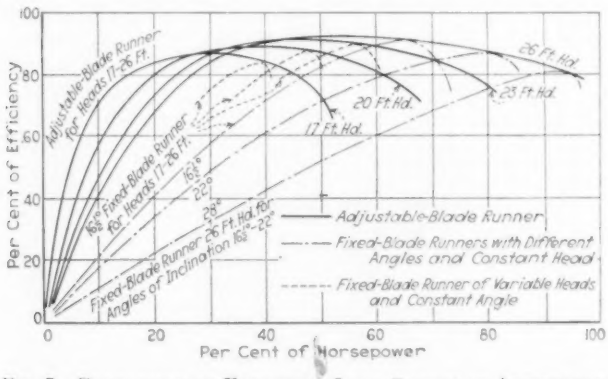


Fig. 7 Comparison of Efficiency-Load Curves of Adjustable-Blade With Fixed-Blade Turbine

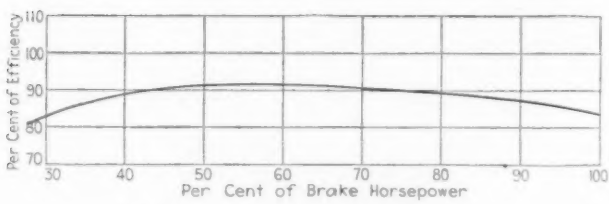


Fig. 8 Typical Efficiency-Load Curve

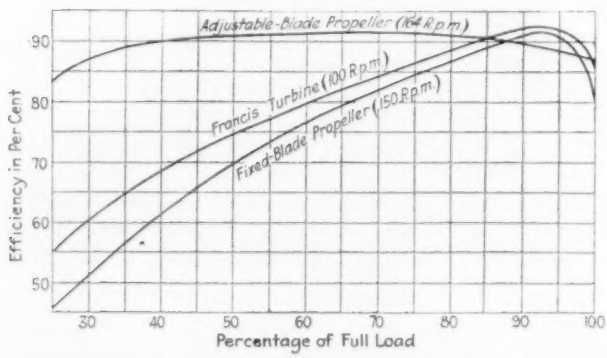


Fig. 9 Comparisons of Efficiencies of Adjustable Blade, Fixed Blade, and Francis Turbines

28 deg., another at 22 deg., and the third at 16½ deg. An infinite number of curves of fixed-blade runners with different blade angles will develop the envelope curve shown by the full line for 26 ft. head. Automatic adjustment of the blades and gates maintains the correct relationship between them to secure the highest efficiency for each blade position. Four such envelope curves have been drawn, using full lines, showing the horsepower and efficiency of an automatic adjustable-blade turbine under heads of 17, 20, 23, and 26 ft. These curves show how the adjustment of the blades holds the efficiency of the turbine up to the maximum which could theoretically be obtained by using a different fixed-blade runner for each special condition of load and head.

A study of this figure and a consideration of the operating features involved clearly show that, as a practical matter, it is impossible to obtain even an approximation of the envelope curves with a hand-operated adjustable-blade runner requiring a shutdown for each adjustment. In addition to the necessarily uneconomical operation of the non-automatic turbine, there is involved a substantial amount of increased physical labor and attendance, and also loss of power during the periods of shutdown required to adjust the blades when variations in load or head occur.

Fig. 8 shows a typical efficiency-load curve of a Kaplan automatic adjustable-blade turbine with a specific speed of 161. The curve is very flat, with more than 85 per cent efficiency from 33 to 96 per cent load and more than 90 per cent efficiency from 43 to 75 per cent load.

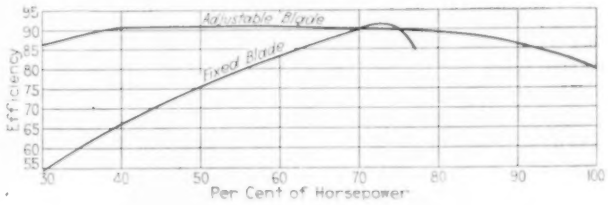


Fig. 10 Comparison of Fixed- and Adjustable-Blade Turbines of Same Size

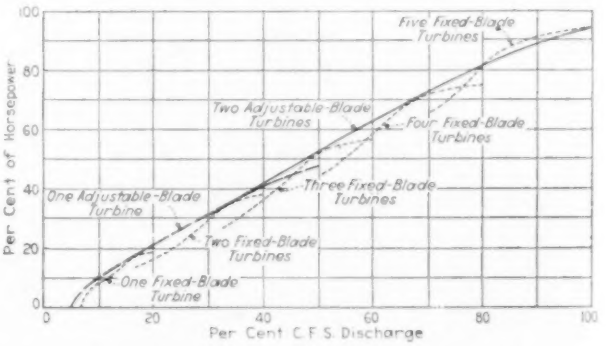


Fig. 11 Comparison of Load and Discharge Curves of Adjustable- and Fixed-Blade Turbines

Fig. 9 shows a comparison of a Kaplan automatic adjustable-blade turbine and a fixed-blade turbine with a high-specific-speed Francis turbine. This figure shows the marked advantage in efficiency of the adjustable-blade turbine at all loads except in a very narrow range at about 90 per cent of full-load capacity. The fixed-blade and the Francis turbines attain very high efficiencies only at nearly full load, and the part-load efficiencies drop off very sharply. Also, for the same full-load capacity, a smaller adjustable-blade turbine can be used, and consequently it can be operated at a higher speed, thus permitting the use of a smaller and cheaper generator. The higher the specific speed for the fixed-blade and the Francis turbine, the faster the efficiency drops at part load. Therefore, with high specific speeds the efficiency of these turbines attains satisfactory values only within a small range of regulation. Consequently such turbines can be considered only where they will be operated with loads varying over only a small range at near full capacity.

The Kaplan adjustable-blade turbine has a considerably larger capacity than a fixed-blade turbine of the same size and speed. Fig. 10 illustrates this difference very clearly. This feature permits the automatic adjustable-blade turbine to be designed and selected to provide a liberal overload capacity. The turbine is very flexible in this respect, and the excess horsepower for peak

loads or for low-head flood conditions can be provided without affecting the normal head economy, and usually with little or no reduction in efficiency at small gate openings.

Fig. 11 shows the possibility of reducing the number of units in a power house without reducing the total capacity, by using adjustable-blade instead of fixed-blade runners. This curve shows that two automatic adjustable-blade units could be substituted for five smaller fixed-blade turbines, and the efficiency at any discharge would be higher than if fixed-blade units were used. The use of the two adjustable-blade units results in material reductions in the cost of the mechanical and electrical equipment, and also reduces the size and consequently the cost of the excavation and of the power house itself.

The adjustable-blade turbine is particularly suited for developments where the quantity of water varies greatly. On account of the high part-load efficiencies, these turbines deliver much more power than fixed-blade axial-flow or Francis-type turbines in times of low water. Thus the total annual kilowatt-hour output available is greatly increased. The realized increase is greater than the computed gain, as in actual plant operation the Francis units are not started and stopped in accordance with the individual efficiency curves.

Fixed-blade axial-flow turbines, on account of their narrow operating range at high efficiency, can be considered only in comparison with adjustable-blade turbines when they can be constantly operated at nearly full load. This condition applies only where such machines are used to carry base load and are tied in electrically with a large system so that other units may be used for regulating purposes. Where there is a large quantity of water to be handled, a number of fixed-blade turbines may be installed and only as many units operated as the immediate quantity of water justifies. Even in this case the total yearly output can be increased by using one or more automatic adjustable-blade turbines for regulating purposes.

As a class, Kaplan turbines are essentially low- and medium-head machines; however, the head under which they may be used is constantly increasing and heads up to 66 ft. may now be developed with perfect safety.

Adjustable-blade turbines are placed in settings similar to fixed-blade axial-flow or Francis-type turbines; however, with these machines the allowable draft head decreases with increasing head and increasing specific speed. Consequently this feature must be carefully considered in designing the setting, or uneven operation and also pitting may result in the turbine. The design of draft tube also affects the operation of this runner. It has been found that the elbow-type draft tube is best suited for all units except small ones where the cost of excavation permits the use of a straight conical tube. A properly designed elbow-type draft tube reduces excavation to a minimum, provides maximum support for the superstructure, simplifies and cheapens the form work, has no underhanging parts requiring expensive reinforcing, and enables the turbine to develop maximum efficiency.

Progress in the Iron and Steel Industry in 1929

Contributed by the Iron and Steel Division

Executive Committee: C. Snelling Robinson, *Chairman*, F. C. Biggert, Jr., Walter Trinks, and W. W. Macon

TECHNICAL progress in the iron and steel industry in 1929 has been rather unusual, and there is every indication that much greater advances will be made from time to time because of the intensive study that is being prosecuted in our research laboratories and the practical application of the results thus obtained. In order for steel companies to keep in a fairly competitive position it is necessary to make a determined effort to utilize all of the technical developments possible. This is no time to be satisfied with present processes, nor to take the attitude that there is no further improvement to be made.

While, in general, the selling prices of materials have not greatly increased, earnings have been much better owing to economies which have been effected in the course of production and to the careful attention given to methods used. Much of the decrease in the cost of production has been brought about by large investment in new machinery and plant facilities. The output of steel products has increased greatly, largely due to the same causes rather than to the building of new mills.

The tendency toward mergers with a view to decrease in over head and to general economies of production has been evident in the iron and steel industry as well. Among the important moves of this kind may be mentioned the following. For the first time in many years the U. S. Steel Corporation has absorbed an existing steel mill by taking over the Columbia Steel Company with plants at Pittsburgh, Calif., and elsewhere on the Pacific Coast, apparently in an effort to provide that growing region with locally made material. A merger of the A. M. Hanna properties, the Weirton Steel Company at Weirton, W. Va., and the Great Lakes Steel Corporation with a plant under construction in the Detroit region, has resulted in the formation of the National Steel Company. The Central Alloy Steel Company, formed some two years ago through a merger of the Central Steel Company, of Massillon, Ohio, and the United Alloy Steel Company, of Canton, Ohio, has absorbed the Interstate Iron and Steel Company in Chicago, which makes it one of the most important alloy-steel producers in the country. The control of the Donner Steel Company has passed to Cleveland interests headed by C. S. Eaton, and this has been followed by an amalgamation of the Donner Steel Company with the Witherow Steel Company. The Republic Iron and Steel Company first acquired the Steel Tubes Company, which gave it control of important patents for the manufacture of electrically welded pipe, for which purpose a new plant has been erected at Warren, Ohio. This was followed by the acquisition of the properties of the Union Drawn Steel Company, in Massillon, Ohio, Beaver Falls, Pa., and elsewhere. The papers have repeatedly announced that a tremendous merger was under way involving, according to some, the Republic, Youngstown, and Inland companies, while others included with these the Otis Steel, Midland Steel Products, and even the Bethlehem Steel Company. No evidence is available to show that any such undertaking is actually being pushed. In the tool-steel field, likewise the Ludlum Steel Company has absorbed the Atlas Steel Corporation of Dunkirk, N. Y., with a subsidiary in Canada. This makes it one of the largest units in its field in this country.

Of late considerable attention has been called to the Detroit district and to the large development which is taking place in that section.

Most of the nine months of 1929 were record-breaking ones in nearly all departments of the steel industry. The production of pig iron during that period totaled 32,649,182 gross tons, or an average per month of 3,627,687 gross tons, which means that were this rate to keep up for the remainder of the year the total would run well over 40,000,000 tons, compared with 37,800,000 in 1928 and 39,100,000 in 1926, a record year. The average production for the first nine months was about 85 per cent of capacity as computed by the American Iron and Steel Institute.

The nine months' production of steel ingots totaled 43,243,404 gross tons, which is also a record tonnage for a like period, and the average per month for the nine months was 4,804,822 gross tons. The production average was about 92 per cent, based on the capacity computed by the American Iron and Steel Institute.

Iron Ore—Coal and Coke

There is little doubt that 1929 will set a new high record for ore shipments from the Lake Superior region. Shipments to October 1 totaled 53,264,927 gross tons, with shipments during the months of June, July, and August being over 10,000,000 tons per month. At this record-breaking rate it is expected that 1929 shipments will exceed 67,000,000 tons as against the record set in 1916, of 66,000,000 tons.

There has been a tremendous development of interest in ores during the past year: more exercising of options on both high- and low-grade ore properties, particularly during the last few months, than there has been in many years. There has been more interest in concentration of ores in the last year than for many years; and there is an increasing appreciation of the vital necessity of ore reserve; and, because of these factors the ore situation is becoming more firm at home and abroad, and beneficiation of the low-grade ores will gradually assume greater importance in production.

In the field of individual developments the following, among others, may be pointed out.

The last year has seen earnest steps taken and numerous plants built to beneficiate our coals. Both wet and dry processes are in operation commercially, but it perhaps is too early to draw definite conclusions as to the best one to use. In fact, there is such a variation in individual coals and the purposes for which they are used that no one process may fit all cases.

When coal has been improved and is of a quality suitable for metallurgical coke, it certainly tends to greater uniformity in the coke.

The coke used for metallurgical purposes has also been greatly improved because of more careful selection of the coals used in making it, and because of proper sizing and screening and the development of a market for the remainder of the product.

Blast Furnaces—Open-Hearth Furnaces

Recent experience seems to have clearly indicated the merits of the wide-hearth blast furnace. One such furnace with a hearth diameter of 24 ft. 6 in. and a bosh of 26 ft. 3 in. is said to have produced in 1928 a daily average output of 1008 tons.

A paper entitled "Recent Developments in Blast-Furnace

Construction and Practice" which was presented at the Society's Cleveland meeting by Arthur G. McKee, covers quite thoroughly the more recent advances in iron making. In it the author discusses construction, size, auxiliaries, hot-blast stoves, gas-cleaning equipment, and the use of blast-furnace gas in other departments of a steel plant. He also pays a greatly deserved compliment to the group of enthusiastic engineers in the Bureau of Mines at its Minneapolis headquarters. The investigations already made there have been of great value, and the watchful cooperation of all interested in the manufacture of iron and steel should be unstintedly given.

Blast-furnace gas is now being used satisfactorily at coke ovens, thereby releasing the coke-oven gas for other purposes. Mixtures of these gases have been found to give excellent results in other heating operations, and with reduced costs.

The more economical utilization of blast-furnace gas in conjunction with powdered coal in the generation of high-pressure steam is important.

Recent developments in blast-furnace design have called for corresponding changes in turbo-blowers, charging and filling equipment, control of charging, heat practice, and more effective stove equipment.

Proper sizing and preparation of coke and other raw materials and sintering of ores and flue dust have done much to make a more uniform and increased product, and increasing knowledge of the advantages of sintering has resulted in an extensive program of sintering-plant building, these plants apparently being largely of the Dwight-Lloyd type.

No definite decision has been reached as to the maximum size of open-hearth furnaces. According to King and others, the large furnaces are the most economical, and their control is satisfactory.

Doubtless what might be most economical for low- and medium-carbon steels where large tonnages could regularly be disposed of, might not prove to be so for other steels calling for great variation in analysis and which are of limited application in use.

Size might well be a plant problem.

The extended use of new designs of ingot molds and hot tops is of interest.

The Metallurgical Section of the Pittsburgh Experiment Station of the U. S. Bureau of Mines in cooperation with Carnegie Institute of Technology and the Metallurgical Advisory Board has conducted experiments, the results of which have recently been released, showing that the amount of iron oxide dissolved in liquid steel is very much higher than formerly believed to be the case. This fact, it is stated, "changes the concept of methods of working open-hearth heats to a considerable extent, in that it emphasizes the importance of chemical composition and physical properties of the slag and the necessity for an even more careful study of deoxidizers than has been made up to the present time."

Experiments on electrolytic methods for the extraction of nonmetallic inclusions from steel are also under way.

A paper was read before the Iron and Steel Institute by E. J. Janetsky entitled "A Study of Basic Open-Hearth Slags by Solidification Tests." Therefore there are good reasons to hope for real enlightenment on open-hearth practice.

The Bessemer Process

Under the leadership of Dr. G. B. Waterhouse the industry can confidently look forward to practical suggestions for improvements in practice in the bessemer process. Already changes in operation and certain refinements have brought about the replacement of open-hearth with bessemer steel in certain lines.

Rolling Mills

Work has been going on looking to the improvement of the newer methods of rolling wide strips by the continuous process and getting the product of proper quality and under control.

Advances have been made to a marked degree on older mills of the ordinary pattern in output and cost, largely due to continuous heating and rolling changes.

Electrification of old mills and installation of new ones have continued throughout the year. One of the newest is a Morgan mill built for the Sharon Steel Hoop Company, which rolls strip from $2^1/_2$ in. to 10 in. wide at speeds up to 2400 ft. per min. in coils of from 150 lb. to as high as 2000 lb. All strips are produced from 30-ft. billets. The mill is so arranged that the pulpit operator can receive instructions as to operations and speed changes from a loud speaker mounted on the pulpit, the roller talking into a microphone located on the operating floor.

Pipe

Many improvements have been made in the methods of production and subsequent treatment of seamless tubing, and expanding up to large sizes has been introduced commercially. New methods of welding have come to the front and a huge tonnage of the larger sizes has been manufactured; smaller sizes and light gages are also being produced on a large scale.

Wrought Iron

Wrought iron is being commercially produced at Warren, Ohio, by the Aston process, in which a blown metal is pounded into molten slag of the proper analysis held at the right temperature, the resulting sponge or ball of iron being then squeezed, after the slag has been poured off, and the bloom rolled without reheating into muck bar or billets.

Alloy Steels

Great progress has been made in the development of steels for special purposes, such as those which will resist corrosion and are capable of withstanding high temperatures, and also in the ability to produce a large output at decreased costs.

Furnaces

Marked improvements have been made in heating, etc. Kathner and other normalizing furnaces have been installed and are producing a satisfactory product of sheets. The Kathner furnaces use insulated shafts throughout as the conveying mechanism to avoid loss of heat. They are 155 ft. long, and have a heat zone of 75 ft, and a cooling zone of 80 ft. The heating zone is divided into a preheating section of about 35 ft. and a soaking section of 40 ft. The preheating section has no burners within 15 ft. of the entering door, and receives its heat by the backflow of gases from the rest of the furnace. Automatic control equipment has been a factor of importance in this development.

The size of arc furnaces has been steadily growing, the largest, it is believed, being operated by the Timken Steel and Tube Co. at Canton, Ohio. These furnaces in their rating closely approach to the smaller size of the regular open hearth. Of late the electric furnace has found an important application in the field of recovery of chromium- and nickel-steel scrap. When handled in an ordinary open hearth this scrap loses the valuable alloying elements to the slag. In an electric furnace it can be melted practically without any such loss. The furnace installed for this purpose is said to be an induction furnace with the coils located above the hearth. It is of 6 tons rated capacity, and while its initial cost is rather high owing to the very low frequency that has to be used, the economic results of operation are said to have proved to be highly attractive.

The high-frequency furnace has been making further progress, unusually large units having been installed in several places, among others, at the Heppenstall Forge and Steel Company.

An interesting feature of the development of this furnace is the continuous reduction in the frequency employed, which makes for cheaper generators and controlling machinery.

Nitriding

This process was developed in Germany some five years ago. Only during the last year has an apparently successful effort been made to introduce it on a commercial scale in the United States. The largest nitriding furnace in the world is being installed in a plant of a midwestern steel company.

Machines and Tool Materials

Wonderful strides have been made in the last two years in machine-tool design. In fact, so many improvements have been made as to render practically obsolete, for purposes now required in up-to-date shop practice, tools but a few years old.

The newer cutting materials have demanded some of these changes. A whole series of materials of great hardness and resistance to wear has been brought out. Among these are Carboloy, Firthite, or Widia, a tungsten carbide embedded in a matrix, the essential constituent of which is cobalt. Others include such materials as Stoodite (a hard chromium-iron alloy with some manganese), Borium (tungsten carbide base), and Blackor (also a tungsten carbide alloy). All of these are either welded on or mechanically attached to a steel extension piece, while some, including Stellite, may also be applied as a facing on a steel base.

Sponge Iron

Blast furnaces have not as yet been eliminated. Rumors are to the effect that headway is being made with the so-called Hornsey process. The Smith process may prove to have a certain field; some of the results look hopeful.

Centrifugal Casting

Maj.-Gen. C. C. Williams has announced that experimental work on the centrifugal casting of guns leads to the belief that the process has been satisfactorily established for the 37-mm. gun, the 75-mm. infantry mortar, and the 75-mm. howitzer. Also that a larger machine is being constructed for production of the 75-mm. field gun and the 105-mm. howitzer. It is asserted that the physical properties obtained from centrifugal castings, after treatment, are superior to those obtained from forgings of the same composition.

Billet Chipping

An earnest effort is being made to do away with the chipping of billets by compressed-air-operated chisels. In the past year several machines have been placed on the market, one of them a miller, another a planer, while a third, for round billets, operates somewhat like a pencil sharpener. A certain amount of research has been done by several steel companies to determine the extent of surface defects which have to be removed by chipping and whether there is any way of reducing their number or eliminating them completely.

Scrap

A research bureau has been launched by the newly formed Institute of Scrap Iron and Steel, Inc., to make a thorough survey of every phase of direct dealing between consumers and producers of scrap. Owing to the wide distribution and use of alloy steels, the proper classification and economical use of scrap steel is becoming of increasing importance.

Foreign Patents

A significant development in the past year is represented by the tendency of American manufacturers to take out licenses under patents originating in Germany. Two of the most important of such developments have been the introduction of cutting metals of the tungsten carbide type, referred to elsewhere, and the introduction of the nitriding process, both inventions of members of the staff of the Krupp Company. Another foreign invention taken up by an American manufacturer is pearlitic cast iron. The Midvale Company, on the other hand, has taken out licenses under French patents owned by the Commentry-Fourchambault et Decazeville Co. for the manufacture of a heat-resisting alloy of the chrome-nickel-iron type.

C. Snelling Robinson, *Chairman.*

Use of Alloy Steels in Iron- and Steel-Mill Equipment

By E. R. JOHNSON[1] and O. BAMBERGER,[2] CANTON, OHIO

Substituting alloy steels for the ordinary carbon steels in its mill equipment is the practice of the Central Alloy Steel Corporation when occasion arises to make a change. Elimination of breakdowns and costly repairs is claimed to be furthered by certain uses of the alloys. A gradual building up of the alloy-steel equipment is taking place. Producing alloy steel itself, the initial cost is low. There are three requirements which permit the substitution of alloy steels: (1) Increase of the physical properties of strength, ductility, and toughness. (2) The ability to withstand wear and abrasion. (3) Resistance to corrosion.

THE steel mill, as well as any other industry, is always aiming to develop a maximum production at a minimum expense. To attain this condition steel-mill equipment must necessarily function with few breakdowns and costly repairs. It is therefore of prime importance that this equipment be designed to operate continuously, for a number of departments in a steel mill operate day and night for the seven days of the week.

To meet this demand it has been found necessary to substitute other materials for the ordinary carbon steels. Our company has been producing alloy steels for many years, and it is therefore only natural that parts are replaced with the various alloy steels. Because of this the initial steel cost of alloy is favorable, and occasionally we can use heats off in analysis for certain classes of work.

In general it can be stated that there are three requirements which permit the substitution of alloy steels for the plain carbon steels:

1 Increasing physical properties with respect to strength, ductility, and toughness. The physical properties that can be obtained depend upon carbon content, alloy content, and resultant heat-treating operations. A comparison of any of the standard physical-property charts brings out the superiority, especially in the larger sections, of alloy steels over carbon steels at similar hardness values. Therefore when weight becomes a factor, it becomes possible to decrease sectional areas by using alloy steels and still have the proper physical qualities.

2 The ability to withstand wear and abrasion. Usually this quality is dependent upon hardness values. High hardness values give the best wearing quality. There are a number of applications where abrasion resistance and toughness are desired. This always is best obtained with alloy steels. Likewise high hardness values with depth of penetration to support surface loading are more easily obtainable by using alloys.

3 Resistance to corrosion, both wet and dry. The latter may be considered scaling at elevated temperatures. Both of these require special alloy materials, with high percentage of alloying content.

To meet these special functions, various alloy steels have been developed by adding additional elements—nickel, chromium, molybdenum, vanadium, tungsten, and copper, either singly or in combination. Nickel steels are used primarily for increased ductility and resistance to shock; chromium steels for resistance to wear at high hardness, along with depth of hardness penetration; molybdenum is generally used in combination with chromium and nickel, bringing out increased physical qualities; while vanadium is used either alone or in combination with chromium. The latter element has a refining influence in grain size and fracture grain size. Tungsten is of course majorly used for high-speed steels. At present the chief function of copper is increasing the resistance of low-carbon materials to weather corrosion.

Heat Treatment

The best qualities of alloy steels are usually enhanced by suitable heat-treating operations. In fact, for certain parts it is necessary to heat-treat. To take care of our parts equipment, we have developed an entirely separate heat-treating unit which includes all facilities for proper treating and handling. This enables one heat-treating unit to concentrate on mill equipment and thus develop a crew of men who understand the various applications and strive to develop the maximum service for each part.

Steel-Mill Equipment

To facilitate the description of the alloy-steel equipment, a large steel mill may be divided conveniently into a number of operations. To begin with, there are coke ovens, followed by blast furnaces, open hearths, pouring and stripping of ingots, soaking pits, blooming mills, chipping beds, finishing mills, sheet mills, strip mills, and a group of miscellaneous departments which include the heat-treat, cold-draw, machine-shop, and fabricating departments. A number of parts equipment, such as cranes, are general to many departments.

At the coke ovens there was a failure of the pushrod on the pusher within three months after receiving the machine. A pushrod was substituted which had been machined from a carbon-vanadium steel of 0.45 to 0.55 C, 0.40 to 0.70 Mn, and 0.15 V. This rod was normalized at 1600 deg. fahr. and reheated to from 1150 to 1200 deg. fahr. The service has now been continuous for 18 months without any indication of failure.

At present the pusher is equipped with alloy-steel track wheels, gears, pinions, and shafting. The track wheels are forged from a chrome-nickel molybdenum steel having an analysis of approximately 0.45 to 0.55 C, 0.50 to 0.80 Mn, 1.50 to 2.00 Ni, 0.70 to 0.90 Cr, and 0.10 to 0.20 Mo. After forging, the track wheels are cooled carefully to prevent the formation of internal ruptures. An annealing treatment is required to make the steel suitable for machining. The heat treatment applied to these wheels is an oil quench of from 1525 to 1550 deg. fahr., followed

[1,2] Central Alloy Steel Corporation. Mr. Johnson was graduated from the University of Michigan in 1921 and obtained a master of science degree in chemical engineering in 1922. He has been connected with the Central Alloy District of the Republic Steel Corporation for the past eight years in various capacities. At present he is acting as assistant chief metallurgist of the Central Alloy District. Mr. Bamberger has been in the employ of the Central Alloy District, Republic Steel Corporation, for the past ten years. He has had extended experience in the heat-treatment of steel for steel-mill equipment and at present has complete charge of this phase of mill work, including the forge shop at the Canton Works.

Presented at the National Iron and Steel Meeting, September 11 to 13, 1929, Cleveland, Ohio.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

by a draw of from 750 to 800 deg. fahr., which gives a hardness of 70 to 75 scleroscope. All the cranes and buggies have been equipped with this type of track wheel. The steel has the property of hardening deeply for resistance to wear and has in addition an inherent toughness to withstand sudden shocks. Previous experience on one particular stretch of track has shown that carbon-steel wheels lasted three months, alloy castings six to eight months, and the foregoing type of track wheel two years, or eight times the life of carbon-steel wheels. In addition there is a saving of numerous installation costs. An illustration of a typical track wheel is shown in Fig. 1.

The gears are made of cast alloy steel having an analysis of 0.35 to 0.45 C, 0.50 to 1.00 Ni, and 0.35 to 0.45 Mo. The gears are annealed, machined, and treated by oil quenching from 1500 to 1525 deg. fahr. and drawn at 900 to 950 deg. fahr., which gives a scleroscope hardness range of 60 to 70 according to size. The use of nickel increases the resistance to impact and at the same time develops sufficient hardness for good wearing qualities. The average life of this type of gear casting is one year to two and one-half years, whereas with plain carbon-steel castings the average life was only three to eight months.

Another part which was changed to alloy steel was the shafting. Generally cold-rolled carbon-steel shafting is supplied with the machines. For shafting we have standardized on a chrome-nickel steel, S.A.E. 3140, which can be oil quenched and drawn to a hardness of 269 to 302. At this hardness the shafts can be machined and yet have 20 per cent greater strength values. This increase in physical properties brought out by heat treating enables one to reduce sectional sizes. Sections under 6 in. diameter are heat-treated; over this diameter the shafts are normalized and drawn.

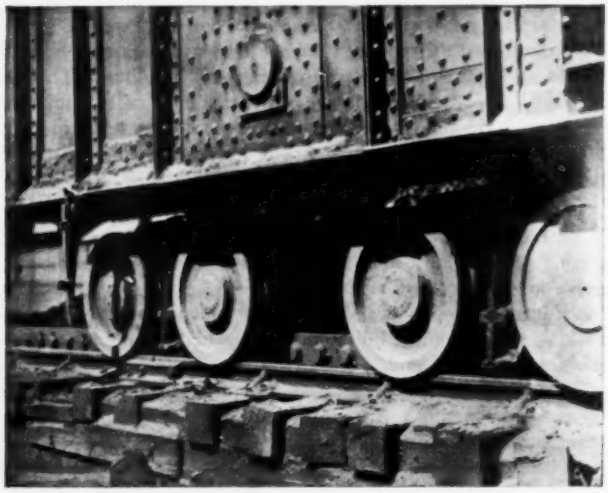


FIG. 1 ALLOY TRACK WHEELS ON ORE-BRIDGE CRANE

Another place for the alloy steel on the pusher, and for that matter on all cranes, is for the pinion gears. For this part fairly high hardness values are required to combat wearing and abrasion. In addition, toughness is essential to withstand sudden load applications. A change therefore has been made to a heat-treated chrome S.A.E. 5150 or chrome-vanadium steel S.A.E. 6150. The hardness value is held within the range of 418 to 477 Brinell.

The foregoing types of steel have been standardized and are used for the particular parts on all cranes in our plant. Armature shafting is also heat-treated S.A.E. 3140 steel.

Along with the substitution of alloy steels for the carbon steel, it is quite essential that design of parts be made to give maximum service life. The various gears and shafting are usually keyed, and it is advantageous to have radius instead of any sharp corners, for it has been found that sharp corners concentrate stress and lead to early failure by fatigue or fracture. Likewise teeth on all gears should be cut to allow a radius at the root of teeth. Fig. 2 presents a view of various alloy-steel gears and pinions.

At the blast furnace stainless-steel tubes have been installed in the power house for soot blowing in the boilers. The type of tubing used previously developed a grain growth and ultimately cracked where the bushings are inserted, thus permitting them to fall out. The present installation of stainless tubes has worked satisfactorily for one year and there is no indication of any failure.



FIG. 2 VIEW OF ALLOY GEAR CASTINGS AND FORGED PINION GEARS

For the automatic stokers there are being used chrome-steel (5150) rollers and chrome-vanadium (6145) shafts. Both of these parts are heat treated to 600 and 550 Brinell, respectively. At this high hardness wear is at a minimum, and due to the great depth of hardness obtainable with these steels, the penetration of wear will be very slow.

After the installation of this power house and the blast furnace, it was found necessary to substitute alloy steels, as a number of parts failed within a short time. On one of the booster pumps the carbon-steel shaft fractured, and it was replaced with a heat-treated S.A.E. 3145 steel at a hardness of 286 to 302 Brinell. The bell rod at the top of the blast furnace was also changed to a heat-treated chrome-nickel steel, for a break of this rod would mean a costly delay.

At the open-hearth furnaces the charging crane has been equipped with various alloy-steel shafts, gears, pinions, and track wheels.

In addition the peel of the crane has been changed to a carbon-vanadium steel (0.45 to 0.55 C, 0.50 to 0.80 Mn, 0.15 V). Due to the sectional size the peel is normalized and then drawn to a hardness of approximately 187 to 207 Brinell. A tensile strength of 90,000 to 100,000 lb. per sq. in. will be obtained.

The dampers on the furnace are equipped with chrome-nickel S.A.E. 3140 connecting rods.

The ladle cranes have the same general alloy-steel parts that other cranes have. The gears are heat-treated to slightly lower hardness values, to minimize danger of fracture or breaks from sudden shocks.

At the blooming mill there are being used adamite rolls, which are high-alloy castings. The alloy content permits much longer life from the standpoint of wear. The table rolls adjacent to the mill on both sides are made of carbon-vanadium steel and have re-

ceived a normalizing treatment followed by a high draw. Carbon steel castings had been used previously, and the constant bumping of the ingots on these table rolls after coming through the mill would break the table rolls. Changing to an alloy-steel forging, the brittleness was eliminated, a table roll was obtained with a tensile strength of 90,000 to 100,000 lb. per sq. in., elongation 20 to 24 per cent, and reduction of area of 40 to 48 per cent. The gears driving these rolls are made from nickel-molybdenum castings and heat-treated to from 400 to 444 Brinell. A carbon-vanadium steel is also used for the spindles which drive the blooming-mill rolls. The fracture of a carbon-steel spindle is shown in Fig. 3.

To take up shocks and excessive loads on the blooming mill, a Falk coupling has been installed. A picture of this is shown in Fig. 4. The springs are made of chrome-vanadium steel, S.A.E.



FIG. 3 VIEW OF BLOOMING-MILL SPINDLE WHICH FRACTURED

6150, heat-treated to from 418 to 444 Brinell hardness. This coupling has enabled increased production and has eliminated the breakage of bolts that constantly occurred in the old type of coupling.

At the blooming mill shears, as well as on all finishing mill shears, we are using either chrome S.A.E. 5155 or chrome-vanadium S.A.E. 6155 steel for shear blades. For the blooming-mill shears where hot steel is sheared, the blades are oil quenched and drawn to a hardness of from 444 to 460 Brinell. On the finishing mills where the steel is black, the blades have higher hardness values, 495 to 512 Brinell. A steel of this type will tend to retain its hardness and is not easily affected by the heat of the bars that are being sheared. At the strip mill it has been found advisable to keep water flow at a minimum to decrease the amount of fire-cracking on the shear-blade edges while shearing the hot steel. The life of the blade is thus materially lengthened.

The finishing mills may be put in the same class as the blooming mill, for the rolls are usually high-alloy-steel castings. The line-shafting gears and pinions on the hot beds are made from the general steels discussed previously.

There are a number of applications in the sheet mills for alloy steels. These include bar-shear knives of high-carbon-vanadium steel which can be treated to high hardness values and still retain the cutting edge without chipping. On the mills proper alloy steel is used for the hot-mill screws and hot-mill boxes. The roller leveler requires very hard rolls, and a high-carbon–low-chromium steel which will heat-treat to high hardness is usually used. A picture of a roller leveler is shown in Fig. 5.

There are in operation several normalizing furnaces for sheets which are equipped with special heat-resisting alloy rolls. These furnaces operate at temperatures of 1800 deg. fahr. and must therefore have rolls which resist scaling, even though the rolls are generally cooled by water.

In addition to the departments discussed, there are a number of miscellaneous applications in the cold-draw, heat-treat, machine-shop, and fabrication departments. In the cold-draw department a high-carbon–low-chrome steel of high hardness is used for all jaws on the benches and for the fliers. The rolls of the fliers and



FIG. 4 BLOOMING MILL COUPLING (FALK) USING CHROME-VANADIUM SPRINGS

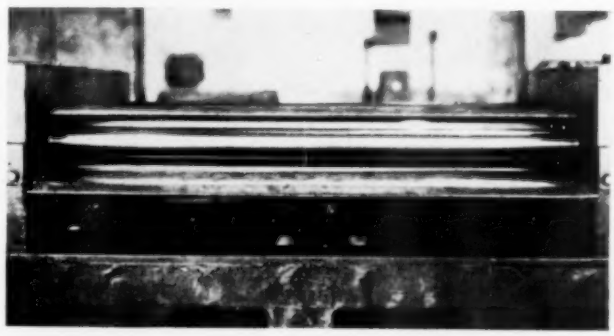


FIG. 5 VIEW OF ROLLER LEVELER ROLLS WITH ALLOY-STEEL ROLLS

straightening machines are made of high-carbon–high-chrome steel, treated to high hardness values. This steel has good wearing qualities and also hardens very deeply, thus permitting high stresses.

Stainless steels find application in the heat-treating department because of their resistance to oxidation at elevated temperatures.

CONCLUSION

We try to analyze the failures and to base the substitutions on metallurgical facts. In certain cases it is not possible to make

substitutions on straight engineering data. It is rather difficult at times to calculate with any degree of accuracy the distribution of loading or stress, and it is necessary to continue substitutions until a satisfactory material is obtained. One instance of this is the substitution of a stainless-steel shaft in an impeller pump. This pump operates constantly, but there is insufficient water supply to maintain a full head, and it has not been possible to use either carbon or heat-treated alloy-steel shafts. The stainless-steel shaft has been in operation for eight months, while previous shafts of other materials averaged one month. A picture of the impeller blade and shaft is shown in Fig. 6.

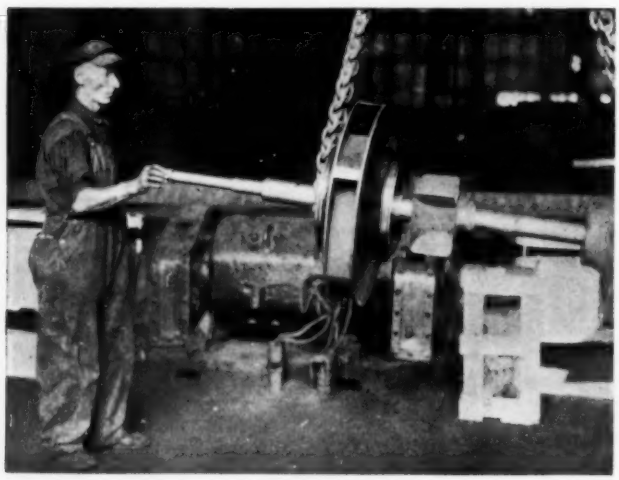


FIG. 6 IMPELLER PUMP BLADE AND SHAFT, THE SHAFT OF STAINLESS STEEL



FIG. 7 FATIGUE FAILURE OF SHAFTING, RESULTING FROM CONCENTRATION OF STRESSES

Failures can be attributed quite often to causes which can be avoided easily, such as misalignment, surface-tool marks, and insufficient radius with change of section. These conditions will cause concentration of stress and lead to a progressive failure. A typical progressive failure due to one of these causes is shown in Fig. 7, while a failure due to excessive torsional stresses is shown in Fig. 8.

A gradual building up of the alloy-steel equipment is taking place, and at present there is being used 100 tons of the various steels each month. It is felt that these changes eliminate costly breakdowns and in addition add to efficiency by giving greater production at less cost.

Discussion

W. M. FRAME.[3] The paper deals with the use of alloy steels in steel-mill equipment from a maintenance standpoint and is very interesting. It would also be interesting to hear a paper regarding the use of alloy steels in new equipment from the builder's standpoint.

A large percentage of the steel-mill equipment now in use



FIG. 8 TORSIONAL FAILURE OF SHAFT DUE TO EXCESSIVE STRESSES

originally contained very little alloy steel. This is in one way perhaps a fortunate condition. Due to modern demands much of the equipment installed a number of years ago is being operated today under conditions of load and tonnage never considered by the designers. Under such severe operation the weak points in the design are quickly indicated. As it is usually very difficult to change the design of the weak part, the solution seems to be the substitution of heat-treated alloy steels for the untreated plain-carbon steels. By the use of the proper alloy steel, the weak part can usually be strengthened enough to stand up indefinitely.

In some mills there is a deplorable tendency to replace the same weak part, time after time, with another part exactly like the one that failed. Surely this is not the most advisable thing to do. If the failure is due to the steel used in the part, the slight additional cost of alloy steel will be negligible compared with the savings effected by continuous operation. Also, the sudden failure of a part may cause the failure of other more expensive parts. Suppose a hot strip mill goes down for two hours to change a broken drive gear on a roll table. The profits lost on the material that would have been rolled while the mill was down would

[3] Otis Steel Company, Cleveland, Ohio.

probably buy several alloy gears, not to mention the operating expenses which go on whether the mill is working or not. Suppose a blooming-mill spindle pin breaks and allows the spindle to kick out and wreck itself. The cost of a new spindle would perhaps buy 50 alloy pins. This example happened recently. The plain-carbon pin was replaced with an alloy pin made of $3^1/_2$ per cent nickel steel, S.A.E. 2330, normalized, machined, and drawn.

The authors' suggestion that a separate heat-treating unit be provided to take care of maintenance work is a very good one. Often the tendency to replace failures with untreated carbon steels is due not to ignorance but to lack of material and equipment and time.

The authors mention the use of gears which have been heat-treated after machining. This is allright for cranes and many other applications, but there is a class of gears for which this method of hardening would not be desirable. For instance, in cold strip mills the gears have to be accurately cut and be free from distortion to prevent chatter marks from appearing on the finished strip. Also, wear which ordinarily would not be considered excessive must be absent from the gears on the finishing stands at least. Untreated carbon-steel gears are too soft and do not wear very long. These gears are being replaced with cast chrome-molybdenum steel and the pinions with forged chrome-molybdenum steel. The treatment is as follows: anneal, rough-machine, normalize, draw, and finish-machine. In this manner gears can be obtained much harder and more wear resisting than untreated carbon-steel gears and still be free from distortion.

The use of alloy steels is not a cure-all, and it is not economical to use them where the cheaper carbon steel will do the work. As the authors mention, failure sometimes occurs due to poor design, misalignment, sharp fillets, etc. These causes can usually be detected and sometimes corrected. However, it is believed that our maintenance costs and our undetermined losses can be materially reduced by the proper supervision and application of material to replacement parts.

The tendency of finishing-mill design seems to be toward continuous mills and multiple units in series operation. In series operation a breakdown of one unit in the series causes a shutdown of the other units. The series represents a large investment and must be kept working in order to show a suitable return. It is believed that the proper use of alloy steels will help to insure the owners against breakdown of this expensive equipment.

Charles McKnight.[4] The iron and steel equipment field has not progressed very rapidly in the use of alloy steels because the incentive which the automotive industry had has been lacking. In the automotive industry it is essential to have strength coupled with lightness. In the steel mills there have been few limitations as to weight or size, and it has been the natural tendency to obtain the requisite strength by increasing sizes. There are, however, as the authors point out, definite advantages to be gained by the use of alloy steels.

Of course, in one department of the steel industry alloy steels have been used for a long time. In this paper reference is made to the use of "Adamite" rolls. This is a metal which might be called either a steel or an iron which owes many of its peculiar properties to the fact that it contains nickel and chromium. There are other mixtures which likewise employ nickel and/or chromium, sometimes in combination with molybdenum, in the manufacture of rolls, and their use has long since demonstrated their value.

One of the newer developments in the manufacture of rolls has been the use of a high-nickel-chrome cast-iron mixture for cold rolls. In cold rolls where it is necessary to obtain a very high finish, recourse has usually been made to the hardened and ground steel rolls. These were forged from high-carbon chrome steel, hardened by quenching and ground to a very high polish. It has been found possible to obtain similar results at a lower cost in many of the sheet-mill applications by using a chilled cast-iron roll with about 5.0 per cent nickel and 3.0 per cent of chromium. In many instances the hardened and ground roll is still superior and should be used, but there also are many other places where a cheaper chilled roll is doing good work.

Mention is made in the paper in two or three cases of a normalized carbon-vanadium steel for such parts as large shafts, push-rods, and charging-machine peels. There is another steel which lends itself peculiarly to these uses. This steel has been developed in the last few years, principally for work in the locomotive field, and contains from 2.0 to 3.5 per cent nickel with low carbon (0.17 to 0.27 per cent). The advantages in its use are that, used in the normalized condition, it will develop very satisfactory strengths, but at the same time has a remarkable toughness and resistance to impact. This toughness is indicated by the fact that in a large forging a tensile strength of 80,000 to 85,000 lb. per sq. in. can be obtained with an elongation in excess of 30 per cent and a reduction of area of over 60 per cent, while the Izod impact value will run between 70 and 80 ft-lb. Steel of this sort is of value where reliability is a paramount consideration; as, for example, for ladle hooks where the breaking will probably lead to an accident having tragic consequences. It has also proved very satisfactory in large shafts where stronger and harder steels break in a much shorter time than the tough low-carbon-nickel steels.

The authors have shown the way to a more general use of alloy steels in the steel mill. It should lead to a much wider utilization of these materials by all the steel mills as there can be no question of the economic possibilities.

[4] In charge of Alloy-Steel Development, The International Nickel Company, Inc.

Recent Developments in Blast-Furnace Design and Construction

By ARTHUR G. McKEE[1] and WM. A. HAVEN,[2] CLEVELAND, OHIO

This paper deals with two recent and highly important developments in blast-furnace construction and practice; namely, the trend toward furnaces having daily capacities of 1000 tons or more, and the profitable utilization of blast-furnace gas in the firing of coke ovens, making open-hearth steel, and in the various types of heating furnaces employed in steel mills. These developments have involved certain structural changes in the stacks, which have in time necessitated changes in such auxiliaries as stock-house and blowing equipment, gas cleaners, hot-blast stoves, and boilers. These are all considered in detail by the authors.

THERE have been two recent developments of major importance in blast-furnace construction and practice; namely, the movement to larger furnaces, having daily capacities of 1000 tons or more, and the very profitable use of blast-furnace gas beyond the confines of the blast-furnace plant, for underfiring coke ovens, for making open-hearth steel, or for use in various types of heating furnaces throughout the steel mills.

These two developments have involved a number of structural changes in the stacks and have made it necessary to give much more consideration than formerly to certain auxiliaries such as stock-house and blowing equipment, gas cleaners, hot-blast stoves, and boilers.

The Furnace Stack

The ability to make 1000 tons from a single stack was accomplished rather suddenly, if not dramatically, by a radical increase in the diameter of the furnace hearth. Whereas, previously, increases in hearth diameter had been a few inches per year, in the summer of 1927 a furnace was built having a diameter of 24 ft. 6 in., which was about 3 ft. greater than that of any previous design. (Fig. 1.)

While this furnace did not immediately produce 1000 tons per day, it did produce more than 900 tons without scrapping, notwithstanding the fact that the top of the furnace had been left disproportionately small, and in spite of inadequate blast-heating facilities and an uneven grade of coke.

A matter of tremendous importance to the industry lay in the fact that this advance was accomplished without operating difficulties, and predictions were promptly made by students of the art that with higher blast temperatures and better fuel the furnace in question would easily reach the 1000-ton goal.

A few months later, under improved conditions in these respects, this furnace was consistently making 1000 tons daily, and for the year 1928 had the surprising total output of 368,000 tons or a daily average production of 1008 tons. This truly remarkable achievement by the personnel of the Jones & Laughlin Steel Company was accomplished without adequate adjustment of the other parts of the equipment to the greatly increased demands made by this large output.

It remained, then, to match this development of hearth size with a corresponding increase in other furnace dimensions as well as in capacities of the furnace auxiliaries.

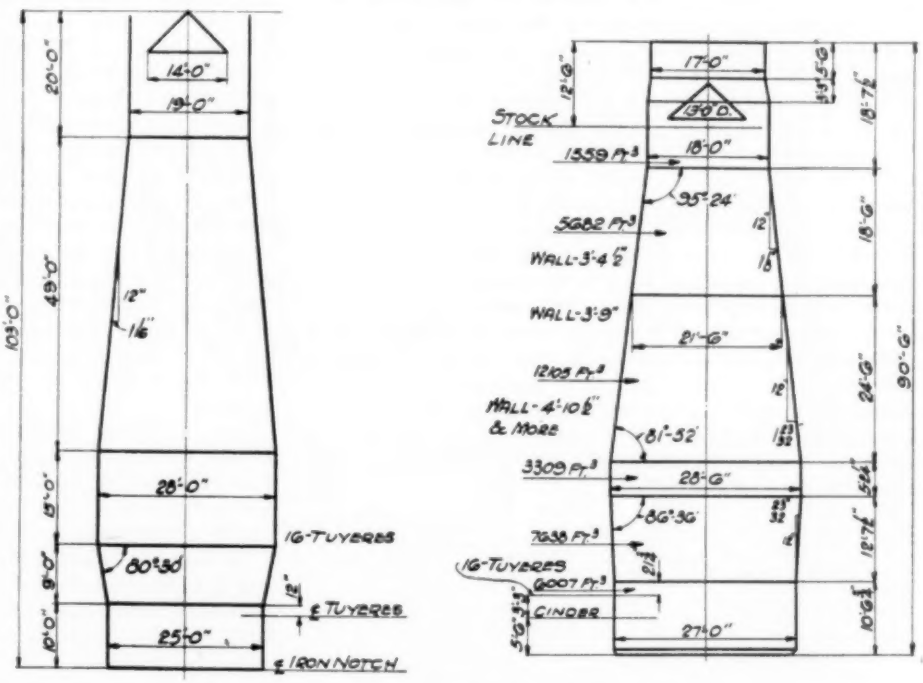


Fig. 1 Lines of Two Modern 1000-Ton Blast Furnaces

Even in furnaces of the 21-ft. 6-in. hearth diameter, there had been considerable hesitancy about enlarging the furnace tops proportionately. In many cases this was a natural consequence of the fact that the larger hearths were fitted to small stacks to avoid the greater cost of a complete rebuilding, but even where new furnace shells were installed the old conservative stock-line diameter of 17 ft. or 17 ft. 6 in. was reproduced.

It had been long the belief of the authors, however, that top development had lagged behind hearth development and had

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

been one, if not the most important, cause of the seriously increased flue-dust losses that have attended the large units and of the much greater volume of blast which they have required.

It is an outstanding development of recent months that three furnaces have been blown in having a top diameter of 19 ft., and hearth diameters of 19 ft., 21 ft. 6 in., and 25 ft., respectively.

In each case the practice of these furnaces has been greatly improved, particularly in the matter of flue-dust production. Losses in each furnace have been in the neighborhood of 150 lb. of dust per ton of pig iron, whereas from 200 to 400 lb. per ton is common practice with smaller tops.

On the last two furnaces mentioned another significant change has been to increase the distance between bell and stock line from the long-customary figure of 2 ft. to that of 2 ft. 6 in. The relationship between large bell and stock-line diameters has always been known to be a matter of vital importance in securing the proper distribution of stock within the furnace, and this apparently small change may well have a very great bearing upon the improved results obtained with the larger tops.

The increasing size of bells has been in itself one of the problems in connection with these large stacks. The casting and machining of 14-ft. or 15-ft. bells taxes the capacities of the largest steel foundries, but the transporting of such a casting to the furnace plant presents almost insurmountable difficulties, as it has dimensions in excess of maxium railroad clearances.

As a practical solution of this difficulty, the authors suggested some years ago the use of bells made of a central portion, to which is riveted and welded a skirt or rim of two or more pieces after delivery to the site. In two cases just mentioned the bells are 14 ft. in diameter, and when it is understood that a $13^1/_2$-ft. 50-deg. bell is the maximum limit of standard railroad clearance, we can appreciate the difficulty of transporting a $14^1/_2$-ft. or 15-ft. bell. The authors understand that one of these 14-ft. bells was delivered by barge from shop to furnace plant.

At the present time the largest furnace in existence has a 27-ft. 6-in. hearth, but the stock line is only 18 ft. Other furnace dimensions have, of course, come in for renewed study and consideration since the advent of the 25-ft. hearth. Without going into detail in each one of these cases, the authors would suggest the following figures as being suitable for a modern furnace of 1000 tons' daily capacity, and believe that these dimensions, or larger, will be recognized as standard in the near future; further it is probable that the stock-line diameter may be increased at a faster rate than the hearth diameter, until they are more nearly equal than is now the practice.

Hearth diameter	25 ft.
Bosh diameter	27 ft.
Stock-line diameter	20 ft.
Bell diameter	15 ft.
Total height	95 ft.
Bosh height	10 ft.
Straight section above mantle	10 ft.
Angle of bosh	84 deg. 17 min.
Slope of inwall	$^{13}/_{16}$ in. per ft.

It may be safely predicted that a furnace of these dimensions, if provided with lake ores of average iron content and with a good grade of coke, will produce 1000 tons of pig iron per day, and that the flue-dust losses should be very low if the downcomers are properly designed. Little or no scrap should be necessary to accomplish this tonnage, but there should be adequate provisions for heating the blast to 1500 deg. fahr., although the operator may elect to use temperatures somewhat less than this figure.

For a furnace of the size under consideration the matter of foundations is one of the utmost importance. The total load on the foundations will be approximately 5000 tons, and the foundations themselves will weigh an equal amount. Very careful study of the soil conditions is consequently called for when the matter comes up of building new or rebuilding an old furnace to these dimensions.

The number of columns and tuyeres employed is optional, generally 8 to 12 columns and 12 to 16 tuyeres.

Furnace Auxiliaries

As previously stated, the advent of the 1000-ton furnace has necessitated a corresponding increase of capacity in the furnace auxiliaries.

It should be noted at this point that none of the larger furnaces of today is a new furnace plant; all are units of old plants wherein in all cases the stock houses, stoves, and blowing equipment were originally designed and built for furnaces of 500 tons capacity or less. It is highly improbable that many additions will be made to the number of furnaces at existing plants, for the obvious reason that capacities are more readily and economically increased by enlarging the existing stacks. As is well known by every one, this has led to a rapid decrease in the total number of active stacks notwithstanding a largely increased potential total capacity and actual output.

It is of course more difficult, though much less expensive, to double the capacity of existing auxiliary equipment than it would be to build entirely new. At each plant, moreover, the problem will be different. It is therefore not possible to describe in detail the many expedients which have been practiced to keep the auxiliary equipment in step with increased stack capacities.

In general, it may be stated that the tendency in stock-house construction is definitely toward central coke bins, preferably supplemented by weigh hoppers for coke weighing, double-compartment larry cars, and double skips of large capacity. Skip hoists and bell dumping rigs are of course of increased size and electrically operated.

For blowing equipment, turbo-blowers have proved particularly well suited for the larger furnaces due to their highly efficient operation and also to the small amount of floor space which they require, so that they can readily be installed in place of the much bulkier old blowing engines which they replace. By the substitution of turbo-blowers for reciprocating engines, it is quite possible to more than double the amount of blowing capacity which can be housed on a given area of foundation or in a given building.

In a similar way modern large-capacity boilers have made it feasible to meet the increased demands of the furnace plant for steam by replacing the many small units of former days with a few very large ones, requiring much less space per unit of capacity except toward the stars, where it is readily available.

While on the subject of boilers, it may be noted that powdered coal is finding more and more favor as an auxiliary fuel for use with blast-furnace gas. The chief reason for this may be found in the fact that the switch from one of these fuels to the other is more readily effected than with stokers. At one large installation of this type the auxiliary firing is automatically controlled, and is stopped and started as the supply of gas is equal to or fails in meeting the demands of the plant for steam.

The most important developments and changes in furnace auxiliary equipment, however, have been in connection with blast-furnace stoves and gas-cleaning equipment. These subjects the authors believe to be worthy of consideration in some detail.

Hot-Blast Stoves

The matter of stove construction has been brought into prominence recently, both by the requirements of the larger furnaces and by a greater appreciation of the value of blast-furnace gas for other purposes. The one has made it imperative to secure greater blast heating capacity and the other has emphasized the

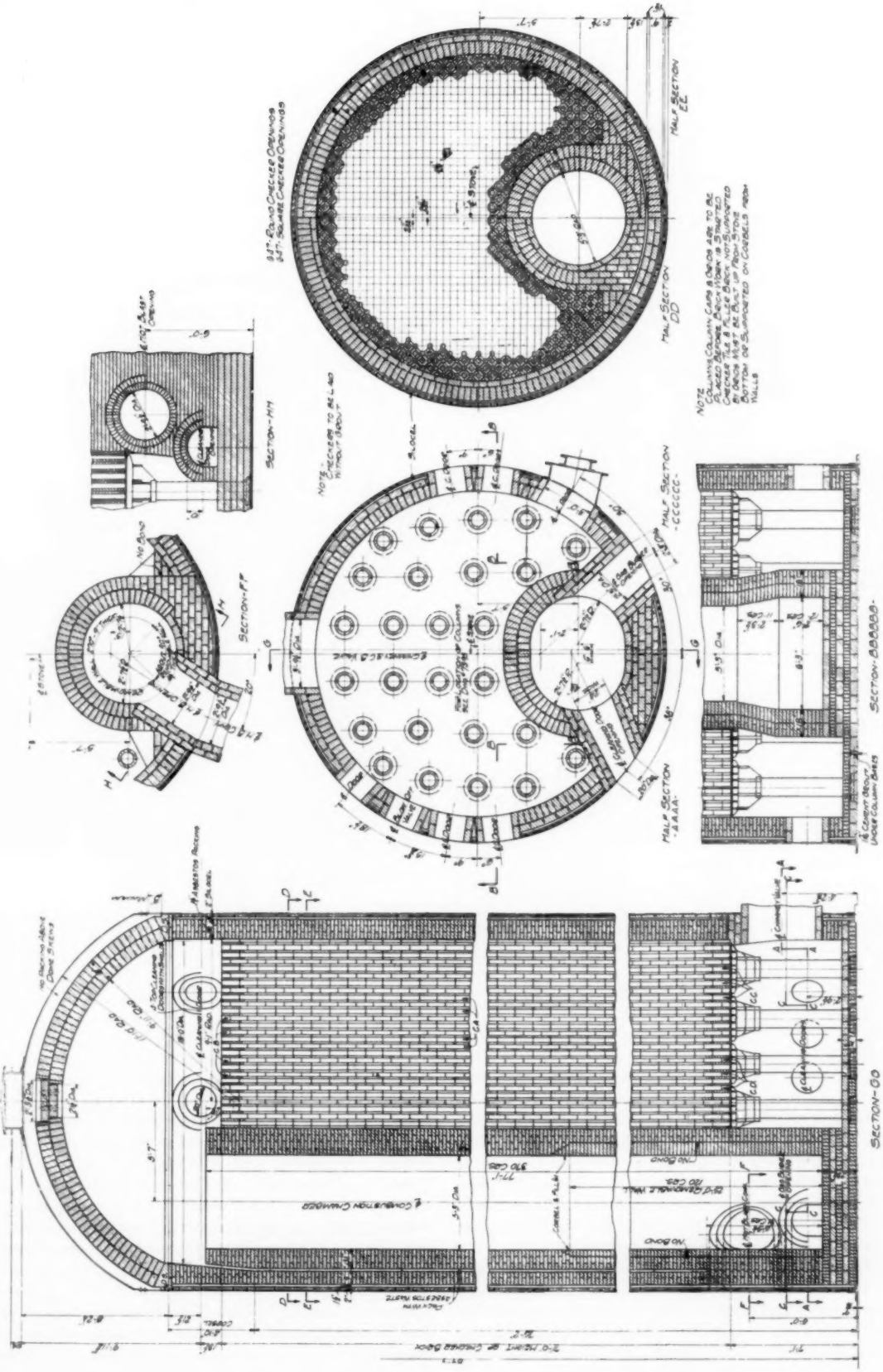


FIG. 2 SECTIONS OF TWO-PASS METAL-BOTTOM STOVE

importance of gas conservation through higher stove efficiency. Another reason for better stoves lies in the generally improved furnace practice, which has made the use of higher blast temperatures feasible. From 1200 to 1500 deg. is now common practice, which is probably two or three hundred degrees higher than the average of a few years ago.

A great majority of the older furnaces were equipped with four stoves each, which varied greatly in size and type of checkerwork. A common or average size of the old two-pass stoves would probably be about 21 ft. × 100 ft. Many of the stoves still in use

With the advent of gas scrubbers the possibility of using smaller checker openings was quickly realized, with the result that many of the old stove shells have been relined during the past decade with openings of from $3\frac{1}{2}$ to 5 in. and with brick as thin as 2 in. With construction of this type it has been possible to practically double the heating capacities at certain plants merely by relining the stoves with smaller checkers. During the same periods the use of better insulating materials such as diatomaceous earth has also accomplished a reduction of radiation losses. The use of pressure burners has further increased the capacity of existing

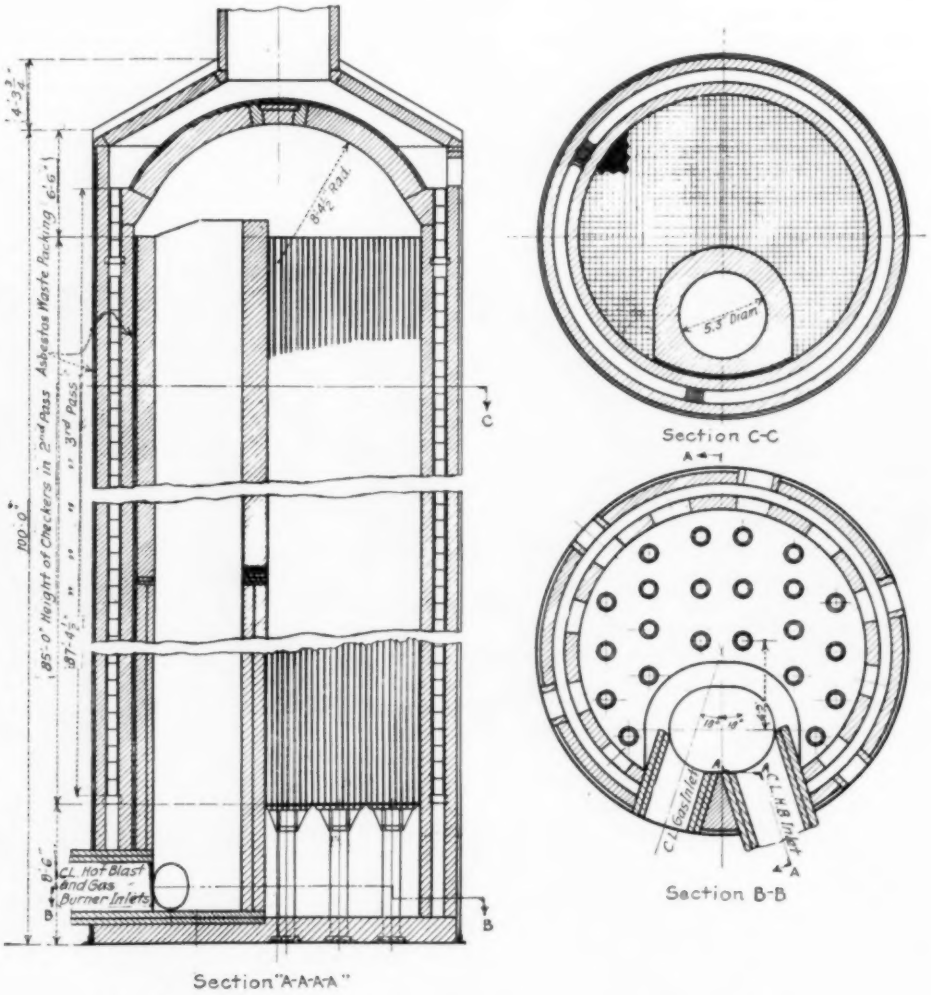


FIG. 3 SECTIONS OF THREE-PASS STOVE HAVING METAL-SUPPORTED CHECKERS

are of the McClure three-pass type, which were generally 21 ft. or 22 ft. × 8 ft. up to 100 ft. Many of these stoves were built more than twenty years ago, previous to the general adoption of wet cleaning of blast-furnace gas, and are still lined with the very large checker openings which the use of dirty gas made necessary. From 35,000 to 50,000 sq. ft. of heating surface per stove was generally the limit for this old equipment.

Under such conditions very high stack temperatures were unavoidable. In many cases the temperature of the stack gases would be equal to or greater than that of the blast leaving the stoves. Insulation was generally blast-furnace slag, and between high radiation losses and stack losses the thermal efficiency would be very low, seldom higher than about 60 per cent.

stoves, although often at the expense of stove efficiency through increased stack temperature, unless the heating surface has been largely increased so as to absorb the heat from the increased volume of hotter gas. The improvements just outlined, viz., smaller checkers, better insulation, and pressure burners, may be defined roughly as taking place in a period extending from 1914 to 1928 and contemporary with a general increase in furnace capacity from 500 to 750 tons per day.

The modern blast furnace, especially when operated in conjunction with steel plants and rolling mills, will, in all probability, soon be supplying blast-furnace gas for many different metallurgical purposes, and must be provided with stove equipment which is ample for blast heating requirements and at the same time

highly efficient in use of gas, which is in that case a very valuable product.

Radiation losses must be kept to a minimum and stack temperatures low. Checker openings must be as small and checker walls as thin as is consistent with the cleanliness of the gas to be used and the structural strength required.

About 400 to 450 sq. ft. of heating surface for each ton of pig iron produced per day is now called for under conditions of normal furnace practice. A furnace making from 1000 to 1200 tons per day should therefore have 400,000 to 550,000 sq. ft. of heating surface in its stoves, or something more than 100,000 sq. ft. per stove where the usual number of stoves is employed.

It will thus be seen that the 1000-ton furnace, together with the increased necessity for gas conservation, has called for still further improvements in stove construction.

Some builders have preferred to obtain the additional heating surface needed by building much larger stoves or by the use of extremely small checkers, as small as 1 in. in diameter. Such procedure is objectionable, however, for two important reasons. It makes imperative very complete cleaning of the gas, and the holes are susceptible to closing from any very slight shifting of the checkerwork. The resistance to the travel of the gases



FIG. 4 COLUMNS AND GRIDS SUPPORTING CHECKERS

through them is also high, requiring more pressure at the burners or a much higher stack to overcome it.

For years past the authors have recognized the fact that smaller checker openings and thinner checker walls were absolutely necessary in order to get the much greater heating surface required into our present small stove shells. At the same time they have realized that it was absolutely impossible to support such small checkers on firebrick arches and girder tiles.

In order to get the required heating surface in small checkers and to support them safely and permanently, they have developed a new type of checker support in the form of columns and high-chrome-nickel alloy-steel grids which support the entire bottom surface of the checker structure so that it is quite impossible for the checkers to shift or break down unless from the failure of the grids. (Fig. 2.)

These grids, on the other hand, are designed for a factor of safety of 15 at a temperature of 1000 deg. fahr., so that the possibility of failure of the checkers at their supports is practically eliminated.

It will be noted that each part of this stove is self-supporting and free to expand vertically, independently of all other units, so that there are no undue strains set up by unequal expansion of the different parts.

The authors have avoided the use of extremely small checkers and at the same time secured a very large treating surface by the combination of desirable features, especially the fact that the checkers may be started very near to the bottom of the stove and at the same time have unusually large passages for the gas and blast.

The checker tiles are so interlocked with each other that they cannot become displaced laterally, and at the same time the individual checker tiles are free to expand vertically without interfering with each other.

The manifold advantages of this construction will be readily apparent in that it offers strength, durability, and unobstructed access to the checker openings, while at the same time permitting a very large reduction in the more or less unutilized space below



FIG. 5 TOPS OF GRIDS SUPPORTING CHECKERS

the checkers. More even distribution of the gases over the entire area of the checkerwork also results, which tends to increase both the capacity and efficiency of the stove. (Fig. 3.)

With blast-furnace gas of the cleanliness obtained from washers of the Feld type—now in common use throughout the industry—and containing less than 0.25 grain of solids per cubic foot, it is quite feasible to use checkers as small as $2^3/_4$-in. square or 3 in. round. A 21 × 100-ft. stove lined with $2^3/_4$-in. checkers and the authors' metal bottoms and having $2^1/_2$ in. of silocel insulation will have 108,000 sq. ft. of heating surface. Four stoves of such design are probably ample to provide adequate blast-heating capacity for the largest-size furnaces while maintaining a high degree of thermal efficiency, although such a happy combination of equipment as a 1000-ton furnace and stoves of 432,000 sq. ft., has never been tried and for this reason the authors are unable to make a positive statement. However, four 22 × 100-ft. stoves would provide 480,000 sq. ft., which is just that much more. (Figs. 4 and 5).

Some actual performance data from a battery of stoves of this construction may be of interest. The following data and the illustrations which pertain to this subject are in connection with three stoves of this type which we have recently designed and built. It was found unnecessary in this case to reline the fourth stove, the very large capacity and high efficiency of the new metal-bottom stoves of only 21 × 90 ft. making it possible to get all the blast temperature desired from only three stoves remodeled in this fashion when blowing 47,000 cu. ft. of blast and making nearly 700 tons average output. Use of the remaining old stove has been entirely discontinued. (Fig. 6.)

Gas-Cleaning Equipment

Closely associated with the improvements in stove construction and particularly with the use of blast-furnace gas at coke plants and in the mills, has been the matter of better gas cleaning.

Previously the complete elimination of dirt from the gas had been held to be necessary or justifiable only when the gas was intended for use in gas engines. The Theisen drum-type scrubber was used in almost every case for this purpose. Where the gas was to be used for stoves or boilers the Feld type of washer has been most generally adopted. The cleaning efficiency of these two types of cleaners may be roughly compared by the statement

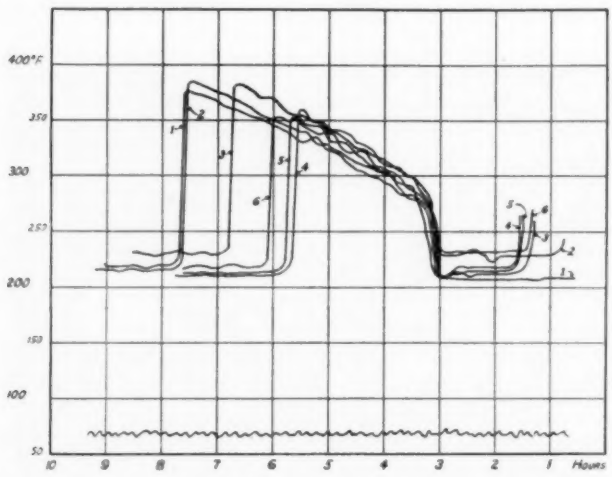


Fig. 6 Blast-Furnace-Stove Stack Temperature Chart With Successive Cycles Superimposed

that the Feld-cleaned gas will usually contain less than 0.2 grain of dust per cu. ft., and Theisen-cleaned gas less than 0.02 grain.

With Feld- or McKee-cleaned gas it has generally been possible to operate hot-blast stoves continuously for several years, oftentimes the entire period of a blast, without interruption for cleaning. Inasmuch as Theisen cleaning has been highly expensive, the power requirement alone being more than five times that of the Feld or McKee type, it may be readily understood why it has not come into common use.

The disintegrating type of cleaner is one which was developed in Germany in an effort to obtain gas of Theisen cleanliness with a smaller expenditure of power. Several German firms now make disintegrators, including Theisen, and they have come into very general use abroad. While accurate data are not yet obtainable on American practice, it is probable that their cleaning effectiveness wil compare favorably with that of the drum-type Theisens. It appears that the power requirements will be about one-half that used in the old-type Theisens.

Another very promising development lies in the application of Cottrell precipitators to wet gas. A small plant of this type used in conjunction with primary tower washers has consistently cleaned down to about 0.01 grain per cu. ft. Apparatus of this type has a distinct advantage in power and water consumption over any mechanical scrubbing device. The following table, based upon the best obtainable data, may be of interest:

	Final cleaning—non-boosting		
	Drum-Theisen	Disintegrators	Cottrells
Water required per 1000 cu. ft. of gas, gals.	10	10	0.6
Total power required per 1000 cu. ft. of gas, kw-hr.	0.1137	0.0564	0.0084

The first cost of the Cottrell equipment completely installed will compare favorably with that of the disintegrators.

It will be noted that while rapid progress is being made in the development of better and more economically operated devices, the final cleaning of gas is still quite an expensive procedure and is as yet clearly justified only where ultimate cleanliness is an operating necessity.

There is, undoubtedly, however, a rapidly growing demand for thoroughly cleaned gas brought about by the extension of its use for underfiring coke ovens and in the steel mills, which is for this country a comparatively recent innovation.

At the same time renewed and vigorous efforts are being made to improve the efficiency and decrease the investment and operating costs of both the primary and the final gas-cleaning equipment, as for instance, the Cottrell development mentioned above, and the authors have well-founded hopes of the early development and availability of further improved devices in both of these fields.

Use of Blast-Furnace Gas at Coke Ovens and Steel Plants

Until quite recently, in what might be called normal American practice, the use of blast-furnace gas has been limited generally to the heating of stoves and the generation of steam. In some few plants it has been used in gas-engine-driven blowers and electric-power generators.

The past year has witnessed the inauguration in this country of underfiring coke ovens with blast-furnace gas. Two merchant furnace plants having adjacent by-product coke ovens are now using this means to increase their revenues by releasing large additional amounts of coal gas for sale to domestic users. Both are in the Chicago district, where there is a large and profitable market for this commodity. The construction and operation of these plants have been fully described in the technical press, and the results are known to be highly satisfactory.

Coke produced in ovens heated by blast-furnace gas has proved to be of more uniform quality than when the same ovens were fired with coal gas, and the blast-furnace practice is said to have been improved as a result of the change.

Where there is a domestic market for coke-oven gas the utilization of blast-furnace gas for firing coke ovens will undoubtedly be most productive of profit. About 25 per cent of the gas produced in the manufacture of a ton of pig iron can be utilized in this fashion, which will be approximately 35,000 cu. ft. per ton of pig iron produced, and will replace and release for sale about 6000 cu. ft. of coal gas. Assuming for the latter a market value of 25 cents per M cu. ft., the gross revenue from this source will be therefore about $1.50 per ton of pig iron. If the blast-furnace gas in question were withdrawn from boilers, it would have had to be replaced with other coal, coke, breeze, oil, or some other fuel. Assuming it were replaced by pulverized coal at $3.50 per ton, its cost of replacement would be approximately 42 cents. Cost of cleaning, boosting, and distributing will not exceed $^1/_2$ cent per M cu. ft. The net credit per ton of pig iron under these conditions would therefore be $0.91 per ton of pig, and in so far as this was formerly surplus gas which was being wasted, the saving would be about $1.32 per ton of pig iron.

Where the blast furnaces are operated in conjunction with and in proximity to both coke ovens and steel plants, another very profitable means of utilizing the gas is in open-hearth furnaces, soaking pits, and heating furnaces throughout the mills. When mixed with various proportions of coke-oven gas, it has proved to be the ideal fuel for these purposes.

It has been the privilege of the authors to participate in the design and construction of the first complete American plant of this type. This plant has now been in operation for several months. It is stated by the officials of the company operating

it that the results to date have been highly satisfactory both from the standpoint of operation and the economies obtained. A brief description of the installation may be of interest.

Primary cleaning of the gas is done in the dry-dust catchers and a wet washer of the Feld or McKee type, which reduces the dust content to less than 0.2 grain per cu. ft. The gas going to the stoves and boilers is taken directly from the McKee washer.

Gas for the steel works is further cleaned in Theisen disintegrating washers to a cleanliness of approximately 0.02 grain per cu. ft. The gas is also boosted by the disintegrators to 20 in. of water pressure and discharged into a 1,000,000-cu. ft. gas holder near by. From the gas holder it is distributed through one main to the open-hearth plant, and through another to the steel mills. These are overhead mains and they have total lengths of 1200 ft. and 1400 ft., respectively.

In close proximity to each point of consumption has been located a station for the mixing of the two gases. The design of the mixing devices is such that any desired proportions of the two gases may be obtained and automatically maintained. Thus the B.t.u. value of the gas may be correctly adjusted to the particular requirements of each steel-making or heating unit.

The peculiar advantages of a mixed-gas system will be readily apparent when one stops to consider that probably no two of the various heat treatments which are employed in the manufacture of steel products will call for the same intensity and quantity of heat, and that mill practice must be greatly improved when the B.t.u. value of the fuel employed is accurately adjusted to each particular use. The two fuels, blast-furnace gas of about 90 B.t.u. and coke-oven gas of about 550 B.t.u., furnish an admirable means of accomplishing an ideal fuel by mixing.

At the plant in question the B.t.u. values in use at the present time are as follows:

Open-hearth furnaces	250
Merchant-mill heating furnace	325
Soaking pit	90
Sheet and pair furnaces	150
Sheet box annealing furnaces	125
Sheet galvanizing pots	90

The monetary advantages which may accrue from the installation of a complete system for cleaning and for distributing blast-furnace gas to the steel mills may only be arrived at by a very careful examination and study of the economic and operating conditions which obtain in the plant and locality under consideration. Among the factors which must be taken into account are the quantity and quality of the blast-furnace gas available and its value for steam raising based upon the cost of steam coal; the supply of coke-oven gas and its commercial market value, if any; the fuel practices in the plants under consideration and the cost of producer gas or other fuels which it is expected to supplant with mixed gases; the cost of buying or producing electric power; and, finally, the cost and practicability of constructing the cleaning and distributing system.

In general, however, it may be stated that where there is no domestic market for coke-oven gas, any steel plant having two or more blast furnaces and by-product coking capacity sufficient for their needs cannot afford to overlook what in many cases will prove to be a highly profitable means of reducing costs, increasing output, and improving the quality of their product.

Investigations by the Bureau of Mines

Another matter which the authors would commend to the most careful consideration of all blast-furnace men is the blast-furnace research work which has been done during the last few years by the very competent engineers of the Bureau of Mines, with headquarters at Minneapolis, Minnesota.

In their opinion this is the most important work which has been done by any group of men for the advancement of the art of smelting iron ore since the days of Sir Lowthian Bell, and all should study most intimately all of the information and data which have been gathered by these engineers.

In this connection all should realize the fact that these investigators cannot be expected to be intimately familiar with all of the practical operating facts and difficulties of blast-furnace construction and operation, and for this reason it is particularly the obligation of the operators to study the information thus made available and draw the logical conclusions therefrom to the end that the construction of blast furnaces and the manufacture of pig iron shall profit in the highest possible degree from this valuable work.

The last reports available seem to the authors to indicate very conclusively the desirability of larger stock-line areas so as to reduce the velocity of the gases through the stock at this point, and also the desirability of a greater distance from the stock-line wall to the edge of the bell so as to make the percentage of voids more uniform from the periphery to the center of the furnace and decrease the wide difference in gas velocity between that near the wall and that at the center of the furnace.

Steeper bells, a greater travel in opening, and a quicker movement of the bell from the closed to the open position are also potent factors in securing a more uniform radial distribution of the coarse and fine particles of the stock and the resultant effects of more uniform rate of gas travel and lower flue-dust losses.

If these results are being secured and may be secured to a greater degree by the use of larger top diameters and bells smaller in proportion to the stock-line diameter and other changes proposed, then the authors would reasonably expect that their latest blast-furnace construction with large stock lines and proportionately smaller bells should give a higher thermal efficiency than was previously possible and the whole industry should benefit thereby.

It is sincerely hoped that there may be a still closer cooperation between blast-furnace operators and builders and the Government engineers, and a much more intensive study of the reports made available from time to time by these experts of the Bureau of Mines. The authors are fully convinced that such cooperation and study will result in very great benefit to the whole industry.

The authors are satisfied that by all odds the greatest benefit that can result from papers of this sort is due to the discussions which they bring forth. For this reason they hope that a free discussion may follow, and particularly that any who may disagree with anything in the paper may fully state their opinions, as they want, above everything else, the most accurate information obtainable.

Conclusion

From the collection of information which the authors have been able to present, it is evident that the industry of making iron is making remarkable strides both in output and in the efficiency of the operation.

All have reason to feel proud of the achievements of the past year or two, and although the industry is now in a high state of development and efficiency, the authors sincerely hope that other valuable improvements may be forthcoming in the near future.

Discussion

H. J. Allen.[3] Regarding size of stacks, it is interesting to note that the large increase in diameter of hearths, originating

[3] The Bourne-Fuller Company, Cleveland, Ohio.

about two years ago, and following very closely certain experimental data presented by the U. S. Bureau of Mines showing that the penetration of the blast of oxygen in front of the tuyeres was limited very much below the radius of the hearth. This paved the way for belief that larger hearth furnaces would function properly without fear of unequal blast distribution at the hearth.

Larger furnace tops have been used recently in the South with success, although Northern operators have been rather hesitant to try them. But from the results of the three furnaces referred to in the paper, they will probably become almost standard.

Another interesting trend in design is that furnaces using Lake ores, and constantly being blown harder, are becoming straighter —that is the bosh angle is larger and, in some cases, the batter of inwall smaller. Heights of furnaces are not increasing.

The turbo-blower has become the standard, and units are being built today of so large a size that a single reciprocating engine to do the same work would be entirely impracticable.

Regarding boilers burning blast-furnace gas, installations are operating efficiently where the air for combustion is preheated from the heat in the stack gases, thereby increasing the boiler efficiency.

As the first and most economical use of blast-furnace gas is in stoves, some construction has come in for considerable development in recent years. For this reason operators are constantly striving for higher blast temperatures, combined with high stove efficiency. This, the authors have stated, has been made possible by improved gas-cleaning equipment, aided by pressure-type burners, in connection with stoves having checker openings of small area and comparatively thin walls. Two-pass stoves of center-combustion-chamber design, the writer believes, will become more predominant, as the pressure burner tends to give uniform distribution of the gases in the checkers, and the construction of the masonry in the center-combustion-chamber stove in the checker section is much freer from operating troubles than in a side-combustion-chamber stove. A stove of this design is just being completed by the concern with which the writer is connected, and uses what is known as a Whelpley checker, which is one of T shape. The shell size of this stove is 20 ft. × 90 ft., has 4-in. square checker openings, with $2^1/_2$-in. checker walls, and will have a heating surface of 68,000 sq. ft.

The authors' metal support for checker areas in stoves, the writer believes, is one of the notable contributions to stove design in recent years, and furnace operators are watching its performance with much interest.

Another principle of stove design that is being applied in some new stoves is decreasing the areas of the checker openings at intervals from top to bottom of the checkers, in order to approximate a constant rate of travel of gas and air through the checkers.

One development that deserves mention in dealing with blast-furnace equipment is the new type of electric-driven mud gun, developed by Mr. John Hopkins of the American Steel and Wire Company, of Cleveland. This gun has made it possible to stop the iron hole without reduction of the blast, and to entirely complete the operation through remote control. Mr. Hopkins has also developed a diverging-nozzle tuyere, for which it is claimed that lower blast pressures are obtained and tonnage is somewhat increased.

HARVEY B. JORDAN.[4] Under the discussion of the furnace stack, the authors give recommended dimensions for a 1000-ton furnace. While, to the writer's knowledge, there is not at the present time a plant having a furnace with these particular dimensions, he is sure there are many in agreement generally on this proposal.

[4] Superintendent, Central Furnaces, American Steel and Wire Company, Cleveland, Ohio.

With the authors, the writer pleads that due consideration be given the apparent value of a greater stock-line diameter. The $^{13}/_{16}$-in. slope per foot of the inwall may perhaps be a sacrifice, and yet commendable if necessary to obtain the 20-ft. stock-line diameter.

In conjunction with top enlargement, some operators and engineers have increased the annular space, which is in itself a progressive step, theoretically and practically sound. There has been a little doubt in the minds of some of these men as to the percentage of value which should be attributed to each of these changes. Just recently, one of the leading operators, without any other adjustment, increased the annular space. This was done on a new, modern furnace after it had been operated for a satisfactory period on what many considered a standard space. The result of this one change was very profitable. The limit to which such enlargement can be profitably extended on the large, modern top type of blast furnace, operating on Lake ores, is not definitely known.

With the advent of larger hearths and larger stock lines, due consideration must be given the vertical measurements. The opportunity which the writer has had to observe the operation of two furnaces of the same cubic-foot capacity operating on the same material, but with a difference of 5 ft. in height, urges him to suggest that the large furnaces be constructed to at least 100 ft. in height.

For several months the Carnegie Steel Company, at its Ohio Works, has had in successful operation a furnace with a 25-ft. hearth diameter, a 19-ft. stock-line diameter, a 14-ft. bell diameter and a height of 103 ft. While he does not have at the present time the practice figures on the operation of this furnace, the writer ventures to state that the results obtained are most pleasing to the management.

In late years there have been many changes in bosh design and construction. These modifications, together with various methods of operation, have produced important results. We do not yet have for use on the bosh, however, a novel design of water cooling which offers to the operator the advantages without the disadvantages of the present-day system. That eventually such difficulties will be largely overcome should not be an exaggerated prediction.

Gas cleaning is an expensive necessity. The economical recovery of the iron-bearing material in the gas-washer water is an important problem. The burdens upon these two operations should be decidedly reduced by the progress in the reduction of flue dust, to which phase of the gas problem our attention should also be attracted.

Among the many improvements in the auxiliaries of the blast furnace are the developments of the clay guns for closing the iron hole. The most noteworthy of these in the past year was the introduction of the pedestal, crane, and screw gun, all operated electrically. By the use of electricity, many have witnessed an innovation in iron hole practice.

The past few years have also shown greater activity in the thought and practice relative to the introduction of the air into the furnace. Among other things developed have been the diverging-nozzle tuyeres. These tuyeres have been used in some instances with satisfactory results by providing an aid in the initial directing of the gases.

Of late there has been a gradual but decided trend toward the use of large hot-metal ladles. The National Tube Company at the National Works plant has operated two furnaces, using large ladles, making the novel scrap production of three pounds per ton of pig iron.

On the subject of the use of blast-furnace gas at coke ovens and steel plants, and the marketing of coke-oven gas, the authors have offered a comprehensive account. The National Tube

Company at its National Works is pumping surplus coke-oven gas into the old McKeesport gas basin. The gas is then withdrawn from this basin as required. This procedure necessitates a suitable geological structure. The efficient utilization of such gas in some districts, when in many it is partially wasted, is a progressive step of which those responsible may be justly proud. These performances should furnish the incentive for the purpose of utilizing such gas in other districts where practicable.

The six-day utilization results in one-day inefficiency. Where such one-day inefficiency is not excessive, and entirely for the benefit of labor, these low losses are justified. However, generally the talented of our industry have before them, yet unsolved, the seventh-day utilization of these by-product gases.

T. L. Joseph.[5] As regards larger stock-line diameters, through the cooperation of several blast-furnace operators the Bureau of Mines has been able to explore the conditions inside of several furnaces. This is done by drilling a hole in the side of the steel shell and extending water-cooled tubes into the charge at varying lengths from the periphery to the center of the furnace.

A very great difference was found, for example, in the velocity of the gas at the wall and at the center of the furnace, a great variation of temperature, and a substantial variation in gas composition. And in regard to the gas velocity at one furnace it was found that in the center near the stock line the gas was moving at a velocity of over 400 ft. per sec., whereas along the wall it was traveling about 80 ft. per sec. What has been found particularly confirms the wisdom of this step of increasing the diameter of the stock line.

Arthur J. Boynton.[6] The paper has discussed in an interesting manner the question of correct diameter of the blast-furnace stock line in proportion to the other dimensions of the furnace and has very correctly stated that these proportions are now of extreme interest to blast-furnace designers and operators. The possibility of obtaining thoroughly satisfactory distribution of the materials making up the stock column becomes progressively less as increase in the diameter of the stock line proceeds, and it therefore seems quite probable that other means besides those of variation of proportion between bell and stock line will require to be employed with the wider stock line which is now definitely being tried. The writer will give his view as to an alternative method of obtaining correct distribution.

There existed a period of several years' duration within which the shape of the stock of the blast furnace above the bosh was, in its essential dimensions, standardized. The total height of the furnace was in almost every case 90 ft., the bosh diameter was 22 ft., the stock-line diameter 17 ft., and the bell diameter 13 ft. The taper of the stock line was $^3/_4$-inch per ft., and the chief variation in the furnaces of the Steel Corporation as built in different plants lay merely in the respective lengths of the upper cylinder at and above the stock line and the lower cylinder immediately above the bosh. These proportions were gradually arrived at with a stationary bosh diameter and height of furnace by successive increases in the diameter of bell and stock line. Taking into consideration the conditions with respect to blast temperature and quality of fuel, furnaces of this size, once the diameter of the hearth had been increased to 18 ft. 6 in. or thereabouts, showed an all-around performance with respect to fuel economy and production of flue dust which was probably more satisfactory than the average of records which have been made since. With respect to the production of flue dust, records are in existence which, while confined to the production of dry-flue-dust recovered from dust catchers, form a guide as to the relative excellence of practice then and now.

When the tendency toward larger hearths than could be combined with a 22-ft. bosh manifested itself, the stock line was for a considerable number of years allowed to remain at 17 ft. diameter and the bell at 13 ft. The tonnage of the furnace increased rapidly as the bottom dimensions were enlarged and the relative area of top to bottom became very considerably less than it had been formerly.

The limitation of increase in the top diameter is based on many and complicated considerations. The distribution of ore over the working annulus of the furnace depends in part on the physical character of the ore with respect to fineness and moisture content and in part on admixture with the coke charge or coke and limestone charge. With the same size of charge the radial distance over which ore is scattered is independent of the size of the top and the proportion of the total diameter of the furnace which is occupied by the ore charge decreases as the diameter at the stock line is enlarged. Apparently, also, the width of the working annulus at the tuyeres is independent of the diameter of the furnace and to some considerable extent independent of the volume of wind blown, the latter of these two statements being in accordance with the researches of the Bureau of Mines. These researches and their determination that the active working part of the hearth was confined to an annulus in front of the tuyeres seem to explain the inability of the American blast furnace to operate otherwise than with an ore charge arranged in annular form at the top. Supplementary to the researches of the bureau are, however, the observations which have been made at the furnace itself. It has long been known that the temperature of the center of the hearth or of any point inside the active area in front of the tuyeres was lower than the temperature at the tuyere area. This can be determined by a steel bar put into the furnace, as was often done. The driving of the bar also gave a practical illustration to the limit of combustion, as subsequently determined by the Bureau of Mines.

In view of the lower temperature and apparent lack of motion existing in the middle of the hearth, the placing of ore in the furnace above this relatively cool and inactive hearth zone naturally results in coldness of any iron which melts and trickles to the bottom of the hearth within it. It has accordingly long been understood that ore projected into the middle of the furnace was productive of a cold hearth.

The researches of the Bureau of Mines, and particularly those of S. P. Kinney, have determined a relatively rapid rate of descent of the materials composing the charge within the annulus around the furnace which corresponds both to the active tuyere zone and to the location within the furnace of the ore as charged. This condition even with a relatively small hearth existed within the writer's observation. For example, when a blast furnace is filled cold, the material in the top of the furnace always forms an inverted cone with the high point next to the walls. If the furnace is blown for any considerable number of minutes without recharging, this cone disappears and the height of the stock at the walls of the furnace is no higher than in the center. Further blowing without charging brings the level of the stock at the walls to a point considerably below that at the center until finally, if a furnace is blown down to the bosh, the remaining stock will form a cone with the apex in the center of the furnace. The writer also has witnessed a demonstration of the same thing at times when a brick lining was being torn out of the blast furnace and has seen a dimple in the brickbats over each tuyere as the men hoed the bats through the tuyere breasts, the height of the brick column in the furnace being at the time 40 ft. or so above the tuyeres.

It is accordingly apparent that recent scientific research and

[5] U. S. Bureau of Mines, Minneapolis, Minn.
[6] Vice-President, H. A. Brassert & Co., Chicago, Ill.

practical experience, not only with the blast furnace, but with structures of similar shape used for gravity bins, coincide in their indication that flow of material within the blast furnace is most rapid along the walls. Exact knowledge as to the speed of the center of the stock column is lacking. It is, however, presumably very slow and decreases in speed from top to bottom.

The researches of the Bureau of Mines, excepting in some cases, where less than ordinarily steep batters of the inwall of the furnace have been employed, indicate a flow of gas within the central part of the furnace which is more rapid, hotter, and higher in carbon-monoxide content than is the gas which comes up through the outer annulus which contains the ore. This condition is natural and inevitable. The coke gets to the center of the furnace by sidewise travel down the inverted cone, to which reference has previously been made, and is consequently composed of the coarser coke in the furnace charge. The resistance to the passage of gases in the outer annulus is increased by the presence of ore and by the relatively small size of the coke. The consequent effect is that of a central chimney which forms a bypass around the ore for the reducing and heating gases.

This bypass naturally increases in extent and in seriousness as the diameter of the stock line is enlarged. As previously stated, the width of the annulus within which ore may be placed by the ordinary bell-and-hopper action is not increased by increase in size of the stock line, although it may be increased in some considerable measure by enlarging the size of the charge. The limit beyond which increase in the diameter of the ore annulus is undesirable is related to the diameter of the lower working annulus in front of the tuyeres, which in all probability corresponds to what has always been called "penetration."

From the foregoing statements it seems evident that such an increase in the stock line of the furnace as would be required to restore the former proportions of the 22-ft. bosh diameter and 17-ft. stock line diameter are not possible without a radical change in the stock column. The desirable diameter of stock line based on the old experience would be as high as 22 ft. in some of the larger furnaces, either built or proposed. This is at least 2 ft. beyond any stock-line diameter which has been projected within our knowledge. It is accordingly apparent that, if mechanical considerations permitted, it would be desirable to build a firebrick pier in the middle of the furnace and to operate an annular structure. While such a conception is evidently not practicable, it is interesting to note that the United Gas and Improvement Company of Philadelphia, while operating water-gas sets through tuyeres instead of a bottom grate extending across the entire bottom, found a decided advantage in exactly such a central firebrick pier and patented the construction. The reason for this was that, with the limited amount of action obtained from the tuyeres of the water-gas set, the center of the charge was inert, relatively cold, and not gasified.

So far as can be seen, the only possible means of producing a condition in the furnace which prevents excessive central bypassing of gas with a stock line of desirably large diameter lies in the creation of a central column within the furnace with relatively small coke screened from the main charge and put into the furnace from time to time as required. The amount of coke so charged and its relative fineness are matters which must be determined by experience. Evidently the coke must not be so fine as to be impervious to a sufficient amount of gas to bring it to the hearth temperature in the course of its descent. Practically, it should be large enough to accomplish this purpose and no larger. The amount of gas which travels up the center of the furnace would then be regulated by the sizing of the central column of coke.

The size of the central column within which such coke is charged is also subject to variation by changes in the hopper which forms the means of distributing the finer coke. The amount of fine coke required will depend on the relative rapidity of descent in the middle of the furnace to that along the walls.

After sufficient experience with respect to size of coke and diameter of the central coke column has been gained, it would seem to be entirely feasible to restore the relative dimensions of stock line and bosh which formerly existed or in all probability to bring these dimensions closer together than was formerly possible. It must be remembered that in the former practice the open central bypass for hot rich gas existed in a very considerable degree.

A consideration of relative areas is interesting in this connection. The area of a 17-ft. circle is 227 sq. ft. The area of a 22-ft. circle is 380 sq. ft. The area of a 12-ft. circle is 113 sq. ft. It would consequently be possible with a 22-ft. stock line to blind the area of a central 12-ft. circle and still gain in effective area within the top of the furnace. This, however, is not proposed, since a considerable amount of gas would necessarily proceed up the center of the furnace. Based on relative areas, which would then become more truly effective than at present, the decrease in the average velocity of gas would be over 40 per cent, while the effect on flue-dust production would be many fold greater than the decrease in velocity.

Another advantage of the proposed construction, and one which with coarser ores would form its primary object, is the enforcing of contact between ascending gas and descending ore in as large a proportion as possible. The effect of this gas contact is certain to be favorable to the fuel requirement of the furnace.

An additional benefit to be derived will be the elimination of the smaller sizes of coke from the annulus in front of the tuyeres. This is universally recognized as desirable to a point which makes comment superfluous.

Owing to the conditions which have generally existed, it has become customary to reckon the capacity of a stove installation in terms of square feet of heating surface without reference to other conditions of construction and operation. It is evident from the foregoing discussion that the effectiveness of heating surface is influenced, not only by its disposition in the individual stove with reference to size of passages and effective thickness of brick, but also by the number of stoves within which it is disposed. The larger the units and the less their number the more effective will be a given amount of heating surface, other things being equal. It is consequently necessary to take into consideration, not merely number and size of units, which should be as large as the general condition of the operations require and as few as continuity of operation permits, but also the disposition of the heating surface within the individual unit. When all the conditions of heat transfer are considered, it is possible to build a stove with maximum heating surface which will be less effective than one of smaller heating surface, but designed with more skilful consideration of size of flues, thickness of walls, and velocities of gas and air.

With reference to the number of stove units which should serve a single furnace, theory and experience both show that the use of more than two stoves is attended with considerable heat losses as well as unnecessary expenditure for equipment. With regard to the first consideration, it is evident that such advantages as are gained by high velocity of heating gases are almost completely lost if the gas is divided between two or three stoves instead of being concentrated in one, and that heat transfer will be correspondingly less rapid and satisfactory. The volume of gases under standard conditions of pressure and temperature will be only about 54 per cent of the corresponding volume of the air. This difference is lessened in a considerable degree by the higher average temperature of gases compared

with that of air, so that the actual average velocities vary by only approximately 31 per cent. Where two or more stoves are used on gas, design for the sake of high velocities ceases to apply to the heating phase of stove operation. It is also evident that losses through radiation and leakage are considerably reduced in a two-stove operation.

The hesitancy in building for two-stove operation is based on two practical difficulties which have always existed, but which are now definitely capable of being overcome. The first of these results from the use of gas insufficiently cleaned to guarantee the operator against the necessity of cleaning a stove and repairing fluxed brickwork within the course of a blast-furnace campaign. The second difficulty is structural and, generally speaking, results from high temperatures in the bottom of the stove, which result in deterioration of the supporting construction members to a point which requires repair. The first of these difficulties requires only the fine cleaning of gas by means which have been demonstrated, while the second requires merely the construction of stoves of correct design and sufficient size to prevent excessive temperatures at the bottom. Trouble due to leakage of valves is too insignificant to justify the erection of a third stove, especially when one considers that a change of hot or cold blast valves requires taking the wind off the furnace whether there be two stoves, or three, or four. Where three stoves exist, or where they are provided as a precaution against interruption to operation, they should, if correctly designed, be operated with one stove on air and one on gas, with the third stove in reserve. In this way the full benefit of smaller radiation losses, less leakage, and wear and tear on valves can be realized.

A practical illustration of the principles which have been discussed may be cited from experience which has been had in many locations where furnaces have been provided with five stoves. In almost every case, at one time or another, the attempt has been made to increase the blast temperature by blowing through two stoves at once. Such a procedure doubled the quantity of hot brickwork in contact with the air, at the same time reducing the velocity of the latter by half. It is believed that no case exists in which any marked increase in temperature was obtainable by blowing through two stoves, even though the temperatures of the brickwork were capable of being maintained by excessive use of gas in three stoves. Generally speaking, the blast temperatures where two stoves were put in blast at the same time was less than that possible of attainment with the blast going through one stove. This phenomenon, which appeared mysterious to many operators a decade ago, is entirely in accordance with the discoveries of recent research.

The provision of cast-iron regenerators in series with the firebrick stoves is complicated in operation and mechanically undesirable. They also fail to increase the maximum temperature within the stove and for this reason are of little use in the creation of higher blast temperatures. The purpose which was sought to be fulfilled by the metallic construction is in large degree served by small passages and high velocities in the cooler portions of the stove, combined with an increased heating surface. However, the question as to whether the refractory stove designed with reference to scientific principles will continue to be used as at present will depend in considerable measure on the future blast temperatures which the operator desires to generate. With cool blast-furnace gas and cool air going into the stove, the temperature of combustion is not sufficiently above that of the hot blast to make the stove a truly efficient heat exchanger. The evident remedy lies not in further changes within the stove itself, nor in continuous increase in size, but in the preheating of the air and gas for combustion by means of waste chimney gases. By this means the temperature of the burned gases entering the checkers can be raised to any point which the brickwork will stand, and hot blast of any conceivably desired temperature may be readily created by the use of stoves considerably smaller than would otherwise be required. It is probable that the addition of heat exchangers to stoves of present size or of less size than some stoves at present projected will be found to be the most economical means of obtaining high blast temperatures. It is to be noted that such a procedure, in common with the other developments which have been discussed, is only feasible with use of fine cleaned gas from which all dirt practically capable of adhering to the surfaces of the stoves has been removed.

J. L. Mauthe.[7] There really is no rigid rule for large bell and stock-line clearances. The experimentation to date has not been of long enough duration to determine the practicability of increased clearances. Justifiable as it may seem, it will only be determined after a period of low, medium, and high blowing rates to ascertain, if possible, the critical velocity where best results are obtained.

All experienced blast-furnace operators know the effect of getting too much ore into the center of the furnace. What the possibilities will be with an increased stock-line diameter, maintaining the present size of the big bell, will only be determined through experiments, as stated above, unless the Bureau of Mines shall be able to conduct experiments that will determine definite results. We who have tinkered with fine materials are not quite ready to agree that this increase will eliminate the flue dust, inasmuch as in furnaces with the same lines and under the same blowing conditions a change in the ores has decreased the flue dust from 600 lb. to less than 100 lb. per ton of iron.

In the writer's opinion, we have not analyzed our raw materials thoroughly enough to determine whether an increase in the stock lines will be sufficient to reduce the flue dust. Some reliable test is needed, such as a plasticity test or something similar, which will determine the flue dust producing abilities of an ore.

F. G. Cutler.[8] With reference to the employment of a larger stock line, it may be stated that the Tennessee Company has had in operation for a year two new furnaces that are identical as regards bosh and hearth diameter, and reports are to the effect that the one with the larger stock-line diameter produces very much less flue dirt.

Authors' Closure

Mr. McKee. The difficulties which have been predicted by one or two of the discussers due to increasing the stock-line diameter may be found to be overestimated. It is doubtful whether there will be any serious difficulties since the experiments by the Bureau of Mines have indicated how we can very accurately determine and provide for the degree of distribution of the ore across the diameter of the furnace to such an extent as may be found desirable for these changes.

In other words, if the differential between the bell and stock-line diameters is increased, the distribution of the ore will be somewhat more toward the center of the furnace. If the bell is operated more quickly, there will be an added effect in this direction; if the bell is made steeper, the same result will be obtained, and likewise if it is made to travel farther, and that indicates, it is believed, that we can have a very accurate control over the distribution of the ore toward the center of the furnace, and the experimentation recommended by Mr. Mauthe to determine just what distribution of the ore toward the center of the furnace is

[7] National Tube Company, Pittsburgh, Pa.

[8] Tennessee Coal, Iron & Railroad Company, Ensley, Ala. Mem. A.S.M.E.

best for the different kinds of ores, is something that will be most valuable to the industry.

The suggestion made by Mr. Jordan that the furnaces be made still higher than suggested in the paper is a valuable one.

The present author was the engineer who laid out the first and up-to-date the tallest furnaces in the country, or in the world, namely, the original Youngstown furnaces, which were 107½ ft. from the iron docks to the top platform. These furnaces at that time were a distinct failure. They were too high for the practice current at the time, but that does not in any sense indicate that with the advance in the art they are too high for the conditions as they are today, and it is believed that experimentation and careful observation to determine the best height will be a very valuable contribution to the art.

The authors wish to thank Messrs. Frame and McKnight for their assistance, and are in agreement with their statements.

IS-52-4

Protection of Movable Material-Handling Bridges Against Windstorm Hazards

By C. O. BURTON,[1] DULUTH, MINN.

Present-day speeds and efficiency in handling enormous tonnages of bulk materials, such as coal, coke, ore, crushed stone, etc., have been made possible only by the development of modern types of material-handling bridges. Such bridge structures now seem to be absolutely essential in the operation of coal docks, blast furnaces, and steel plants for unloading, stocking, and reloading the bulky raw materials of industry. Since they are so indispensable, it follows that their protection against all possible causes of loss or damage becomes of vital importance. By reason of their great length and height, such bridges present large wind-resistive areas, making them particularly susceptible to windstorm damage. This paper discusses some of the factors of design and construction that affect the safety of these structures, and describes and discusses the special equipment designed to increase the operating safety of bridges. The author tells of the development of material-handling bridges, and their constructional elements, so that the more specific features of bridge protection against storm damage may be better understood.

GROWTH of the coal-handling industry at the head of Lake Superior, ports of Duluth and Superior, undoubtedly parallels in many ways the early and modern development of coal- and ore-handling equipment in other industrial sections of this country. Prior to about 1880 coal received by boat at the Duluth-Superior harbor was handled almost entirely by hand. It was hoisted, first by horses, later by steam hoists, in tubs filled by hand shovellers in the hold of the boat, then transferred by wheelbarrows or hand-operated dump cars to the receiving dock. In 1883 eight wooden bridges of about 150-ft. span were erected at one dock, with steam-driven hoists and rope-operated trolleys handling a one-ton tub. These bridges were moved along the dock by hand power, and were the first at the head of the lakes.

Steel bridges followed and electrical drive began to replace steam operation. The first electrically operated bridges were apparently those erected at Duquesne, Pa., in 1895. Not until 1902, however, did the first electrically operated man-trolley bridges appear. Three stocking and reclaiming bridges were then built for the North Western Fuel Company of Superior, Wis., and these are said to be the first of their type in the world. These were used for stocking and reclaiming only, the unloading of the boat being done by a group of four hoisting towers.

In 1909, Heyl and Patterson built the first man-trolley bridge at the head of the lakes for doing all the work connected with unloading from the boat, stocking, reclaiming, screening, and loading into cars for shipment. This bridge had a 5-ton clam-shell bucket. Soon after 1910 a large number of other bridges were built at the head of the lakes, nearly all of them of the modern man-trolley type. One of these bridges on the same dock of the North Western Fuel Company, on which the first three man-trolley bridges were erected, is claimed to be the largest coal bridge in the world. It has a single span of 551 ft., is 712 ft. long, 86 ft. from dock floor to lower chord of the truss, and about 130 ft. high to the top chord. This bridge alone, with only two men per shift for its operation, can unload a 10,000-ton boat in less than 24 hr. While this development was going on in the coal-handling industry, similar advances were in progress in other parts of the country, notably with the handling and storing of iron ore at the great iron and steel centers. As a definite illustration of this growth, there are now in the eight states of Minnesota, Wisconsin, Illinois, Missouri, Indiana, Michigan, West Virginia, and Ohio about 387 movable coal-, stone-, and ore-handling rigs, of which 210 are bridge structures and 177 towers or unloaders of various types. Over 50 of these bridges are located at the head of the lakes, Duluth and Superior. Many more bridges are included in Pennsylvania, New York, Massachusetts, Maryland, and other Eastern states, as well as a considerable number in Canada, and a scattering distribution in foreign countries.

The cost of a modern man-trolley bridge of medium or large size will range from $200,000 to $300,000 and upward. Individual docks or steel plants may operate as many as 9 or 10 of these structures. This great investment alone represents sufficient reason for the adoption of all practicable safeguards against operating disasters. But the original cost of the bridge alone cannot represent the true industrial value of the structure. Many coal docks and many blast-furnace plants depend so completely on continuous bridge service that they would be forced to curtail or abandon operations in case of a major wreck. The indirect losses sustained through loss of use and curtailment of dependent industrial operations, as in case of a blast furnace or steel plant, may easily be of greater magnitude than the direct and tangible losses.

That bridges do suffer severe damage from windstorms is convincingly proved by yearly records of losses sustained. In the period of 1920–1927, there were at least 27 complete or partial bridge wrecks; many around the Great Lakes. Between 1923 and 1928, inclusive, ten docks in Ohio, two in Illinois, and one in Minnesota suffered actual bridge losses aggregating approximately $1,375,000. A single storm, that of June 2, 1924, caused three losses in Ohio alone, totaling $465,000, and only eight days later, June 28, another storm in Ohio caused two additional losses amounting to $350,000. One dock at Lorain was struck by both storms with a loss of $75,000 in the first and $200,000 in the second. Other losses elsewhere in the United States and in Canada during the same period would greatly increase the above totals. These figures represent only the cost of rebuilding or replacing the wrecked structures. They do not include any of the indirect losses.

Windstorm damages or wrecks are caused in several ways. The most general cause of wrecks lies in the tendency for one end of the bridge to move faster than the other due to unequal

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NOTE: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

wind-pressure distribution and unequal driving and controlling forces. The bridge does not remain square with the runways, and eventually reaches a position where the runway rails fail or the legs supporting the truss twist and collapse. This is usually the result of movements of the bridge in the same direction that the wind travels. Unless the bridge is fitted with adequate modern-control equipment and safety devices, it is apt to be blown along the runways much faster than normal operating speeds; even with no power applied, and only the brakes released, most bridges will drift rapidly with the wind. Without adequate control and slowdown action, the application of brakes at such high speeds is almost certain to result in disastrous skids, since the brakes are usually effective enough to lock the wheels. This is especially true on wet or icy rails. Fig. 1 illustrates a bridge wrecked by lack of control and excessive skewage.

A bridge may be blown along the runways and crash into another bridge, with great damage to both. It may be blown along and crash into the runway bumpers with sufficient violence to overturn or collapse. In one case a bridge was wrecked by a wind striking it almost end on, while at a standstill. In this case, oscillations were set up which finally resulted in the failure of a poorly designed and defective runway, and a total collapse.

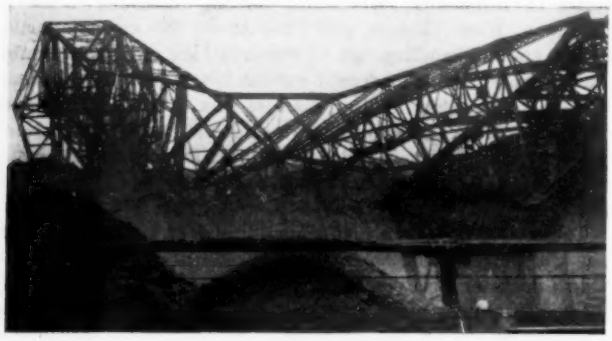


FIG. 1 WRECK OF RIGID-LEG, CABLE-HAULED BRIDGE AT CLEVELAND, OHIO, IN 1923

Storms of sufficient violence to twist bodily or overturn bridges are fortunately rare. It is very doubtful whether any movable bridge structure, however well fitted with safety devices, can successfully resist the direct force of a major tornado. The fluctuating currents and whirls result in tremendous alternating stresses; the enormous lifting power of the rotating center currents may conceivably lift the structure bodily and crash it down to a total collapse. The illustration, Fig. 2, shows a wreck apparently of this sort which has created a great deal of discussion. The portal-type pier leg and the truss were moved almost directly endwise into the vessel slip. There appears to have been almost no dragging of the trucks across the portal railroad tracks, with the exception of one broken rail. The bridge-runway rails themselves seem to be intact. The operator and another eye-witness to the wreck declared that the bridge seemed to rise and lurch through the air. This bridge had some rather large floor areas at the portal leg, and it is assumed that sufficient lifting force was exerted on these and other platform and cab areas to lift a large part of the 800 tons representing the weight of the bridge. Such lifting power is almost incredible, but competent observers and the evidence of the wreck itself point to that conclusion as being probably correct.

Winds of any intensity whatever are the result of difference in atmospheric pressure between two or more points on the earth's surface. These pressure variations may be local or may extend halfway across a continent, and are caused primarily by temperature differences at various points. The greater the pressure gradient, the more violent the atmospheric movement tending to equalize the pressure differences. In the northern hemisphere, a succession of high- and low-pressure areas travel from west to east. Alternate low- and high-pressure areas cross the United States with more or less regularity at the rate of about two per week. As a result of the earth's rotation, the inward movement of the atmosphere is not directly toward the center of the low-pressure area, but in a spiral counter-clockwise direction toward the center. Since the whole low-pressure



FIG. 2 SKEW-OPERATED, PORTAL-TYPE PIER LEG, MAN-TROLLEY BRIDGE, WRECKED BY TORNADO, SOUTH CHICAGO—1928

area is moving toward the east, the resultant wind velocities are generally much greater in the eastern and southern areas of this great circulatory movement. Such a movement is correctly defined as a "cyclone." The same term is often loosely and incorrectly applied to any violent storm, as a tornado, thunderstorm, or line squall. The term "cyclone" in its proper meaning does not refer to a destructive storm. The circulation of the atmosphere away from an area of relatively high pressure is spirally in a clockwise direction away from the center, and is properly termed an "anticyclone." The wind movement away from a high-pressure center is seldom as violent as the movement toward a low-pressure area.

Due to local differences in pressure, atmospheric disturbances of minor area, but often of extreme violence, are superimposed on the relatively harmless cyclone. Such storms are classed as thunderstorms, tornados, line squalls, etc. Local heating and cooling of the atmosphere result in violent horizontal and vertical movements of the air. Areas of heated air begin to rise and cold air from above flows down to replace it. The resultant changes in temperatures and pressures acting on air more or less saturated with water vapor result in snow, rain, or hail, sometimes accompanied by relatively undirectional, violently rotating windstorms. The violently rotating storm of small area and relatively short path is called a "tornado." Tornados are characterized by extremely great pressure drops at the center of the whirl, the whole spinning mass of air moving at a relatively slow speed along a path which does not usually exceed 40 or 50 miles in length. The rotating velocities in a tornado may exceed 200 m.p.h., and in conjunction with the very low pressure at the center are responsible for strange explosive and lifting actions. Nothing but the most massive structures can resist the forces existing in a major tornado.

The wind velocity in any storm is seldom even approximately constant. Continually recurring velocity changes and gusts occur, so that momentary stresses are imposed on wind-resistive

areas far greater than might be expected from a consideration of the ordinary weather-bureau velocity reports, which until very recently were invariably average velocities over one-, two-, or four-minute periods. Even now only a few prominent stations have apparatus capable of recording gust velocities. Fig. 3 is a reproduction of a velocity chart made with the Burton anemometer. The chart was run at high speed (each curved time division represents 15 sec.) so that the momentary variations and gust effects could be studied. Fig. 4 shows a section of a chart when operated at normal speed, each curved time line representing 15 min. This illustrates very well the extremely rapid velocity increase that characterizes many storms. The broad record band with its many peaks also graphically portrays the unsteadiness and gust phenomena of storms. Anemometers located quite close together often show wide differences in simultaneous wind velocities. For this reason, one anemometer cannot be expected to offer protection for a large group of bridges, nor can reports telephoned from a distant weather-bureau station have any local value.

Since the destructive energy of a storm varies approximately as the square of the wind velocity, the extreme importance of properly determining gust velocities in the protection of bridge structures is readily apparent. For safety, bridge-traversing movement must be discontinued when dangerous gust velocities are reached, without regard to average velocity determination.

Before proceeding to discuss safety in design and safety devices and attachments, a brief description of the principal elements of a modern material-handling bridge may be useful. Such a bridge in its essential form is made up of a steel truss structure supported on structural-steel legs which are equipped with multiple-wheel trucks for propelling the bridge along the dock runways. The truss may be of various standard forms such as the Howe, Warren, Pratt, and Catenary, riveted or pin connected. The supporting legs are usually of the pier and shear type, which names indicate fairly well their respective functions. The pier leg gives the truss lateral, longitudinal, and vertical support; the shear leg gives only lateral and vertical support, with little or no longitudinal stability.

The pier leg is usually either of "V" or portal type (also called "A" frame). The "V" type of pier leg resembles a skeletonized wedge, with the trucks and connecting sills representing the tip of the wedge and the broad upper end attached to and supporting the truss. This type of leg travels on a single rail or on two rails relatively close together. The portal-type leg may be like a square tower or more or less like the letter "A." This is supported on track-runway rails of very wide gages, 20 to 40 ft. or more, with two or three standard railroad tracks between them. This type usually has conditioning or screening apparatus built into it, and is equipped to load into railroad cars below. The shear leg, as its name would indicate, is simply a pair of structural legs or "shears" with a connecting sill and trucks at the lower end running on a single or double rails. The upper end supports the truss. The bridge truss may have cantilever extensions on one or both ends, or may have a folding "boom" extension at the slip end, which can be raised to clear the mast of boats; the trolley runs out on this extension when unloading the vessel.

Bridge construction may be further differentiated into "rigid" and "skew" types. Rigid bridges in general are those in which the pier and shear legs are rigidly attached to the truss, and are generally driven by horizontal and vertical shafting and spur and bevel gearing from one large motor located at some convenient point along the truss. Since both ends have a common drive motor, they presumably should travel at the same speed, and should keep square with the runways. Jaw clutches are commonly fitted in the horizontal drive shaft so that accidental minor "out-of-square" conditions can be corrected. The safety advantages and disadvantages will be discussed later.

The skew-type bridge, as its name would indicate, does not necessarily remain square with the runways. The trusses of

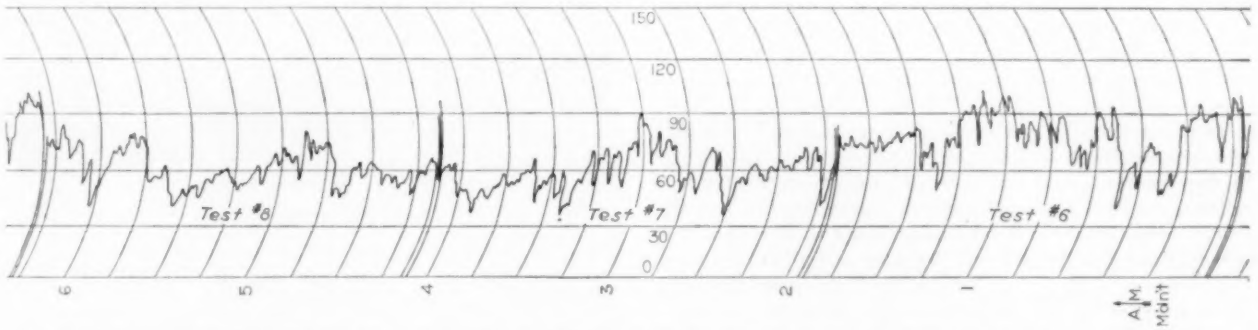


Fig. 3 Burton Anemometer Chart Showing Gust Variations in a Windstorm—Chart Run at High Speed—Each Time Division 15 Seconds

such bridges are supported on "skew" plates or turn tables on the tower or "V"-frame pier legs, centered by a king pin so as to permit horizontal rotation of the truss relative to the leg. Hanger eye bars, ball joints, or similar supports are provided for supporting the truss at the shear leg, usually also centered by a king pin or its equivalent. The two legs are equipped with single or multiple motors, with no shafting or other device for mechanical equalization of travel rates. Some bridges of this type may be operated as much as 40 or 50 ft. out of square with the runways. This flexibility has both operating and safety advantages and disadvantages which will be considered later. Fig. 5 illustrates skew-operated "V" pier-leg bridges, and Fig. 6 is a closer view of the "V" leg.

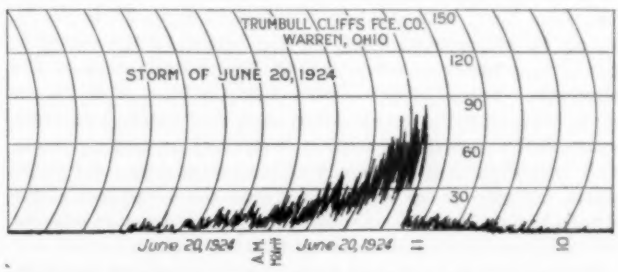


Fig. 4 Burton Anemometer Chart Run at Normal Speed—This Record Made at Warren, Ohio, at Time of Destructive Lorain Tornado, June, 1924

The hoisting and trolleying machinery proper has little

bearing on the protection of the bridge against storm hazards. Some bridges have rope-operated trolleys and hoists, with the motor and operating drives in a stationary machine house in the truss. The operator's or hoister's cab is also fixed, often in duplicate, so that either unloading from a vessel or working on stored material may be done to the best advantage. Practically all large modern bridges are of the man-trolley type, with the



FIG. 5 TWO SKEW-TYPE, V-FRAME, PIER-LEG, MAN-TROLLEY BRIDGES ON LAND STORAGE DOCK. MINNESOTA STEEL COMPANY, DULUTH, MINN.

hoist and trolley mechanisms supported in a movable carriage which also carries the control equipment and the operator, hence the name "man (riding) trolley." This arrangement is preferred because the operator moves with his load, has a better view of his work, and because this system eliminates the great lengths of wire rope and the multiplicity of sheaves and guides required by the rope trolley.

Many bridges are equipped with built-in or trailing car-loading and screening plants for properly preparing "run-of-pile" coal for domestic and industrial use. While these may add to the weight and stability of the bridge, they also add to the wind-resistive areas, so that they indirectly affect the safety of the structure under storm conditions. A trailing-type plant is shown in Fig. 7, attached to the shear leg of a rigid, shaft-driven bridge.

The actual forces exerted against the structural elements of a bridge are difficult to determine. The variety of rolled shapes and built-up sections employed, the varying angles of attack of the wind, the influence of mutual interferences of the eddies and gusts involved, all serve to complicate the problem. Large machine houses and enclosed cabs and screening plants all greatly increase the areas exposed to the wind. The general design of the bridge, therefore, has a considerable influence on its safety under storm conditions.

Good design demands a sufficiently broad wheel base to insure stability. Wind pressures are assumed at from 30 to 45 lb. per sq. ft. of projected area, and the heights of the center of gravity and center of pressure must bear proper ratios to the wheelbase. Recent bridges designed on the basis of 45 lb. per sq. ft. should prove exceptionally safe and stable. Machine houses in the truss should be kept as small as possible consistent with accessibility of the equipment they enclose. Screening plants, hoppers, or car loaders built into the lower part of the pier and shear legs add stabilizing weight and traction without adding disproportionately to wind-resistive areas.

Rigid, non-skew types of bridges with single- or multiple-motor drive, but shafted for simultaneous movement of the two ends, are often considered by their users to be safer in storms than skew-type bridges. It is claimed that they are easier to keep square with the runways, and that simultaneous starting and stopping are facilitated. There is probably smaller chance for controller failures or operator's mistakes to put the bridge into a hazardous position. On the other hand, due to the very rigidity of its construction, a breakage of the drive mechanism at one end, or a skid of one end on slippery or icy rails, may severely strain or damage the structure through its very lack of flexibility.

Skew-type bridges are intentionally made flexible enough to withstand large degrees of skew with relatively small hazard, but there is greater danger of excessive skews developing since either end travels independently of the other at a speed determined by the type of motors and control used, by the operator's manipulation of the controllers, by the driving or restraining force of the wind, by unequal friction losses in the drive mechanism, and by variable track conditions. Such bridges, for maximum safety in motion, must be equipped with properly designed controllers, motors, and brakes, and with auxiliary devices such as skew indicators, skew limit switches, and wind gages.

All bridges, whether rigid or skew operated, must have safe propelling equipment. All parts must be designed with ample

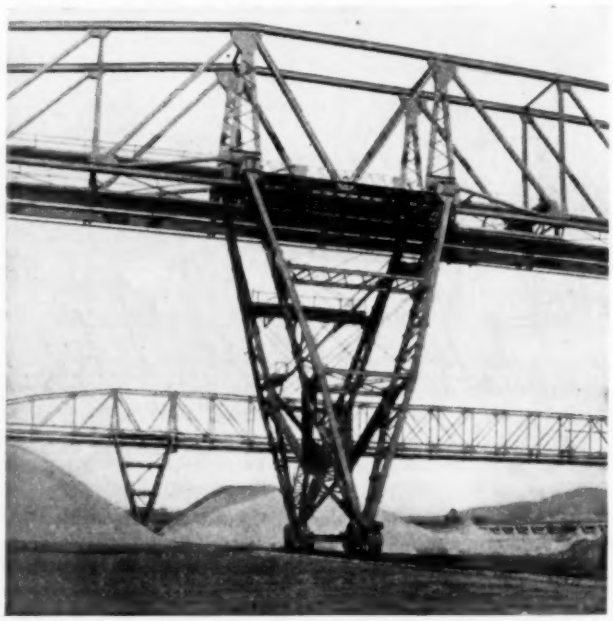


FIG. 6 V-FRAME PIER LEG, 100 PER CENT TRACTION DRIVE
(Same bridges as Fig. 5.)

factors of safety, and protected against accidental damage. Slow propelling speeds are highly desirable from a safety standpoint, though not so satisfactory from the operator's viewpoint, especially if frequent movements or long runs are necessary. Travel speeds range from 20 or 25 ft. per min. up to 150 ft. or more. Speeds of 20 to 50 ft. are very slow; from 50 to 80 ft. per min. is generally considered about right for most bridges. Propelling speeds over 80 ft. per min. are rather fast and require exceptional care and unusually good driving and braking equipment for maximum safety.

Bridges may be traction driven by the supporting wheels, either by a single motor shaft connected to part or all of the wheels, or by one or more motors at each end geared to part or all of the wheels, either end being free to move independently of the other. The mechanical drive may use bevel gears, spur-gear trains, worm gears, or various combinations of these. Flat-pitch worm gears are often considered advantageous on account of their so-called "self-locking" features against overhauling winds. Modern forms of worm gearing, however, are apt to be so efficient that too much dependence cannot be placed on their resistance to overhauling. The bridge of Fig. 7

is of the single-motor shaft-driven type with worm final drive, but only four of the ten bridge-supporting wheels at this leg are so driven. Fig. 8 illustrates another worm-driven bridge, but in this case all wheels are drivers, a pinion on each worm-wheel shaft meshing with two drive-wheel gears. Each four-wheel truck has its own motor. Fig. 6 shows a straight spur-gear drive to all supporting wheels. If wheel traction drive is used, maximum safety in motion and at rest demands that driving and braking effort be applied to 100 per cent of the supporting wheels.

Another method of moving the bridge along the runways is by means of cable haulage drums attached to the bridge trucks or sills. The cables are securely attached to the end of the runways, and make a number of traction wraps around the drums. Theoretically this may appear to be a safe drive, but in practice several wrecks have occurred which detract from its apparent safety. The cables must be kept tight enough to insure good frictional adhesion to the drum. The cables must be kept well oiled or doped to prevent rapid rusting and weakening. This weather-preservative treatment seriously reduces their adhesion to the drum, and encourages slipping. If two independent cables are used on each runway for each bridge, it is difficult properly to equalize the load carried on each, especially the braking loads. If several bridges operate on the same runways, each must have its own cable system, which in the aggregate requires considerable space and also increases the danger of fouling and other damage to the cables. The elasticity and

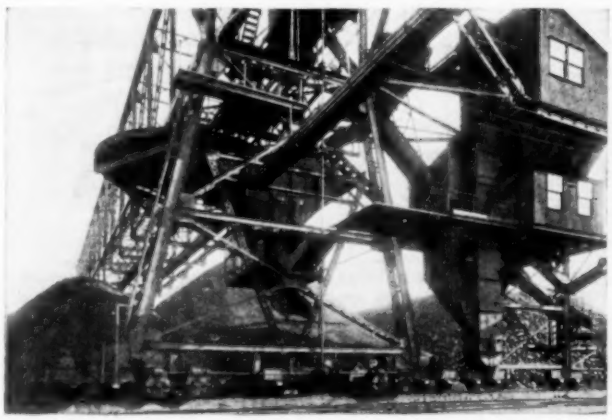


Fig. 7 Rigid-Type, Non-Skew, 40 Per Cent Worm-Driven Bridge. With Trailing Screening Plant—Note Automatic Rail Clamps. M. A. Hanna Coal & Dock Co., Superior, Wis.

"stretch" of cables on a long dock are also objectionable. This system is becoming obsolete; the author is not aware of any recent installation.

Both wheel and cable-drum traction systems of drive are subject to slippage, while in motion and on starting and stopping. As a means of avoiding such hazards, one of the prominent bridge-engineering companies is now engaged in modernizing seven old bridges at South Chicago by the installation of a rack-and-pinion system of drive. This system has been used on other bridges, and while it increases the dock cost, it undoubtedly provides a safer means of propulsion. A substantial rack is installed on each runway, into which are meshed two large-toothed drums or pinions. Each pinion has its individual driving motor, and all supporting wheels are idlers. All braking and driving loads are taken by the rack and pinion. Assuming that the drive mechanism is of ample strength and reliability, this method of propulsion provides the most positive control yet developed for bridges, even under severe storm and weather conditions.

However perfect the mechanical details of the driving mechanism, the selection of proper motor, control, and braking equipment is also essential. The propelling motors may be for either direct or alternating current. If for direct-current operation, the choice lies between series-, shunt-, and compound-wound motors. Alternating-current motors may be either squirrel-cage or wound-rotor construction.

Series motors are particularly well adapted for starting and running under heavy loads. The current-torque characteristic of such a motor is shown by curve (*a*) of Fig. 9. However, series motors have no inherently stable speed characteristic, and under light loads, excessive speeds may be attained. This condition exists when a bridge is moved in a strong following wind. Cutting in starting resistance has no appreciable slowdown effect if the wind is strong. The speed-torque characteristic is shown by curve (*a*) Fig. 10. This unstable speed char-

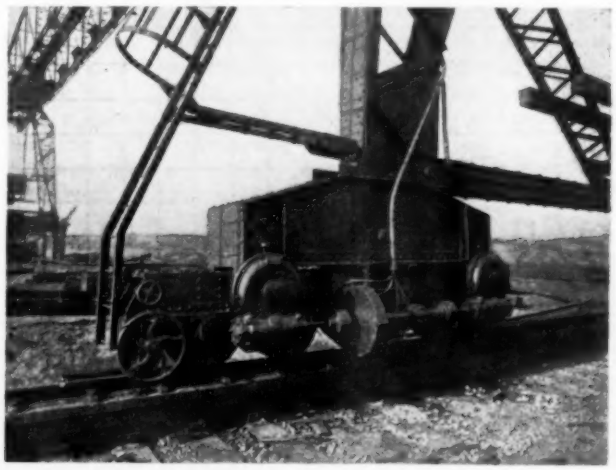


Fig. 8 Shear-Leg Truck Details—Worm Geared 100 Per Cent Traction Drive—Note Hand Wheel for Setting Manually Operated Rail Clamps

acteristic may be corrected by the use of a shunted armature connection, as shown in Fig. 11 (*b*), and a slow positive speed thus obtained under light loads as shown by curves (*b*) and (*c*) in Fig. 10. This connection also provides extremely effective electric braking in strong overhauling winds. Another advantage of series motors is that they divide the load equally when two or more motors are used on each end of a bridge.

Shunt-type motors, on the other hand, provide very stable and definite driving speeds, both under heavy and light loads and under severe overhauling wind conditions. The starting and heavy-load torque characteristics are, however, not as satisfactory as those of a series motor. Current-torque characteristics are shown by curve (*b*) of Fig. 9. The speed-torque curve is illustrated at (*d*) Fig. 10. The flatness of this curve indicates that unequal load distribution may be expected between two or more motors geared to the same load. The shunt motor can be run at reduced no-load speeds only by the use of armature-shunt circuits similar to those used with series motors. The slowdown action is not quite as effective, because there is no series field to be strengthened by the armature-shunt current. Series resistance in the armature circuit alone can reduce the travel speeds only with a positive load; as the load decreases, this speed rises to the normal no-load value, and rises still higher under overhauling conditions. As a matter of fact, in a strong following wind, the bridge will run faster on the starting points than with the controller full on.

The compound motor has some of the advantages as well as

disadvantages of both shunt and series motors. It has a better starting torque and a better current-torque characteristic curve than the shunt motor, but not quite as good as a series motor. It has a definite no-load speed determined by the shunt-field strength, but the no-load speed is considerably higher than at full load, and under following winds the bridge speeds are higher than with shunt motors. The current-torque curve (*c*) of Fig. 9 and the speed-torque curve (*e*) of Fig. 10 show its inter-

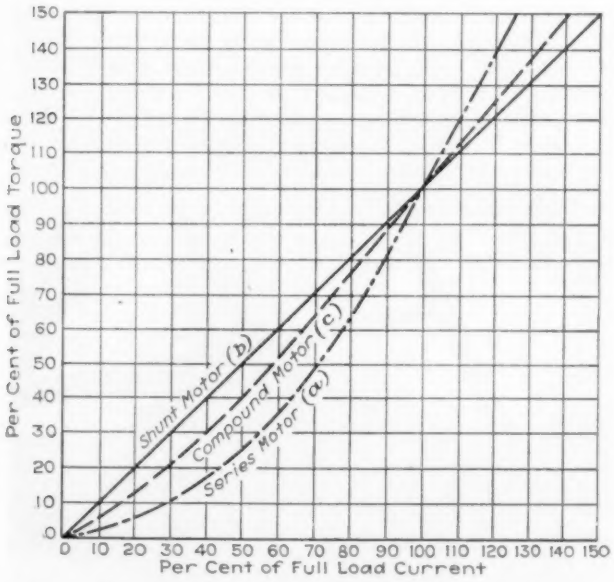


FIG. 9 CURRENT-TORQUE CHARACTERISTICS OF D.C. MOTORS

mediate characteristics. This motor may also be used with an armature shunt more effectively than with the shunt motor, not so well as with the series motor. The division of load between two motors on the same drive is also better than with the shunt motor.

The series motor is most generally used as a bridge-propelling motor, because of its desirable heavy-load characteristics and because by the use of armature shunts it can so readily insure the slow creeping speeds that are necessary in making smooth stops

Alternating-current motors for bridge drive are generally three-phase induction type, with either squirrel-cage or wound rotors. The large motor used on single-motor shaft-driven bridges is almost certain to be of the wound-rotor or slip-ring type, since it has much better torque characteristics at heavy loads. Multiple motor drives are preferably of rotor-wound type also, because of better performance, though squirrel-cage motors might be used. Induction motors have limiting no-load speeds a little higher than full-load speeds. Strong overhauling action of following winds causes the speed to rise only a little, accompanied by a regenerative-braking action. Unless an auxiliary supply of direct current is available, slow down or dynamic braking points are not practicable, with induction motors.

While many bridges have been built for alternating-current operation, the modern trend seems to be toward the use of direct-current power supply at 250 to 600 volts, derived from a dock substation. The characteristics of direct-current series motors which make them valuable for bridge-propelling drive also make them more suitable for hoist and trolley, as well as for boom hoists and other bridge auxiliaries. Screening plants may be considered an exception to this, because the loads are fairly steady and the absence of commutator and brushes is an advantage of the alternating-current motor.

It seems scarcely necessary to suggest that only weatherproof totally enclosed motors of heavy-duty type should be used for multiple-motor drives, where the motors are mounted directly on the trucks. Single-motor drives are usually provided with a roomy enclosure, which frequently houses other machinery, and these motors do not need to be totally enclosed or weather-proof.

Perhaps equally as important as the correct choice of driving motors is the selection of proper control equipment. Desirable features of a good control system include the following points: smooth starting and acceleration, moderate speeds in strong overhauling winds, powerful slowdown operating points to be used for stopping, and have suitable overload and no-power protective features. Slowdown speeds 20 per cent to 30 per cent of normal full-load speeds will permit smooth application of the brakes with little shock or jar.

Manually operated drum-type or face-plate controllers are no longer considered desirable. Unless the operator is par-

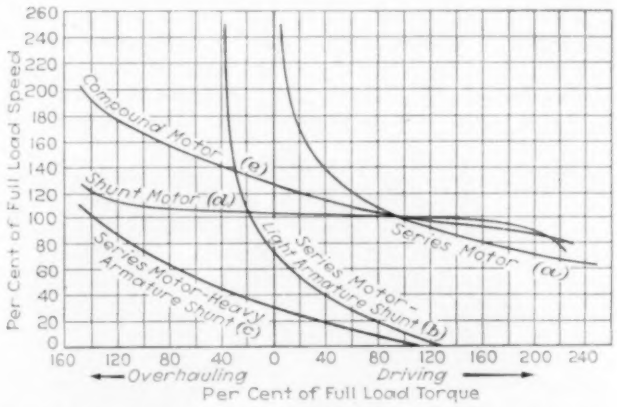


FIG. 10 SPEED-TORQUE CHARACTERISTICS OF D.C. MOTORS—ALSO SHOWING EFFECT OF ARMATURE SHUNT CONNECTIONS

ticularly careful, they do not give smooth acceleration, and the fact that they usually do nothing more than notch back starting resistance in stopping means that they are not apt to slow the bridge down before it is stopped by the brakes. Manual controllers may be equipped with "armature-shunt" slow down points, which greatly improve their action, but many of the good features of magnetic controllers are lacking.

Modern magnetic controllers provide for uniform acceleration to full speed without undue shock or jerking. They also provide under-voltage and overload protection, and should be arranged so that after a power failure the controller must be centered before movement can again be resumed. This is necessary to prevent a bridge from starting up and running away in case the controller should be left "on" and the operator leaves his post. This particular protective feature may also be obtained by the use of platform or dead-man switches, on which the operator must stand while the bridge is in motion. Such platform switches ought to be so designed that a lazy operator cannot render them inoperative by blocking or weighting them down. Spring-centered master controllers are also useful, since they must be forcibly held in the operating position, and return to center if released. A desirable feature of some controllers is a low-torque first operating point, designed to take up the slack in the drive gearing without positively starting the bridge in motion. Secondary points provide for slow (armature-shunt) operation, and then acceleration controlled by current

or timing relays to full speed. Sometimes more than one point of slowdown operation is provided so the graduated slowdown action similar to the accelerating points can be obtained. The maximum slowdown point should positively provide a very slow final speed before power is cut off the drive motors and the brakes are applied.

Keeping the bridge square or within certain reasonable degrees of skew is one of the most important functions of a good magnetic-switch controller. Skew limit-switch arrangements, which may be simple or elaborate in construction and operation, are a positive necessity on skew-operated bridges.

The older types of limit switches were arranged so that they merely cut power off the bridge whenever the skewage exceeded a fixed amount. Generally the correcting circuits were not affected, so that by reversing the controllers the bridge could be squared up and the limit switch automatically reset ready for forward movement again. Improved types of limit switches have graduated control, first slowing down the leading end by series resistance, and then armature-shunt operation, giving the lagging end a chance to catch up. If it gained on the leader, full-speed operation would again be resumed. Should the slowdown action be insufficient and the skewage still increased,

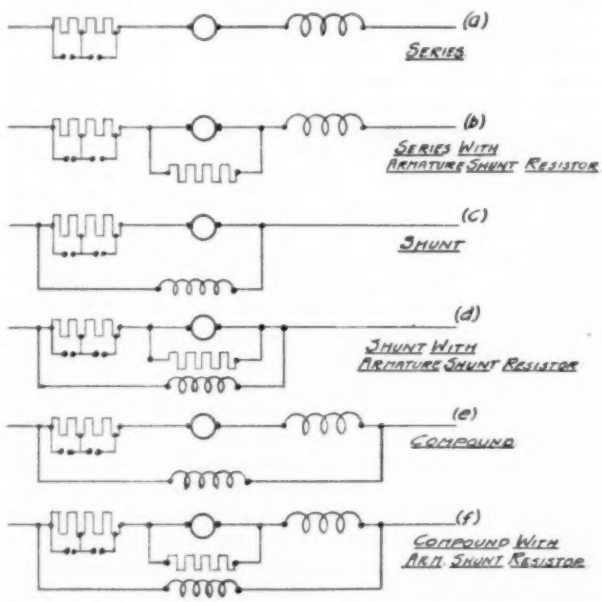


FIG. 11 ELEMENTARY CIRCUITS, DIRECT-CURRENT MOTORS

a final limit point was reached at which both ends were stopped, and movement could be had only in the corrective directions.

Still more elaborate and effective systems of skew control are in use. One form developed and patented by the author and used on a number of bridges, operates as indicated in the following brief description:

The special magnetic-switch contactor board controls both ends of the bridge, subject to the manipulation of two master controllers and a special multiple-point skew limit switch. The two ends of the bridge may be moved simultaneously or independently either in the same or opposite directions, either at slow armature-shunt speeds or automatically accelerated to full speed at the will of the operator, as long as the bridge remains in the normal "center" zone of skewage. As soon as this degree of skew is exceeded, the leading end is automatically slowed down, and the lagging end either maintained at or brought up to full speed. If for any reason the two ends are moved in opposite directions, both ends slow down, but continue to move. If the slowdown action is still insufficient to correct the skew, a point is reached where the leading end stops, while the laggard end still continues at full speed. At the same time that this action takes place (through the operation of the skew limit switch), circuits are established whereby all control by the operator is eliminated, and the bridge squares itself up and stops. Stopping is accomplished by armature-shunt action, then a no-power dynamic-braking circuit is established as the power is cut off and the brakes are applied. After the automatic squaring action is instituted no manipulation of the controller by an excited operator has any effect until the bridge has stopped and both controllers are centered preparatory to a fresh start. The underlying aim and intention is to keep the bridge square as long as possible under any unusual load or wind condition; if certain limits are still exceeded, then to make squaring up and stopping fully automatic and non-interferable by the operator.

For bringing the bridge to a complete stop, and for preventing the rotation of the drive mechanism even under heavy winds, some form of brake operating on the armature shaft or some part of the drive mechanism is absolutely essential. These may be operated by magnets or solenoids, by compressed-air cylinders, or by mechanical devices. Electrically operated brakes are used in practically all cases, so connected that they are released by power circuits established as the operating controller is moved to the running points, and set at once when the controller is centered or the power supply is interrupted. These brakes may take the form of wheels on the armature shaft, with friction bands or shoes set by gravity or powerful springs and released by a solenoid. Other brakes are of disk type in which alternate disks are keyed to the armature shaft and to the brake case, and pressed together on application of heavy springs. To be effective, the brakes must, of course, be fully weatherproof or properly housed. They should release quickly and completely, and should set smoothly, but without grabbing or sudden locking, yet hold so securely as positively to prevent rotation of the drive mechanism even under great wind pressures.

As mentioned under the discussion of controllers, it is also highly desirable that slowdown control points be provided, so that the brakes do not have to absorb all the energy of the moving bridge. In some cases, timing relays or dashpots have been used to secure graduated application of the brakes, but these should not be necessary if good slowdown action is provided. If the bridge does not have 100 per cent drive, it is quite desirable to have groups of idler wheels geared together and locked by effective brakes. One hundred per cent braking is particularly desirable in stormy weather.

In order to increase the safety of slowdown points and holding brakes, some bridges are equipped with track sanding devices. These may be air or electrically operated, and should be electrically heated to keep the sand dry and free from clogging or caking. The sand used must be screened carefully so that lumps or pebbles do not foul the sand valves. The sand pipes should discharge the sand as near the wheels as possible. If sanders are kept filled and in good order, they are valuable aids in making safe stops. In some cases, they have been interconnected with the travel controllers, so that on the slowdown points sand is automatically fed to the rails.

Some form of skew indicator should be installed on every bridge that is even slightly skew operated. Under certain conditions, an operator might be able to estimate the degree of skew without an indicator, but under storm conditions or at night such estimation would be too hazardous. Skew indicators usually take the form of a graduated arc or dial, over which a mechanically operated pointer or hand moves as the bridge is skewed. The indicator, of course, should be mounted so as to be

plainly visible to the operator as he stands at the travel controllers.

So far, the equipment considered has been concerned principally with the propulsion of the bridge. A class of devices must now be considered, which from the standpoint of bridge protection against windstorm hazards are unquestionably of superlative importance. This group of devices includes manual and automatic rail clamps, manual and automatic wheel chocks and wedges, safety struts, sprags, spuds, and mooring cables, all of which are designed for the sole purpose of immovably anchoring the bridge. Some of these devices are released each time the bridge is moved, and reengage immediately when the bridge movement is stopped, either by the controllers or because of power failure. Others are intended to be used only in the



FIG. 12 EARLY-TYPE, MANUALLY OPERATED, SELF-TIGHTENING RAIL CLAMP—MEAD-MORRISON MANUFACTURING COMPANY

event of an approaching storm, or in case the bridge is to be idle for some time, as between shifts or overnight.

Rail clamps are devices for gripping the head of the rail powerfully, so as frictionally to increase the resistance of the bridge against wind forces. Originally, they were all hand operated, with relatively limited holding power. Some were of plain vise construction, with smooth or roughened jaws. Others had self-tightening features which increased the restraining ability when the bridge moved slightly. One of the earliest self-locking clamps is shown in Fig. 12. This was initially applied by the action of the toggle links and two traveling nuts, and had self-tightening cams which tended to hold tighter as the force against the clamp increased. However, bridge structures are large and movements are so frequent that it is physically impossible to set such clamps every time the bridge is moved. They require considerable time to apply, and in a sudden storm it would be hard to get them tightened quick enough. Manual clamps of various sorts are still used as auxiliary locking devices, but power-operated clamps are now almost universally used, and are much more powerful and effective than hand clamps.

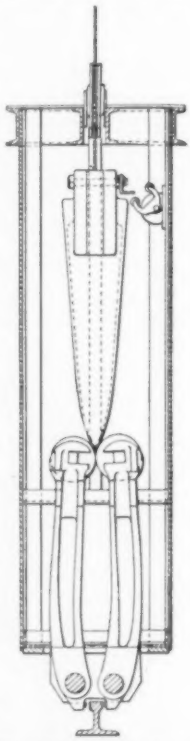
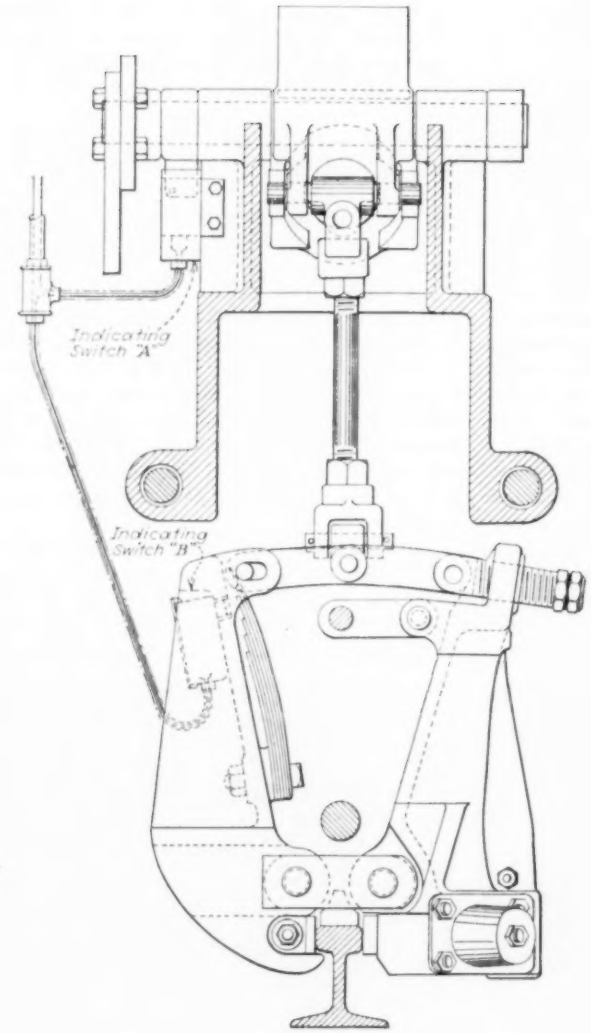


FIG. 13 GRAVITY-SET, DIRECT-PRESSURE, AUTOMATIC RAIL CLAMP—HEYL & PATTERSON, INC.

All modern automatic rail clamps are arranged so that they are released by some power-operated device (winch or a compressed-air cylinder), and are set independently of any power circuit by means of gravity operation of heavy weights or by powerful springs. Fig. 13 illustrates a very powerful direct-acting clamp using serrated and hardened replaceable gripping shoes. These are released by a motor-operated winch and cable system which hoists the wedge-shaped weight, the jaws being released by a spring. The weight is held up by stalling the winch motor through a bank of resistance, or by a power-applied brake on the winch-motor shaft, as long as the bridge is in motion. As soon as the travel controller is centered or the operator steps off a platform switch, or power fails, the weight descends, wedging the roller-equipped levers apart and exerting a powerful grip on the head of the rail.

Another type illustrated in Fig. 14 is the modern development of the manual clamps shown in Fig. 12. This is released by a motor-operated winch, held released by a shunt-operated solenoid brake, and set either by a weighted lever or a powerful spring. This clamp does not depend for its holding ability on the sole action of the weight or spring. These supply only the initial force of application. As the bridge starts to move, the circular cam with its double-tapered wearing shoe immediately rocks over, and creates a tremendous locking pressure that increases as the force against the bridge is increased. The arrangement of the gripping shoes and cams is shown in the upper corner of Fig. 14. As soon as the clamp is released, centering springs return the cam to its normal position. These clamps are claimed to have a restraining ability of 40 tons for a single rail clamp or 80 tons for a double clamp. Present models of these clamps are built with telltale switch devices attached, so that the lighting of a signal lamp shows when they are in proper adjustment and corrrectly applied. Fig. 15 shows one of these double rail clamps built into the sill of a bridge. This cut also shows a sander and the sand pipes in front of the wheels.

Fig. 17 illustrates another type of self-locking clamp which creates a very effective grip on the rail by means of a cramping or jamming action. Two levers, one free to move toward and along the rail, and the other fixed longitudinally, but free to



CLAMP APPLIED

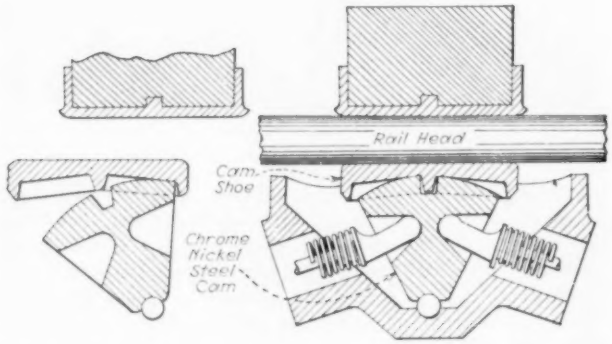


FIG. 14 SPRING-OPERATED, SELF-LOCKING, CAM-TYPE, AUTOMATIC RAIL CLAMP—NOTE TELL-TALE SWITCHES FOR INDICATING PROPER FUNCTIONING

move toward the rail, are connected by a heavy T-bolt. A weighted wedge behind the head of the T-bolt forces the levers into contact with rails, and any motion causes a cramping or jamming action which is very effective. Other forms of effective clamps are in use, but need not be described here.

A recently developed automatic locking device is shown in Fig. 16. This takes the form of an automatic power-operated wedge or wheel chock. While the bridge is in motion, this device is suspended on links clear of the rail, being released by a motor-operated system of pull rods and cranks. When power is cut off, a magnetic clutch is released allowing the wedge mechanism to drop down to the rail, under control of adjustable fluid-filled dash pots, and rests directly in front of the track wheel. A heavy weight and a system of toggles causes the wedge to be clamped solidly to the rail head. If the bridge moves a very short distance, the wheel moves up onto the point of the wedge, and under continued movement the face of the wheel bears against the curved surface of the chock, effectively blocking it. The grip arms are required to hold the device only against slipping until the wheel rides up on the wedge, after that the whole load on that particular wheel is effective in keeping the chock from slipping. The time of release is very short, and therefore delays bridge movement very little. The restraining force is developed directly against the truck, and heavy sill members are therefore not required.

All of these automatic-locking devices are intended to be set all the time the bridge is stationary, and should be released only when the bridge is to be moved. They are generally controlled by a platform or treadle switch on which the operator must stand to manipulate the traverse controllers. The instant he steps off of the platform or the power supply fails, the clamps or wedges begin to set; they should be invariable interlocked with the travel-motor contactors so that power cannot be applied to the motors until the clamps are fully released, and so that it is impossible to set the clamps without cutting power off the motors.

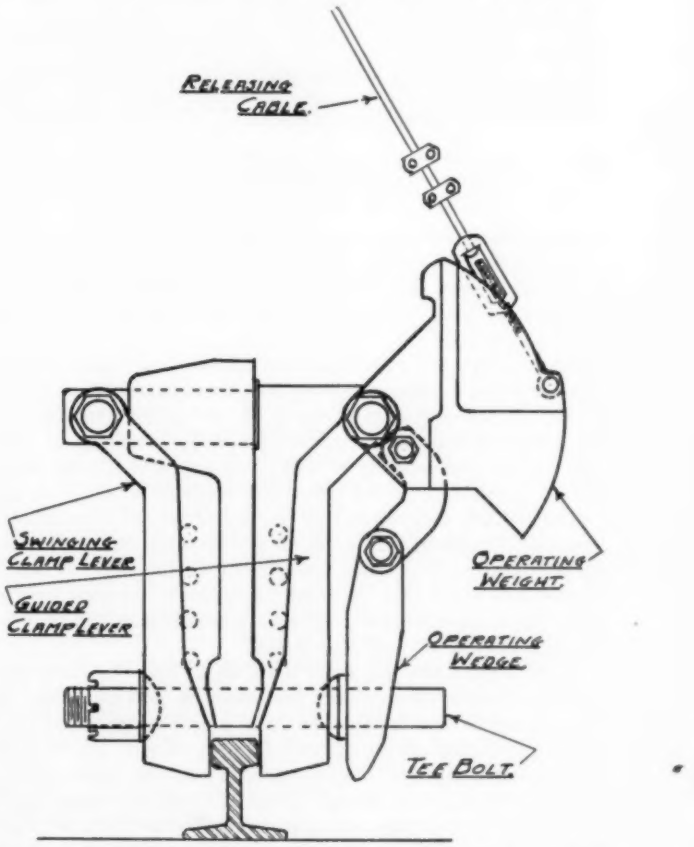


FIG. 15 JAMMING-TYPE AUTOMATIC RAIL CLAMP—INDUSTRIAL BROWN HOIST CORPORATION

Sufficient time must elapse in setting such clamps or wedges so that brakes have time to stop the bridge. For test purposes, a cut-out or short-circuiting switch is sometimes installed, so that routine tests of the holding power of the clamps can be made conveniently, by leaving the clamps set and energizing the motors. If they are in proper operating condition, they should resist the movement of the bridge under full-load current. Such periodical tests are essential in maintaining the clamps at maximum effectiveness.

When the bridge is to be idle for a considerable length of time, between shifts or overnight, or on evident approach of a storm, many dock operators supplement the automatic devices just described by other manually operated equipment. Manually applied rail clamps may be used; hardwood wedges about 18 in. or 2 ft. long may be placed under a number of the supporting wheels. Should the bridge move, the wheels will run up on these

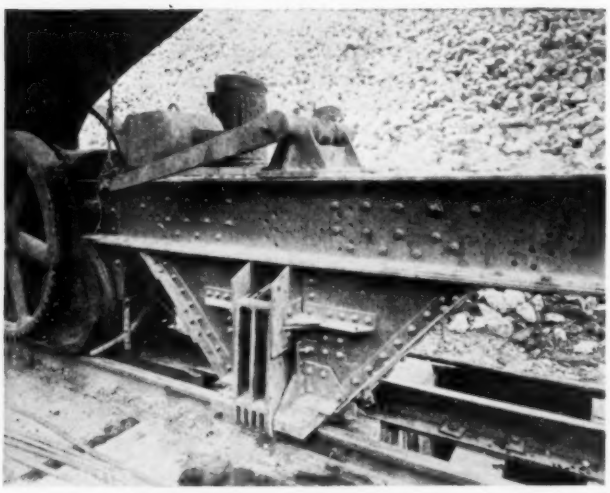


FIG. 16 GRAVITY-OPERATED, CAM-TYPE, AUTOMATIC CLAMP BUILT INTO BRIDGE SILL
(Note also track sander and sand-delivery pipes under track wheels.)

wedges, which really are very effective in blocking further movement. A very desirable arrangement is to have these wedges fitted with chains and to supply hooks on which to hang them when not in use. If loose they are apt to be lost or used for other purposes.

Struts or sprags of various sorts are used on many bridges. In some cases, these are built-up structural beams substantially pivoted at one end to the bridge truck, and pointed at the other end, with the hope and expectation that in an emergency they will dig into the concrete or timber runway and thereby block further movement. Other struts are similarly attached to the trucks, but are designed to grip the ball of the rail by manually tightened clamps or shoes. Their effectiveness depends largely on the degree of frictional attachment to the rail. If heavy enough and fitted with setscrews or hardened serrated inserts, they may be quite effective. They are usually arranged to make an angle of 30 to 40 deg. with the rail.

A much more effective strut is shown in Fig. 17. These are of much heavier construction, are seated in pockets formed in the monolithic foundation, and have a jack-screw adjustment so that they may be set up very tightly. These require more careful spotting of the bridge in order to reach the track pockets which are installed at regular intervals along the whole length of the runway, but they are much more effective than the other types of struts. When the bridge is in motion, all such struts are held up out of the way by counterweights or latches of some sort.

Another locking device takes the form of a heavy vertical structural spud or plunger, which may be raised or lowered by a hand wheel, and drops into sockets in the track foundations. While these require accurate spotting, they are very effective in preventing movement.

Some bridges are fitted with mooring cables intended both to anchor the bridge and to stabilize it against overturning forces. The cables are attached to the track by bolts and clevises or clamps, and generally run over a sheave high up on the leg and thence down to a hand-operated winch for tightening them. In some cases, the upper end is simply wrapped and tied around bridge members, and anchored in the nearest available hole, not being tightened. In most cases, the cable is not much larger than about $^{3}/_{4}$-in. plow steel, and the possible restraining force when extended at the customary angle of about 45 deg. is relatively very small when compared to the wind thrusts that often occur. Larger cables would be more effective, but too stiff and heavy to handle. The exposed condition of the cables leads to their rapid deterioration and short life. In many cases, the cables are poorly attached to the track or the bridge, so their value is extremely limited.

The physical condition of the track rails has an important bearing on safety in windstorms. This is particularly true with respect to fins and beads on the ball of the rail caused by the rolling action of the wheels. Beaded rails seriously limit the effectiveness of most forms of rail clamps, and it is very necessary

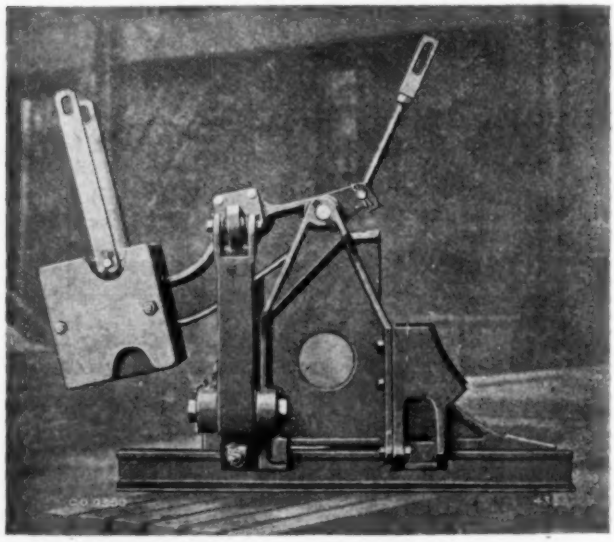


FIG. 17 AUTOMATIC-WEDGE OR WHEEL-CHOCK SAFETY STOP—WELLMAN ENGINEERING CORPORATION

that such beads be kept ground, chipped, or planed off. Some bridge trucks have been fitted with tool holders so that beads may be planed as the bridge runs along the rail. Other planing devices are intended to be dragged along the track by the bridge. Cutting tools may be shaped and adjusted to form a fairly large radius, which tends to lengthen the time before it is necessary to repeat the operation.

To insure safe anchorage of the bridge by automatic locking devices, it is necessary that the track rails themselves have good support. Most modern docks have monolithic concrete runway foundation; if on "made" ground or waterfront docks, it is necessary to support the foundation on piling. The rails should not be supported directly on the concrete, but on heavy longitudinal or transverse steel plates, or in some cases on treated ties or timbers embedded in the concrete. Rails bearing di-

rectly on the foundation tend to cut and wear into the concrete, resulting in loose rails, bad joints, and breakages. Anchor bolts must be amply strong, deeply anchored, and large enough to discount weathering and corrosion. Some dock operators make it a point to oil or paint the threads and replace the nuts at regular intervals. The rail joints must be well bolted, though the use of improved forms of splice bars may be impossible due to interference with some types of clamping devices. The top of the foundation should be sloped properly for drainage. If the rails are laid double on close centers, slots must be provided to prevent water from filling up the space between the rails and freezing. The rails and foundation must be kept clean and clear of lumps of coal, ore, stone, etc., which might cause derailment or might damage or destroy the rail clamps.

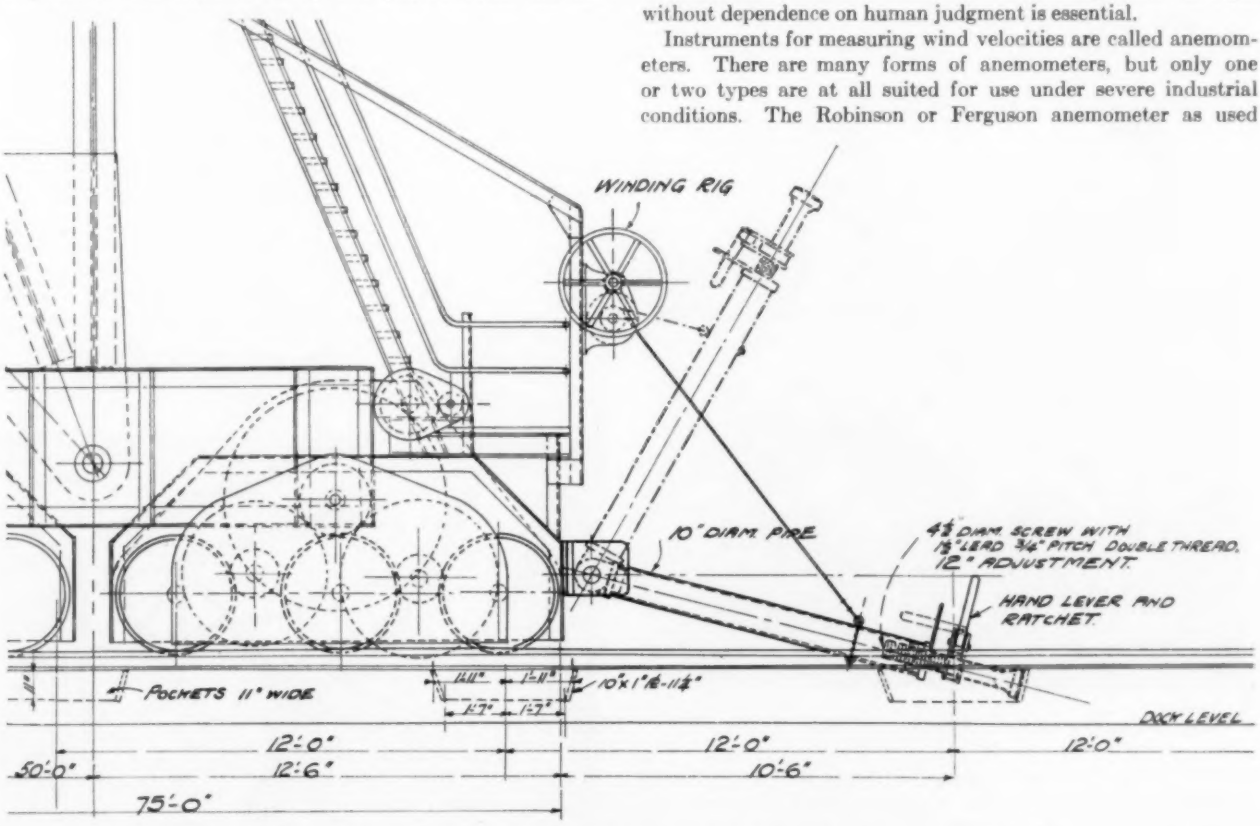


FIG. 18 HEAVY-DUTY, ADJUSTABLE SAFETY STRUT—NORTH WESTERN FUEL CO., SUPERIOR, WIS.

The ends of the track runways should be fitted with substantial and properly designed bumpers to prevent the bridge from running off the end and overturning. Bumper design preferably should be such as to prevent extremely abrupt stopping of the bridge. Gradually increasing slopes or rails bent vertically into a radius larger than the wheels and substantially supported are desirable; the lifting effect helps to counteract the overturning moment. Even a heap of earth or a timber or tie cribwork backed up by earth on top of the runway in the absence of regular bumpers may slow down and stop the bridge. Naturally, if inadequate control equipment is used, and the bridge is driven at high speed by the wind and crashes into any sort of bumpers, the result is very apt to be a serious wreck. The arrangement and design of the truck members themselves often limit the safety and effectiveness of any sort of bumpers.

The foregoing discussion has attempted to cover the factors of bridge design and equipment which have a more or less definite bearing on the ability of the bridge to withstand storms successfully. If the bridge is well designed and fitted with a full complement of adequate protective equipment, the probability of loss is remote. But how are dangerous storm conditions to be recognized? Upon what basis shall the operator or dock foreman decide to discontinue traversing movements? The vessel must be unloaded; the waiting cars must be loaded; who is to be responsible for delays, or who is to be blamed for a wreck? Of what value are elaborate systems of brakes and rail clamps, wedges, and struts, if they are not used in time? A consistent and accurate means of determining wind strength without dependence on human judgment is essential.

Instruments for measuring wind velocities are called anemometers. There are many forms of anemometers, but only one or two types are at all suited for use under severe industrial conditions. The Robinson or Ferguson anemometer as used by the Weather Bureau furnishes only average velocities, while wind-resistant structures require peak and gust protection. Pitot- or venturi-tube anemometers might be used, but they are unsuitable for long-distance operation or making permanent records. The indicating instrument is delicate and may easily become deranged. The tube openings are almost certain to clog with ice and snow during the winter months. Direct-pressure plate anemometers are simple and at first thought might be considered well adapted to bridge protection. However, they are apt to freeze up; snow and sleet loadings change their calibration. They are almost useless for distance observation, and do not make permanent records. Gust conditions set up pendulum-like oscillations, resulting in false indications. The constant oscillation likewise is destructive to contact-making devices which may be fitted for distance operation of signal devices. The chief merits of this type of wind gage are its simplicity and low cost.

For accurate and reliable industrial determination of wind strength, the revolving cup-wheel magneto-generator type of anemometer seems to be the most successful. This consists

of a three- or four-arm spider, on the spokes of which are mounted hollow hemispherical or parabolic cups. The cups all face the same direction relative to the imaginary "rim" of the wheel, and the assembly is mounted on the vertical shaft of a magneto generator, so that the cup wheel revolves in a horizontal plane. This mounting is non-directional and requires no vanes or tails to keep it headed into the wind. The generator is fitted with permanent-magnet fields, so that the generated voltage is proportional to the rotative speed. Fig. 18 illustrates the component parts of this type of generator and Fig. 19 its installation on a bridge. The armature revolves on high-grade ball bearings, the weight of the moving parts is small, and the large driving cup-wheel assembly is powerful enough to render the effects of

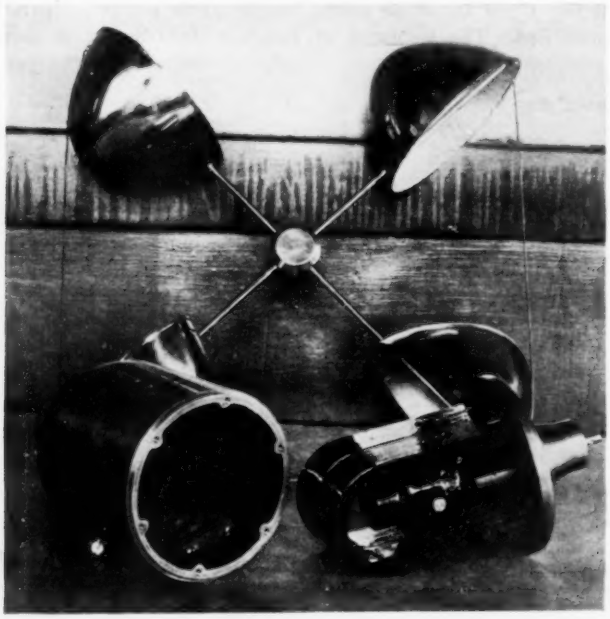


FIG. 19 CONSTRUCTIONAL DETAIL OF BURTON ANEMOMETER

bearing and brush friction negligible except at the very lowest velocities, which are of little industrial importance any way.

An anemometer of this type is especially suited for bridge protection. The generator is simple, rugged, weatherproof, and requires very little care or attention. The generator may be mounted at any suitably exposed point on the bridge, and is connected by two small wires to an indicating or recording meter, which may be located where most convenient for observation. It is equally well adapted for operating a distance of $^1/_2$ mile or more as for 50 ft. Either indicating or recording instruments may be used. It is sometimes desirable to use both, mounting the indicator directly in front of the travel controllers where it is most readily seen by the operator when he wants to move the bridge. The indicator requires little room and its readings are plain and understandable. The recording meter requires more room and other locations may be more convenient. A permanent record of wind velocity is made, showing the general trend of the storm. The fact that a permanent record is being made for the superintendent's inspection has a tendency to enforce cautious operating practices.

But suppose the operator ignores and disregards the indicating meter or recorder? How is his rash judgment to be curbed? For this purpose, a relay system has been developed, which is designed to prevent such carelessness. The relay panel carries an adjustable relay connected to the anemometer generator, and the relay in turn controls a small power contactor on the same panel. Signal devices such as lamps or sirens are controlled by this contactor, and in addition it may be connected into rail-clamp or travel-controller circuits so that continued movement is impossible in high winds. If the bridge is idle, the clamps cannot be released; if in motion, it is stopped. This means that the bridge will always be stationary and anchored the moment the wind exceeds whatever velocity has been chosen as the limit for a safe operation. If the bridge is thus stopped before the wind has a chance to reach a destructive velocity, and the bridge is promptly anchored to the runway, the possibility of runaways and wrecks are minimized. The whole secret of safe control lies in the prompt automatic enforcement of safe-operating practices. There is no opportunity or incentive for the operator to risk a hazardous movement.

Auxiliary recording pens are often fitted to the recorder and connected so as to record additional functions or details of operation. For instance, the time, duration, and direction of bridge movement may be recorded accurately, so synchronized with the velocity record that the concurrent operating condition can be determined readily. The proper functioning of the rail clamps may be made a part of the record, and any lack of adjustment or blocking of platform-switch or clamp mechanisms may be detected. Sometimes a record of wind direction is considered desirable. By using a suitable contact-making wind vane and a set of direction pens writing on the velocity chart, this record can be obtained readily.



FIG. 20 BURTON ANEMOMETER GENERATOR ON COAL BRIDGE

Sometimes dock operators object to having bridge movement forcibly discontinued while the wind strikes the bridge nearly "end on." Such end winds do not create propelling forces and are relatively safe for bridge movement. A wind vane may be arranged for selective control of the anemometer relay so that in certain "end zones" the relay-tripping velocity may be considerably increased, or automatic relay action may be eliminated entirely. In either case, full automatic control is resumed the instant the wind swings into the hazardous side directions.

Watchman service is also important in the protection of the bridges. Regular inspection should be provided at stated intervals between shifts and at night or other times when the equipment is idle, with definite instructions relative to the checking of manual and automatic safety devices. Regularity of inspections should be checked by a watchman's recording clock, with key stations at each end of each bridge. Written reports on special forms covering all protective features should be sent to the superintendent daily.

The complete bridge structure including all automatic and manual protective devices should be rigidly inspected at least twice a year by a competent inspecting engineer, preferably not connected with the regular dock-operating organization. Such inspections are a regular part of most insurance services and maintenance agreements. All valid recommendations by such inspectors should receive prompt attention.

Having considered the causes of windstorm wrecks and the devices designed to prevent such losses, in conclusion a tentative set of specifications may be framed, which if adhered to ought to afford the maximum degree of protection against storm hazards. The specifications are considered from three definite angles: (1) Safety while in motion, (2) safety while at rest, but with bridge in regular operation, and (3) safety while at rest during idle periods.

Safety While in Motion

1 The bridge builder must make ample allowance in his design for wind pressures of 40 to 50 lb. per sq. ft. of projected area.

2 The bridge must have a safe ratio of height of center of gravity and center of pressure to wheelbase.

3 The bridge should be 100 per cent traction driven or rack-and-pinion propelled; in either case with multiple motors and duplicate drive mechanisms at each leg.

4 The bridge should have smooth, powerful brakes, locking 100 per cent of the supporting wheels or all rack pinions. Graduated or retarded brake setting may be desirable, unless effective slowdown control is used.

5 Propelling motors should have inherent constant-speed characteristics, or the control system should provide such constant-speed notches. If direct-current series motors are used, one or more points of powerful armature-shunt slowdown should be provided.

6 Skew limit switches must be provided, preferably of types which afford graduated slowdown or full-automatic skew correction.

7 The operating master controller should be self centered, or have a platform or dead-man switch for controlling bridge-travel power.

8 Controllers should provide overload and no-voltage protection, and be interlocked with rail clamps or wedges so that power cannot be applied until they are released.

9 Accurate skew indicators should be plainly visible to the operator.

10 A reliable anemometer should at all times show gust and peak velocities, and also operate signal devices or cut power off in dangerous winds; as soon as power is cut off, brakes must be applied and setting of clamp devices initiated.

11 All parts of controllers and drive mechanisms should be kept in first-class condition, and runways kept tight and clean.

12 Adequate lighting for night operation should be furnished.

Safety at Rest During Normal Operation

1 The supporting wheels should be locked 100 per cent by effective brakes, or all pinions of rack-and-pinion drive systems should be locked.

2 Effective automatic clamps or track wedges must be applied at all times when the bridge is not in motion, operating on rails maintained in good condition.

3 An anemometer should prevent release of rail clamps or wedges if the wind exceeds a set danger velocity, usually 35 to 40 m.p.h.

Safety at Rest During Prolonged Idle Periods

1 In addition to automatic brakes, clamps, wedges, and struts should be furnished seated in foundation pockets or

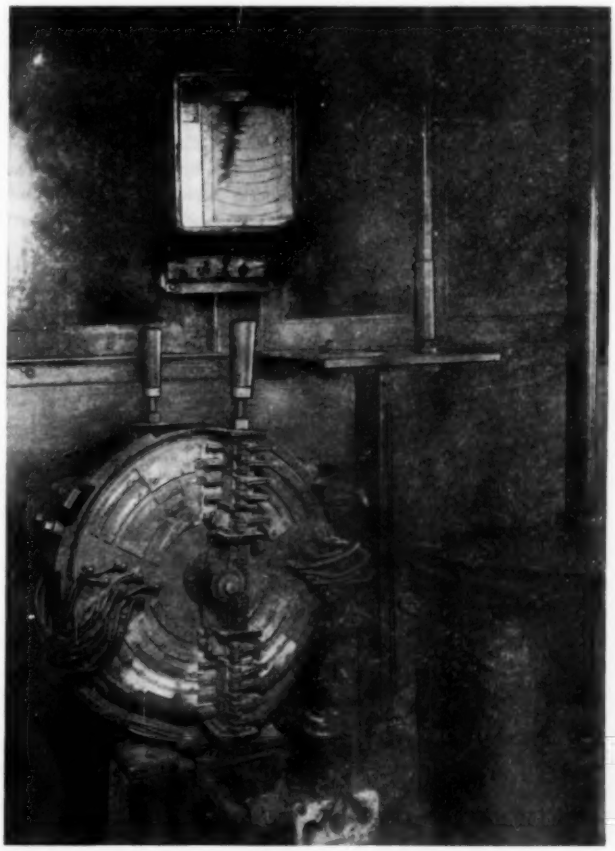


Fig. 21 Bridge-Travel Master Controllers and Anemometer Recorder

attached to ball or rail with effective clamps, or through holes in the web of rails.

2 Vertical spuds, seated in sockets in the track foundation, might be used instead of struts.

3 Manually operated rail clamps may also be used.

4 Properly shaped hardwood wedges should be placed under all wheels.

5 Watchman service should be supplied with regular clock-controlled inspection of the placing of all safety devices, with written daily reports covering conditions noted.

6 Regular inspection and tests should be made of all safety devices.

7 All parts should be kept in good mechanical and electrical condition.

Conclusion

In this discussion, the aim has not been to present new theories of protection, but rather to consider the many standard features of design and protective equipment which tend to promote

safety. If certain points have been inadvertently overlooked, it is hoped that the discussion of the paper will bring them out. Personal opinions as to relative values of various protective devices will undoubtedly vary. Operating conditions and practices may in some cases modify the conclusions set forth. While the discussion has primarily been concerned with bridge protection, many of the points taken up apply equally well to gantry cranes, unloading towers, car loaders, dumpers, etc.

For the historical references made in the first section of this paper, the author was greatly assisted by an earlier paper presented at the Minneapolis meeting of the Society in 1914 by G. H. Hutchinson, entitled "The Development of the Coal Handling Industry at the Head of the Lakes." The author also wishes gratefully to acknowledge the assistance and suggestions made by engineers of the Heyl & Patterson Company, Mead-Morrison Manufacturing Company, Wellman Engineering Corporation, General Electric Company, Electric Controller & Manufacturing Company, Cutler-Hammer Company, The Western Actuarial Bureau, and others connected with construction, controller, and dock-operating organizations.

Discussion

C. K. Steff.[2] The author has given a clear and comprehensive description of modern bridge protective devices. The paper should serve as a valuable guide in planning the protection for new equipments.

In some cases it may be possible to arrange the runway so that the most dangerous winds will strike the structure end-on. A first-class runway is worth its cost in reduced maintenance and freedom from trouble. Properly spaced creosoted ties with tie plates and supported on a heavy concrete base should be very satisfactory. The bridge builder has a definite responsibility to furnish a structure of ample stability. If wind pressures as low as 30 or 40 lb. per sq. ft. are taken, allowance should be made for windage on the leeward truss of the bridge span. New bridges should certainly have brakes acting through all wheels, utilizing the full weight of the structure, unless they are of the rack-drive type. Rail clamps have been developed to reasonable perfection only after painstaking design and field experience. This subject is not one for superficial experimenting. Rail clamps should be placed near the trucks, or preferably on a pony truck, rather than rigidly built into the center of the bridge sill where they cannot conform to irregularities in the runway beneath. Braking through all wheels is suggested as the simplest and best protective measure for new gantry cranes and unloading towers. Usually the only further protection needed is manual anchoring devices.

New equipments offer an opportunity to start right, but the problem of the majority of bridge operators is that of preserving old equipments built when modern protective devices were in stages of development and the seriousness of the wind hazard less fully realized. The problem then becomes one of estimating the value of the present protection and its shortcomings and building it up according as the value of the equipment and its importance to the plant warrants. The fundamental principles remain the same as Mr. Burton has outlined. Assuming good stability for the structure, reasonable safety during movement should be afforded by brakes acting through at least 80 per cent of the wheels and by proper electric controls. Quick anchorage is needed for important equipments, especially in windy districts, and may be obtained by a full or partial set of automatic rail clamps. Ample manual anchorage should be provided and warning given of dangerous wind velocity. Good progress is being made in building up the protection of old equipments.

Of equal importance to protective devices are the character of the operating personnel and the maintenance of the equipment. Bridges are too valuable and important to be entrusted to green operators. A bridge which is overskewed receives a very severe wrenching and is never as good again. Thorough periodic and systematic inspections by responsible plant officials are of prime importance. Bridge service is severe and parts break frequently. Employees may be slow to report trouble, especially if they or fellow employees are in part responsible for the condition. Some types of automatic rail clamps require frequent checking by the power test to insure they will be in proper adjustment to hold when needed. Outside inspection bureaus may do their best to point out unsatisfactory conditions, but at best their inspections are made some little time apart and are not a substitute for plant inspections.

W. C. Raube.[3] Although the writer is interested in all phases of material-handling bridges, he is of course most interested in the electrical equipment used to control various operations on a bridge. The author pointed out that the unstable speed characteristic of a series motor can be corrected by the use of a so-called shunted armature connection. Possibly the terms "light" and "heavy" armature shunts as used in Fig. 10 might be explained so that any one not thoroughly familiar with speed-torque characteristics of series motors would clearly understand the difference between a light armature shunt and a heavy armature shunt. Under Fig. 11 (*b*), the author shows the combination of resistances used for a shunted armature connection on a series motor. It will be noted that there is a series resistance, as shown at extreme right of Fig. 11 (B), and a resistance in parallel with the armature. For a heavy armature shunt, the resistance in parallel with the armature must have a low resistance and relatively high current-carrying capacity, while for a light armature shunt the resistance in parallel with the armature must be higher and the current capacity can be lower. Representative speed-torque characteristics for light and heavy armature shunts are shown in Fig. 10, curves (*b*) and (*c*). The high resistance in parallel with armature (light armature shunt) gives a much steeper curve than the low parallel resistance (heavy armature shunt). It will be noted, therefore, that increasing the parallel resistance increases the slope of the resulting speed-torque curve, and if this resistance is made infinitely large such as would be obtained in the path of air which we might consider connecting the armature terminals in a standard series motor, the speed-torque curve becomes very steep, and we actually obtain the more familiar series-motor characteristic curve obtained with resistance in series with the armature. The curve corresponding to this latter condition is not shown on Fig. 10, but can be visualized by sketching in a curve on Fig. 10 which at zero speed would start near the point where curve (*b*) intersects the zero speed ordinate and would then curve up so as to come very close to the upper end of curve (*a*) at 250 per cent speed. Such curve, it will be noted, is now all to the right of the vertical zero torque line.

The writer has always found it very helpful to have a good idea of the general shape of speed-torque characteristics for varying degrees of shunted armature connections and finds that an easy way to keep straight on this point is to start with the curve for a standard series motor with resistance connected in series with the armature (and which automatically has infinite resistance in parallel with the armature in the form of the air path between armature terminals), and then remembering that the speed-torque curve can be flattened out by simply reducing the resistance in parallel with the armature. The lower the parallel re-

[2] Engineer, Inspection Department, Associated Factory Mutual Fire Insurance Companies, Boston, Mass.

[3] General Electric Company, Schenectady, N. Y.

sistance, the flatter the curve becomes. Moreover, with such lower parallel resistance, the speed-torque curves will reach farther over into the braking quadrant to the left of the zero torque line.

The author made the statement at one point that the slowdown action of an armature shunt on a shunt-type motor is not quite as effective as in the case of the series motor. This statement may be misinterpreted. As the author indicates, in the case of the shunt motor, no change is made in the field strength of such motor when using a shunted armature connection. In the case of the series motor, the presence of the series resistance in the shunted armature connection limits the current taken from the line and therefore also limits the current that can pass through the series field. Maximum current will flow through the series field with the motor at standstill, but as the speed of the motor increases and the c.e.m.f. builds up, less current is taken from the line, and therefore the series field current is reduced. This process continues at higher speeds where the speed-torque curve crosses the zero torque line and passes on into the braking quadrant. This condition of field weakening of course allows the motor to run faster, and the speed-torque curve starts out at a fairly uniform slope, but breaks upward in the braking quadrant. This is particularly noticeable in the case of the light armature shunt curve, Fig. 10 (*b*). In contrast with this, we find the speed-torque curve for shunted armature connection on a shunt motor starting at zero speed and rising at a fairly uniform slope even into the braking quadrant. This, then, means that, assuming both the series and shunt motors operating at some definite speed and then applying the shunted armature connections, it will be necessary to go farther to the left in the braking quadrant before intersecting the resulting speed-torque curve for the shunt motor than for the series motor. This results in a higher dynamic braking effort in the case of the shunt motor than for the series motor. At first thought it might seem, therefore, that the shunt motor has an advantage over the series motor in this regard, and it actually has as far as getting the higher dynamic braking torque; but in order to decelerate smoothly with the shunt motor, it is necessary to put in a large number of shunted armature points, whereas with the series motor advantage can be taken of the increasing slope of the shunted armature speed-torque curves and arrange the series and parallel resistances so that the dynamic braking torque would be limited to a reasonable value, thus giving smooth deceleration with fewer steps of armature shunt connections.

D. C. Wright.[4] The author has covered safety devices for protection against wind so thoroughly that little can be added. However, the writer would like to bring up some points as regards safety of operation.

In the past it was customary to mount the master controllers for the control of the bridge-propelling motion at some point along the trolley or track runway. The operator would stop his man-trolley opposite the master controllers, walk out of his cab, and operate the masters to move the bridge to the desired point, and then would return to his cab to continue the operations of hoisting and trolleying. This mounting of the masters required no collector bars along the runway for the operation of the bridge, and made it possible to run as many wires as required from the masters to the magnetic controllers, to secure smooth acceleration from the slow creeping speed to full speed. In later years, however, the trend has been to mount the bridge master switches in the man-trolley, thus allowing the operator to move the bridge with the man-trolley in any position. This arrangement saves time and reduces the number of trolley movements considerably.

[4] Chief Engineer, Electric Controller and Manufacturing Company, Cleveland, Ohio.

There is one objection to this arrangement, in that it is too expensive to run 14 to 16 collector bars the whole length of the runway, and therefore full speed control from the master switches to the magnetic bridge controllers has to be dispensed with. Acceleration from the creeping speed to full speed has to be automatic. The rate of acceleration should not be too fast, or the wheels are likely to slip if the rails are wet or greasy. Also, too rapid acceleration will cause whipping action in the tall bridge structure and subject it to severe strains. A controller allowing only a second or less to accelerate, such as is used for mill drives, is not suitable for heavy bridge drives. Comparatively long time between closure of successive contactors is necessary to give smooth acceleration, or else a large number of acceleration contactors should be used. The bridge controllers should be selected with this point in view.

A patented control system, without extra special relays, has been devised, which allows one wire or collector bar to be used for more than one purpose, and gives proper automatic control of the two rail clamps and the two sets of bridge motors, individually or together, with but seven collector bars along the runway. Three of these are main power bars to supply power to the rail clamps and bridge motors. The other four, two for the pier end and two for the shear end, provide control for the bridge motors, giving forward and reverse motions, a slow creeping speed, and automatic smooth acceleration to full speed.

Another feature entering into safety of operation is the control of the man-trolley. Present practice is to operate these at speeds of 1200 ft. per min. or more. Weights may run up to 75 or 90 tons. Bearings are of frictionless type. Under these conditions it becomes a greater problem to stop the trolley than to start it. If it fails to start, loss of production occurs; but if it fails to stop there might be considerable damage done or loss of life might occur, besides loss of production. The more common methods used for stopping are to plug the motor or use dynamic braking. Both these methods require continuity of power supply. If the power supply fails at the critical moment, then the emergency means of stopping, by applying air brakes or magnetic brakes, has to be resorted to. These may not be in proper working order or the operator may be excited or inexperienced, and sad results may occur. The most frequent cause of power interruption is poor shoe design or rough collector bars, causing an open circuit sufficiently long to drop out the no-voltage relays during the drifting portion of travel when no power is being taken. The writer would therefore strongly recommend that all important collector bars be equipped with two collector shoes of good design, mounted, not just a few inches apart so that a particular rough spot on the bar might momentarily throw off both shoes, but several feet apart, so that at least one shoe will always be in circuit.

The rivets in bridge structure are subject to severe strain and frequently lose their heads or become loose. The writer would like an expression of opinion from bridge builders as to how frequently and thoroughly the riveting should be inspected.

The writer would like to ask the author a question as regards adjustment of the anemometer relay. Assuming that it is adjusted to lift and stop operation of the bridge at a wind of 35 m.p.h., then at what wind speed will it drop and allow resumption of bridge operation?

Frequently, with wind speeds of less than 35 m.p.h., the bridge may travel at a dangerous speed, particularly if the motor fields are inadvertently weakened or if the bridge is overmotored. It seems to me that an overspeed switch is a good safety device. This could be connected in the same circuit as the anemometer to stop the bridge and set the clamps in any case of overspeeding.

In addition to making the bridge master spring return to the off position, the connections should be such that, after the bridge

has been stopped by the anemometer or by the overspeed device, the operator should have to return the master to the off position before the bridge could be started again.

E. M. HEINSELMAN.[5] The first reading of the paper disclosed that the various items pertaining to windstorm hazards was very well covered; and even after careful study, there is but little that can be added to the discussion in so far as the installation and use of mechanical appliances are concerned. The paper is well worth the careful study of any one who is interested directly or indirectly in the construction or operation of a movable bridge.

However, it is felt that something would be lacking from this discussion if nothing were said about the importance of training the operating forces so that they will act promptly and efficiently in an emergency.

Even when a bridge is equipped with the most practical and up-to-date mechanical devices for preventing bridge losses, one still must consider the human element of first importance. Successful operation of these structures requires trained men, and the writer believes that it is a serious mistake to take the attitude that full dependence can be placed on the automatic devices for protection against windstorm damage. The operator of a bridge or a tower must be something more than a mere "lever puller," and when an emergency arises, the ground crew on a dock is just as important as the ground crew at an airport.

Without going into too much detail the writer also would like to mention a few points pertaining to bridge design, construction, and operation which he feels are of some importance. From the standpoint of windstorm protection, the driving mechanism of the bridge is important. The types of drive in common use are the lineshaft worm-gear drive, motor-driven direct worm gear, lineshaft bevel and straight gear, motor-driven direct bevel or straight gear, the cable haul with drum, and the pinion rack. In addition there is a type of drive with four individual motors, one on each corner of the bridge trucks. Each back-geared motor drives through a flexible shaft, which drives an intermediate pinion, which in turn drives a gear, which in turn through beveled gears drives a line of flexible shafting having beveled pinions mounted thereon. There are three beveled pinions on this shaft. Each pinion drives a vertical shaft through gears which in turn drive each pair of wheels independently. The writer considers this a very safe and efficient drive.

It requires a large, substantial, and expensive foundation for the pinion-rack drive bridge, but it is believed to be as safe, if not safer, in a windstorm than that mentioned. The traversing motors and brakes on this type of bridge are at the drive pinion.

There is no denying the fact that there is a great safety factor in the worm-gear drive, but it is slow moving, and for this reason it is not very popular in this swiftly moving age, and few, if any, are being built now.

An essential aim in windstorm protection is to prevent the truck wheel from beginning to turn, and for this reason the writer would place the motor-driven direct bridge ahead of either the lineshaft drive or the cable and drum haul.

On a cable-haul bridge the cable is anchored at each end of the bridge-rail foundation, and about six wraps of the bite is reaved around the drum. The cables would be anywhere from 1000 to 3000 ft. long, depending on the length of the dock. For an illustration, take a bridge standing north and south, and which moves east and west. If the bridge is moved in a westerly direction, the cable will become tight from the west end, or the end from which it is being pulled, and if there is any slack in the entire length of cable, it will be at the east side of the bridge. If a wind comes up from the east after the last move of the bridge, and there is slack in the cable from that side, then there would be some chance for the wheels to turn slightly before the moving mechanism becomes effective as a brake, and that little start may be sufficient to lose a bridge.

The writer considers the lineshaft somewhat similar to the cable haul in the aforementioned respect as the moving motor and brake on a lineshaft bridge is on top of the bridge at one end of the span. It may be a span of 300 or 400 ft. and very high, in which case there are a considerable number of couplings on the shafting between the motor and brake and the drive wheels. If the moving controller is opened slowly and the action of the lineshaft is watched, it will be noticed that the shaft will turn over from one to two turns before the truck wheel starts, and if there is any wear or lost motion in the pins and universal couplings, the lost motion in the shaft will be still more noticeable. Therefore, if a wind should start such a bridge into motion, the truck wheel is free to turn the amount of lost motion there is in the length of the lineshaft before the brake at the moving motor becomes effective.

On the motor-driven direct bridge the brake and motors are right at the truck wheels where there can be no lost motion except what little wear might come in the gearing, and that could not be more than in the other types mentioned.

Making allowance for these differences in safety, due to the various types of drive in use, it is believed that if the structure of any of these types is kept in first-class condition, and is equipped with suitable safety devices, properly maintained and applied, and is in the hands of a careful and efficient crew, it will be reasonably safe in a severe storm.

Track-sanding devices are of some value in bridge operation. However, the writer strongly recommends the electrical type and does not favor the air type even if electrically heated—due to the amount of moisture that will accumulate. The main value of sanders is for normal operations, in starting or stopping on wet or icy rails. Their value in a bad wind is doubtful, as the sand will not stay on the rails, or it may even blow away before it reaches the rail.

In addition to the bridge controls and skew-limit cut-out switches it is believed to be absolutely necessary to have a good skew indicator on each bridge. The operator, by all means, should be trained to keep his span straight by the indicator, and must not depend on any electrical or mechanical device to keep him out of trouble. In fact, he might well be told that these devices are for emergency protection only, and for this purpose they are extremely valuable.

The author has given a very good description of automatic clamps, but words about testing them would not be out of place. Apparently he suggests that all the clamps on one end of a bridge should be tested together. That is likely to be a rather severe test on the span itself. The writer prefers to test the clamps singly, and one can then cut down the power to about half. That is, if it takes two points of power to move one end of the bridge, give it one extra point. When testing two clamps together, one can never be sure that both clamps are holding. If each clamp holds, one has a good idea of what it would do in a wind. For instance, with a motor-driven direct bridge, with two moving motors on each end ranging between 35 and 45 hp. each, a three-point test on each clamp with that power would probably equal the force of a wind of 50 to 55 m.p.h. Therefore if each clamp will hold with that much power, there is reason to believe that the bridge is well secured—if it has four clamps (two on each end).

When testing clamps, if one tests the last clamp on the bridge and walks away from it, he is leaving a severe strain on that clamp. Therefore when the last clamp on the bridge is tested, all the clamps should be raised and allowed to set themselves.

[5] Manager, Tornado Inter-Insurance Exchange Bridge Operators Engineering and Inspection Bureau, Duluth, Minn.

If a bridge stands in one spot for several hours, there is some chance of the clamps loosening up from the weaving and vibration of the structure. Therefore it is a good idea for the operator to raise his clamps and reset them at quitting time, especially if the bridge has been standing in one spot for a long time.

Weather conditions affect the operation of clamps to some extent. Snow will sometimes pack into the clamps in such a way as to make them ineffective. The writer therefore recommends shoveling out the bridge tracks, keeping them free from ice, snow, coal, or other obstacles. Even under normal winter conditions, the clamps will not work as fast in cold weather as during the summer. A special test to time these clamps was made during the month of November and they were found to set in from 5 to 15 seconds. It is the writer's understanding that one bridge manufacturer thinks the clamps should not set in less than 5 seconds, and if they set in less than that they are setting too fast and will jar the structure too much.

Automatic clamps are usually installed under the truck sills, near each set of wheels, so that the clamps when set will grip the same set of rails on which the wheels rest. However, in some cases the clamps are set at the ends of the truck sill. When this is done, it may happen that the clamps will grip a different rail than the one on which the truck wheels are resting. If a heavy wind came up under such conditions, the clamps would afford little protection, as there would be no added weight on the rails they were gripping.

Placing the clamps on the outside ends of the truck sills may also interfere with operations. For instance, with two or more bridges on one dock (working on a boat), with clamps taking up 2 ft. or more of added space on each end of the truck sill, it means that the rigs would necessarily have to be 4 ft. or more further apart than without the clamps. This would often be a big handicap in working the bridges.

It is common practice to call gripping devices of this kind automatic clamps. The maintenance of these devices is very important. If neglected, they may still work automatically in an emergency, but prove entirely ineffective. It should be the aim to keep them in such condition that they will be not only automatic when needed, but also effective.

The bureau commenced inspection service during the fall of 1920 season. In looking through the engineer's reports the writer finds this notation: "When making my first trip over these properties, I found only a limited number of clamps. When the tests were made, about half of them slipped. Some of them were not in working order at all."

During the intervening years, a great many additional clamps have been installed, the management has taken a greater interest in the maintenance of this equipment, and the men in charge of the work on the docks have realized that expensive devices of this kind are useless if not properly kept up; therefore the record for the last few years shows a tremendous improvement in the effectiveness of the clamps when tested. The dock organizations now seem to take the initiative in so far as the upkeep of their equipment is concerned and do not wait to be asked to do it. In 1929 our clamp tests show that only 5.2 per cent slipped at the time of the inspection, and many of these were merely out of adjustment and were put in first class condition at the time. To make some of the others hold it was necessary to install new shoes, inserts, or liners.

The increasing efficiency in the gripping power of automatic clamps is due to several reasons. First of all, the various bridge-construction companies are keenly interested in this feature, and all of them have made improvements in the kind of clamps they furnish. Second, as already stated, the management and employees now realize that it is very important to maintain the clamps in first-class shape.

In the early days of our work it sometimes happened that we would find all the clamps on a bridge ineffective at the time of inspection. This has not occurred for several years. Of course, it is possible for the clamps on one end to slide because of a beaded rail, uneven foundation, etc., but the other end should hold.

The author has mentioned the importance of keeping the rails and foundations in good condition.

There are still some steam rigs in service, and there are still a few places where it is necessary to build steam rigs, due to the fact that electric power is not available or there is a limit to the power that can be furnished.

Many of the steam bridges are equipped with automatic clamps, but on such installations a power test of the automatic clamps is almost impossible, and we do not believe should be attempted.

On these rigs there is no correct way of measuring the amount of power, and the operator is liable to give a clamp much more than it should have. Occasionally a test can be made by pulling a locomotive or similar machine against the clamps. When pulling against the bridge in this way, it may be possible to measure the pulling strain, but in no case do we consider it advisable to apply the steam against the clamps.

On steam installations about all an inspector can do is to examine the clamps carefully when he makes his rounds, and check up the imprints on the rails from the gripping surfaces, and inspect all wearing parts of the clamping mechanism.

There is still another type of bridge which has not been mentioned; that is the pivot bridge. This is pivoted at one end and circles the yard. The value of clamping down such a bridge in case of an approaching storm is doubtful. Some take the position that it should not be tied down, for if a wind came up, the pivot bridge would swing in a wide circle to the point where it would settle down with an end wind behind it.

The writer would also like to say a little more about the wooden wedges mentioned by the author. His first choice for material is basswood. However, basswood planking is rather scarce and cannot always be readily obtained. The second choice is maple, and this can be obtained in any part of the country.

In shaping these wedges, care should be taken so that the contour will allow the truck wheel to ride the point first before it engages the main protion of the block; otherwise the wheel will have a tendency to push the block ahead of the wheel.

Some operators think it is a good idea to paint the wedges. If they are painted a distinctive color they are easy to find when there is an emergency and they must be put in quickly. They are also easy to find when it comes time to take them out, and there is less chance of leaving one block in under the wheel when starting up the bridge.

As to the effectiveness of wedges, with sound wedges of proper shape it has been found impossible to start up a 50 per cent drive bridge with four wedges under the wheels, or with eight under a 100 per cent drive bridge, but of course one can run over them if the bridge is under way.

There is some difference of opinion as to the relative value of wood and steel wedges. The writer prefers the wooden wedge. It is easier to insert, it costs less, and there is less chance of damage to the flange of a wheel when bucking against a wooden wedge.

There is but little that can be added to the outline given by the author in dealing with rails and foundations. However, the writer has in mind several dock properties that are served with railroad tracks that cut across the bridge rails. This is a big handicap in the maintenance and use of both manual and automatic clamps. These crossovers also tend to break the flanges of the bridge wheels. In planning a dock, this condition should be avoided if at all possible.

During the past season it was found on a few properties that when renewing rails they put them in "staggered fashion" on the two-rail runways. The disadvantage is that there are twice as many places for the clamps to strike against the fishplates or the bolts. However, when clamps are found to be striking, they can usually be elevated with little difficulty, and still engage the rail in proper fashion. The advantage of staggering the rails (and it is a big one in the writer's opinion) is that it minimizes the pounding on the foundation at the rail joints. Where there is a double rail joint and the bridge hits the ends of the rails, there is a big weight at the ends of the rails, and eventually it pounds a hole in the concrete. If the rails are staggered, there is also some advantage when applying the clamps (either automatic or manual) as they are grabbing different sets of rails.

When there is a wind of 35 m.p.h. or more, it is dangerous to move the bridge, even if it is an end wind, as the wind can veer without an instant's warning. If the bridge is in motion when the wind turns and the clamps set automatically, that in itself may create a dangerous condition; besides, if the bridge is moving at a rapid rate when the clamps grab, it is doubtful if they would halt it.

In addition to the foregoing comments, there are a few odds and ends that may be mentioned to good advantage. For instance, it may not be out of place to say something about the type of bridge which is constructed with the skewage at the bottom, rather than at the top. In such structures it does not give a very wide spread at the bottom. Therefore, it is believed that such a bridge is more hazardous in a storm. The writer knows of two cases where such bridges have gone down, because the bottom skewed right off the skew rollers. In one case this happened when the bridge came into contact with another bridge.

One superintendent recently suggested that it might be a good thing to provide a trough of sand over the rails near the bumpers.

There is no hard and fast rule for estimating the operating life of a bridge, but there seems to be an opinion that with a good structure, operated under normal conditions, the owner should get at least 20 years of satisfactory service from it.

It is of course understood that the parts that are subject to wear should be replaced from time to time as needed. However, the writer knows of one bridge in ore and coal service that is approaching 30 years of use. This bridge is in very good condition today. The parts exposed to wear have been renewed when necessary, and rust-eaten members have been replaced. On the other hand, the writer knows of a bridge in ore service exclusively where the structure and driving mechanism were so bad after 18 years of service that the bridge was condemned and taken down.

In the earlier days of bridge building, it is believed, the percentage of overstress allowed was considerably less than today, and increasing the percentage should tend toward longer life for the newer structures.

Mention of the life of a bridge may seem like straying away from the subject under discussion, but the main reason for bringing it up is because the bureau's records indicate that two of the losses occurring during the last ten years were due to weakening of the structure by corrosion, battering of sills, and members by the bucket, etc., and this factor is of increasing importance as the structures grow older.

A study of the wrecks which have occurred during the past ten years shows that there were comparatively few losses due to cyclones or tornados. The majority of the bridges were lost in strong gales. Some were lost because the bridges were moving in dangerous wind velocities, when they should have been tied down. Others were lost because the safety equipment had been neglected. For instance, in one case the solenoid brakes were entirely ineffective. In another case the bridge was lost in a 48-mile wind, because the clamps were not kept in adjustment.

Others were lost because of certain peculiarities in construction to conform to the type of service required; for instance, light bridges on top of lime and salt storage warehouses.

Probably some bridges were lost because there was no effort on the part of the operating force to tie up in the face of an approaching storm, or because those efforts were delayed too long.

Due to the fact that it is very expensive to cut out a furnace for want of raw material, there has at times been a tendency to take a chance on the safety of a bridge in ore service, when wind conditions were dangerous. When engineers are planning an ore dock, they should give full consideration to the value of furnishing sufficient ore bins so that the operator could have a supply on hand, available for the furnace, during a storm, and the bridge could then be tied up without interfering with the blast-furnace operations.

It may not be out of place to end this discussion with a few words on operating practice. Even after one has installed all possible safety devices, he is still dependent upon the human element for operation of the bridge, and to some extent for the application of the safety devices.

It is the bureau's practice to invite all superintendents of the various properties to a two-day meeting, held each spring, at which time all problems in connection with the operation of bridges and towers are discussed in detail. At these meetings our operating rules are revised. We also issue windstorm bulletins annually, and these bulletins and the rules are posted in each dock office and in each bridge cab, where they are in full view of the operator.

In addition to the visits made by the inspecting engineers, a committee of superintendents goes over each property once a year.

At various times during the year unusual conditions are taken up in circular letters and bulletins are sent to the interested parties, and four times each year we issue a small paper entitled "Loss Prevention," which deals with subjects pertaining to the operation of bridges and towers.

These things are done to acquaint all the interested people in all conditions which may affect the operation of their properties and to enable them to take steps to prevent damage to their equipment.

Just what an operator can do in an emergency depends on the man and also on the conditions under which he is working. If he is moving down the track with a wind and sees that the wind is getting the best of the structure, he may signal for help. If there are men at both ends of the bridge, they can put the blocks or wedges into place. If the clamps fail him, he has an opportunity to reverse the power.

Sometimes when unloading boats, if a high wind comes up, and it is necessary to move, it may be far safer to move the boat rather than the bridge. However, if an operator is working a certain section of the dock and this gets filled up, then he will have to move the rig or shut down.

From the standpoint of preventing windstorm losses, the bridge operator is an important man. Employers realize this and take particular pains in securing good men for this position. They also make certain that these men are properly trained before the bridge is left in their hands. Some employers make it a practice to hire oilers with the idea that they will eventually qualify as operators. When an intelligent man is employed as an oiler, he has an excellent opportunity to learn the duties of an operator within a comparatively short time.

When the first bridge is installed on a new property, the training of the operator is often left to a special man employed by the bridge builder. When this is done, it is advisable to get as many men as possible familiar with the operator's job, as, if the regular

operafor quits, is discharged, or is disabled, the employer might have some difficulty in securing a competent man on short notice.

We were recently asked the question as to the proper place to tie up a bridge. If you are going to tie up a bridge for a week or ten days, put it up against the bumper at the far end from the direction of the prevailing winds. We know of cases where the owner has constructed the dock foundation with a slight grade against the prevailing wind, making the bridge move harder in that direction than it would in the other.

In an emergency, the operator will of course tie up the bridge wherever he may be. If the boom is up, he will leave it up, as this is probably the safest place for it. For instance there may be a boat in the slip, and if the bridge goes down, with the boom lowered, it is going to strike the boat.

In spite of the many devices that are designed to guard against or prevent windstorm losses to these expensive structures, the writer wishes to again emphasize the importance of the human equation. There is a safety axiom to the effect that the careful workman is the best safety device known. That is almost equally true in the prevention of bridge losses.

With a structure in the hands of a careful operator who has been thoroughly coached as to his responsibilities and with a ground crew that is ready to assist in anchoring the bridge when there is a possibility of a destructive wind, one can be sure that the protective devices will be used to good advantage.

No matter how good the mechanical safeguards may be, if they are not kept up at all times and used in an emergency, they are of small value. One cannot stress too strongly the importance of bringing home to the men their individual responsibility for the safety of the bridges under their care.

G. H. Hutchinson.[6] The author has rendered a distinct service, not only in clearly setting forth the importance of adequately safeguarding movable material-handling bridges and similar structures, but also in bringing together such a full statement of present-day practice in this important field.

It is evident that the better we understand wind and its action in general, and the fuller knowledge we have in regard to local wind conditions to be dealt with in individual installation, the more effectively provision can be made to avoid wind damage. In safeguarding structures of the movable bridge type against wind each installation requires that consideration be given to the magnitude and general proportions of the structure, to the features of exposure and accessibility, to the requirements and limitations of operation, to the direction, intensity, and general character of the winds prevalent or likely to occur in the district or at the site, to the probable effect on the structure of the wind found likely to occur and finally to the provisions for making the structure safe under conditions found to obtain.

Fluctuating Winds

There appears to be no fixed relation between velocities of momentary peaks or gusts and average wind velocities, even for comparatively short periods, the ratio fluctuating widely with the character of the wind, which in turn is influenced by geographical location, topography, seasonal conditions, and by the the direction of the wind, especially when near large bodies of water. A casual examination of the accompanying anemometer charts, Figs. 22 to 29, moving at the moderate speed 3 in. per hour, would appear to indicate that in high winds at the head of the Great Lakes where the charts were made in exposed location at elevation of about 140 ft. above lake level, momentary peak velocities ranging from 50 to 100 per cent above the average velocity for 5-min. periods are not unusual. While the isolated maximum momentary peak alone may not be the critical wind, this, together with the accompanying repeated lower peaks or gusts of longer duration, may at times be the determining factor in causing damage. This makes it essential in selecting the governing velocities in safeguarding movable bridges that the peak velocities be given due weight in relation to the average velocities reported by the U. S. Weather Bureau as "maximum" and "extreme" velocities for 5-min. period and for 1 mile, respectively.

In commenting on results of the Mt. Washington experiments which he made for the U. S. Weather Bureau, (1)[7] Professor Marvin states in regard to peak velocities and the possible cumulative effect of recurring peaks, "It is important to note that momentary pressures as much as 35 per cent above the mean pressure (for a 5-min. period) may continually occur and recur. If their rate of occurrence be at all synchronous with a natural time of vibration of the structure of any part thereof, remarkable effects may follow."

Value of Experiments

The experimental work of Langley (4), of the U. S. Weather Bureau (1), and of Eiffel (5), together with that of numerous earlier experimenters, largely with winds of low or moderate velocity, has been of great value in establishing formulas for "corrected" wind velocity and for velocity-pressure ratio and in determining tentative values of pressure coefficients obtaining under various conditions as to size, inclination, shape and aspect, and interrelation of surfaces subject to wind pressure

The aforementioned experiments were confined largely to low-velocity winds acting on thin, flat plates, and were thus limited in respect to two important features. For ultimate data applicable to skeleton steel construction, experiments with high-velocity winds acting on structural members of various shapes are needed. The modern large high-velocity wind tunnels which have been devoted thus far principally to the development of aeronautics would appear to be adaptable to experiments on composite structural members or even on models of entire structures of skeleton type.

Several years ago the U. S. Bureau of Standards conducted a series of tests on a model of a tall, square building with wind at "true" velocity of 76 m.p.h. However, owing to the shape of the model, the results are not applicable to skeleton construction. The report (9) on these experiments states, "Loads produced depend on the form of structure. . . . We know now that measurements on a flat plate do not give results applicable to the side of a building. There is no definite pressure corresponding to a given wind speed applicable to all types of objects; many experiments are now available which enable a general survey of the effect of form variation, but many more measurements are needed on models resembling more closely actual structures."

However, the report further states, "We believe that we must know what takes place in a uniform wind under controlled conditions before we can hope to find out what happens in a gusty wind. . . . When this preliminary information has been obtained, work may be begun on a structure of simple type on exposed location in natural wind."

The investigation (10) of wind pressure on several spans of a railway bridge, for which preparation was being made in 1929 by the National Physical Laboratory in continuation of the investigation at Eddington initiated by Dr., now Sir Thomas Stanton, in 1902, should throw considerable light on the characteristics of wind pressure on full-sized skeleton steel structures in exposed location. The scope of these experiments is indicated by the fact that it is intended to record the maximum and the mean pressure occurring simultaneously on the two halves of a half mile section of a rail-

[6] Engineer, Davison Coke and Iron Company, Pittsburgh, Pa. Mem. A.S.M.E.

[7] Numbers in parentheses refer to Bibliography at end of the discussion.

way bridge composed of 22 spans ranging from 134 feet to 327 feet in length.

Experiments of the nature outlined by the Bureau of Standards are greatly needed, but until data specifically applicable to each of various types of construction are made available, the results obtained by the earlier experimenters, which in considerable measure still form the basis of our present knowledge of wind action, will continue to be of great value in engineering calculations. With this in mind, the writer presents the following data taken from matter gathered from various sources for personal use, believing that this information has not been previously presented in convenient form for reference.

Inclined Wind

Duchemin's (2, 3) formula,[8]

$$P_N = (P_H^2 + P_V^2)^{1/2} = P \frac{2 \sin \theta}{1 + \sin^2 \theta}$$

for normal component of pressure from inclined wind acting on plane surfaces gives normal pressures much greater than would be derived from resolution of forces. By this formula, the normal pressure of a wind, the axis of which is inclined at an angle of

25 deg. to the surface is about 72%
65 deg. to the surface is about 99.5%
45 deg. to the surface is about 94%

of the pressure on a plane placed at right-angles to the direction of the wind.

The results of Langley's (4) experiments on wind at velocities ranging from 11 to 67 m.p.h. correspond closely with those deduced from Duchemin's formula.

The foregoing indicates the need of giving careful consideration to the effect of the large resultant pressure due to horizontal winds making critical angles with the axes of structures of the type in question. In the absence of experiments with inclined winds on the types of members involved, it would appear advisable to make provision to withstand the resultant of normal pressures from the large axial and lateral components on the exposed transverse and longitudinal surfaces, respectively. According to the formula mentioned, it will be noted that wind inclined at an angle of 45 deg. to the bridge axis would simultaneously produce axial and lateral pressures equal to about 94 per cent of the corresponding pressures which would result from axial and lateral winds acting separately at different times. However, it should be borne in mind that the area of transverse and longitudinal surfaces subject to normal components of pressures from a diagonal wind will be somewhat different from the area of corresponding surfaces exposed to the aforementioned axial and lateral winds acting separately, the difference depending upon the type of construction and the arrangement and sections of members.

It may be of general interest to note that Eiffel (5) in commenting on his experiments with low-velocity winds says: "The most unexpected result was the sudden increase in the resistance of a square plate inclined 37 deg. to the wind to a value 50 per cent greater than the resistance of the same plate normal to the wind." Eiffel with other experimenters recognized that a considerable percentage of the high normal pressure from inclined wind is due to vacuum on the rear of the plate. In view of the limited possible total vacuum at any velocity, the ratio of vacuum to positive pressure in high destructive winds will necessarily be much smaller than with low winds, indicating the probability that the relatively high percentage of normal pressure from inclined wind noted by Eiffel would not be maintained in high winds.

Langley's (4) experiments do not show any sudden rise in normal pressure from wind inclined at 37 deg. or that there is a critical point in the pressure curve for winds inclined at from 0 deg. to 45 deg., the range for which experiments were made.

Uplift

The uplifting tendency frequently attributed to tornadic winds and the vertical component of winds deflected upward by the inclined sides of high coal and ore piles indicate the desirability of recognizing the effect of this possible uplift of unknown but apparently considerable amount in calculating the stability of bridges against overturning and skidding and in reduced tractive effort available in traveling against the wind. Entirely aside from the element of reduced available tractive effort due to uplift, it is not the custom under conservative management to attempt to traverse bridges in either direction in high winds unless in emergency it be for the purpose of squaring them up with their tracks when blown into skewed position by the wind.

Pressure Formula and Coefficients

The U. S. Weather Bureau (1) formula, as derived by Marvin from the 1890 Mt. Washington experiment which he conducted for the bureau, expressing the relation between wind velocity and pressure, is:

$$P = 0.0040 \frac{B}{30} V_c^2$$

in which

P = pressure in pounds on a plane surface exposed normally to the direction of the wind
0.0040 = pressure coefficient, frequently expressed by K
B = barometric reading in inches of mercury
($V_c = V$, used by Marvin) = the "corrected," "true" wind velocity in miles per hour.

At elevations near sea level, the barometric factor becomes approximately unity and can be disregarded. The Weather Bureau still retains the value 0.0040 for the pressure coefficient, which may be expressed by K, while recognizing a lower value deduced from later experiments as probably more correct.

In 1915 one of the division chiefs of the Weather Bureau stated: "However, care should be used in applying wind-pressure formulas, since it is obvious that the wind-velocity factor is quite an uncertain one. Even where there is a free wind exposure for a given structure, the wind velocity would increase with altitude, and be modified by the peculiarities of shape of the structure itself. A large margin of safety in the application of the formula

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[8] Duchemin's formulas for ordinary air pressures per square foot on thin stationary flat plates inclined to the wind are, with slight changes in wording to adapt them to structural design, as follows:

$$P_H = P \frac{2 \sin^2 \theta}{1 + \sin^2 \theta}; \qquad P_V = P \frac{2 \sin \theta \cos \theta}{1 + \sin^2 \theta};$$

$$P_N = (P_H^2 + P_V^2)^{1/2} = P \frac{2 \sin \theta}{1 + \sin^2 \theta}$$

in which

P_H = pressure parallel to the wind
P_V = pressure perpendicular to the wind
P_N = pressure normal to the surface
P = pressure on the same plane when normal to the wind, for which limiting condition $N = P \times 1 = P$
θ = angle between the wind and the surface.

The derivation of the last member of the expression for P_N is evident from the following:

$$P_N = (P_H^2 + P_V^2)^{1/2} = \left[\left(P \frac{2 \sin^2 \theta}{1 + \sin^2 \theta}\right)^2 + \left(P \frac{2 \sin \theta \cos \theta}{1 + \sin^2 \theta}\right)^2\right]^{1/2}$$

$$= \frac{P}{1 + \sin^2 \theta}\left[4 \sin^2 \theta (\sin^2 \theta + \cos^2 \theta)\right]^{1/2} = P \frac{2 \sin \theta}{1 + \sin^2 \theta}$$

as given above.

Pressures are per square foot of surface pressed.

is therefore necessary. In this connection, recent experiments indicate that the constant in the formula 0.0040 is somewhat too high, and that 0.0032 would probably be more correct." This statement is pertinent, and in consideration of its source, should carry much weight. While the Weather Bureau obviously is wise in not adopting a new coefficient precipitately or until one sufficiently reliable to merit general recognition has been deduced and confirmed by extended and repeated experiments, it would seem logical for engineers in their calculations to make the necessary allowance for uncertainties as to velocity incident to high

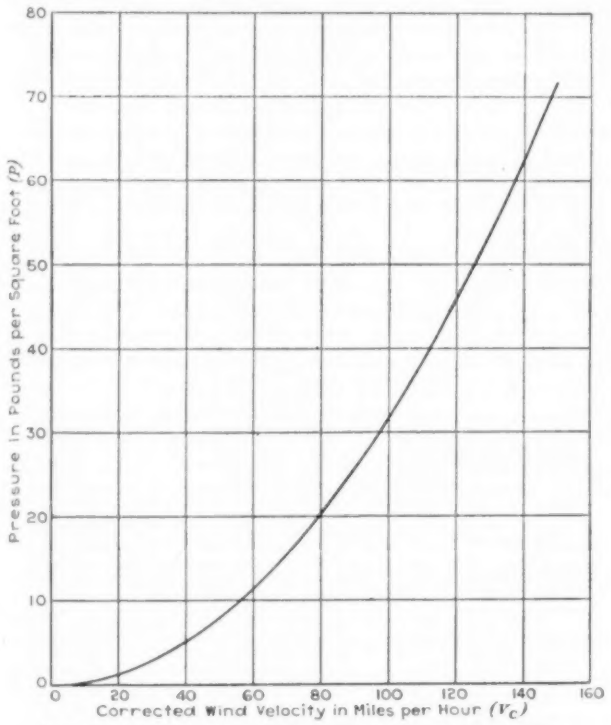


FIG. 22 WIND VELOCITY-PRESSURE CURVE FROM FORMULA

$$P = K\,(B/30)\,V_c^2$$

(For elevations at or near sea level, for which altitude $(B/30) = 1$ approximately.)

in which K = pressure coefficient, assumed at value of 0.0032 for this curve (U. S. Weather Bureau uses 0.0040 for value of K)
B = barometer reading in inches of mercury
V_c = corrected wind velocity in miles per hour
P = pressure in pounds per square foot

(From Weather Bureau formula with modified coeffcient K.)

V_c	P	V_c	P	V_c	P	V_c	P
5	0.08	45	6.48	85	23.1	125	50.0
10	0.32	50	8.00	90	25.9	130	54.1
15	0.72	55	9.7	95	28.9	135	58.3
20	1.28	60	11.5	100	32.0	140	62.7
25	2.00	65	13.5	105	35.3	145	67.3
30	2.88	70	15.7	110	38.7	150	72.0
35	3.92	75	18.0	115	42.3	...	....
40	5.12	80	20.5	120	46.1	...	....

elevation, wind fluctuations, and other causes in the velocity factor V_c, leaving them free to select for the pressure coefficient k, the best authenticated value available at the time.

Langley (4) from his Allegheny experiments, 1888 to 1890, deduced an average value for k of 0.00357.

Eiffel (5) from his experiments, conducted from 1902 to 1913, derived a value of k applicable to English units for square plane surfaces normal to the wind ranging from 0.00266 for plates 10 cm. square to 0.00323 for plates 1 meter square. Eiffel remarks, "This latter figure is probably the upper limit."

Stanton (6) found for k the value of 0.0027 for plane surfaces 2 in. sq. and 0.0032 for plane surfaces 5 ft. sq. and also for plane surfaces 10 ft. sq.

Values for pressure coefficient k ranging from well above 0.0040 to considerably below 0.0032 have been derived by various other experimenters with wind at moderate velocities. The range is so wide as to indicate the need of further experiments to determine a pressure coefficient or coefficients which can be safely and economically applied to moderate and also to high destructive velocities. A velocity-pressure curve and table are given in Fig. 22.

Also there is need of fuller and more definite knowledge in regard to velocity and pressure fluctuatons in high-velocity, highly fluctuating winds from which can be determined at least an approximate relation of the practical effect of the high peaks to the effect which the pressure formula would indicate for the average velocity for a 5-min. period and for 1 mile reported by the Weather Bureau as "maximum" velocity and "extreme" velocity, respectively.

Fortunately, fairly full information in regard to velocity fluctuations is obtainable, and the inclusion by the author in his paper of a somewhat extended series of typical high-speed charts similar to that shown in his Fig. 3, traveling fast enough to clearly indicate the velocity fluctuations and to permit determination of average velocities for selected periods, and the general range of ratios of peak to average velocities, would be of much value. From these and similar high-speed velocity charts, the corresponding theoretical pressure fluctuations could be determined by the pressure formula.

The direct determination of pressure fluctuations for high-

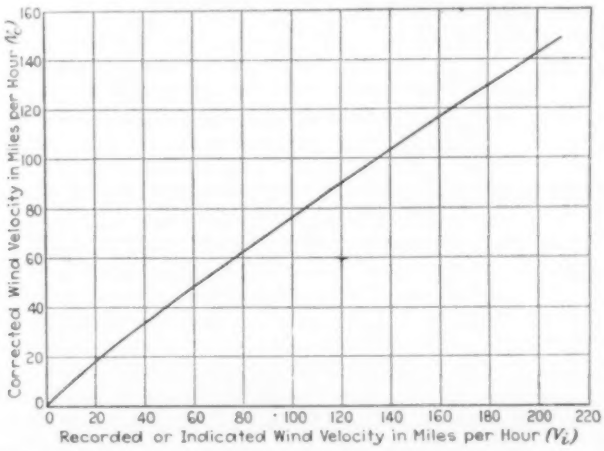


FIG. 23 CURVE SHOWING RELATION BETWEEN CORRECTED WIND VELOCITY AND VELOCITY AS RECORDED BY THE ROBINSON ANEMOMETER

From formula $\log V_c = 0.509 + 0.9012 \log V_i/3$

in which V_c = corrected velocity in m.p.h.
V_i = velocity in m.p.h. as recorded by standard Robinson anemometer

(From Weather Bureau formula slightly modified as to symbols)

V_i	V_c	V_i	V_c	V_i	V_c	V_i	V_c
5	5.1	50	40.8	95	72.7	150	109.7
10	9.6	55	44.4	100	76.1	160	116.2
15	13.8	60	48.0	105	79.5	170	122.8
20	17.8	65	51.6	110	82.9	180	129.3
25	21.8	70	55.2	115	86.3	190	135.7
30	25.7	75	58.7	120	89.7	200	142.1
35	29.5	80	62.2	125	93.1	210	148.5
40	33.3	85	65.7	130	96.4	...	.....
45	37.1	90	69.2	140	103.1	...	.....

velocity, widely fluctuating, natural outdoor winds in exposed location would not be so readily accomplished, but if a simple practical means were devised of directly graphically registering not only velocity but also corresponding pressure fluctuations for normal and inclined winds under the foregoing conditions, the

data thus made available would be of still greater value and make possible the derivation of pressure coefficients applicable to normal and inclined destructive winds.

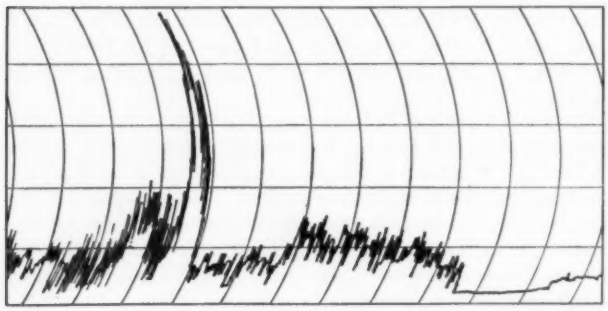


FIG. 24 SINGLE TORNADIC PEAK AT 148 M.P.H. (6/19/23)

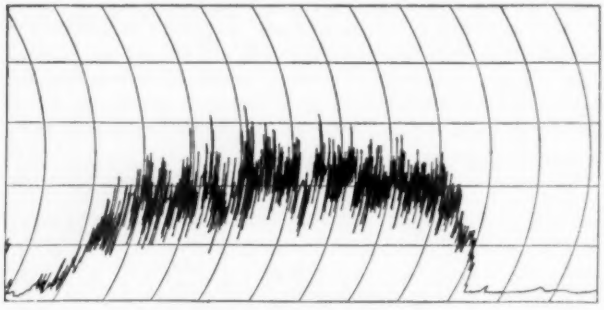


FIG. 25 RECURRING HIGH PEAKS (8/21/24)

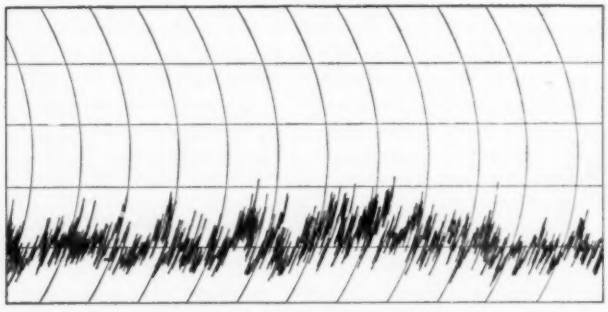


FIG. 26 HIGHLY FLUCTUATING WIND WITH 50 TO 64 M.P.H. PEAKS (7/15/25)
(Peaks continuing about 12 hours, with similar but somewhat lower velocity wind for 20 hours' additional period.)

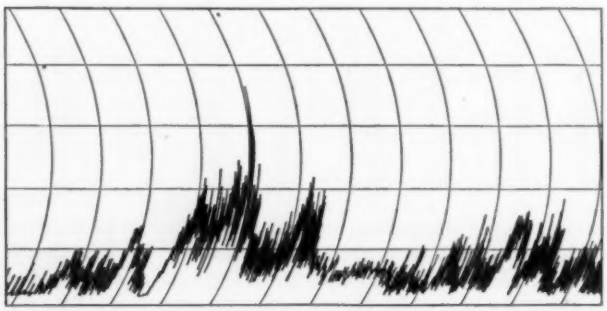


FIG. 27 FLUCTUATING WIND WITH 40 TO 110 M.P.H. PEAKS (7/11/24)

VELOCITIES

When wind velocity is mentioned, reference may be intended to "recorded," "observed," or "indicated" velocity requiring

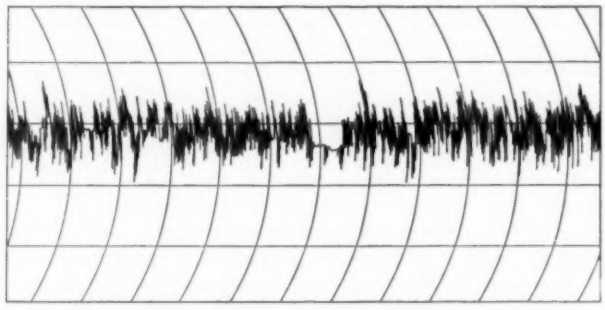


FIG. 28 HIGH-VELOCITY FLUCTUATING WIND (3/29/24)
(Similar wind with 90 to 111 m.p.h. recurring peaks continued for 11 hours and with 50 to 85 m.p.h. recurring peaks for an additional 21 hours.)

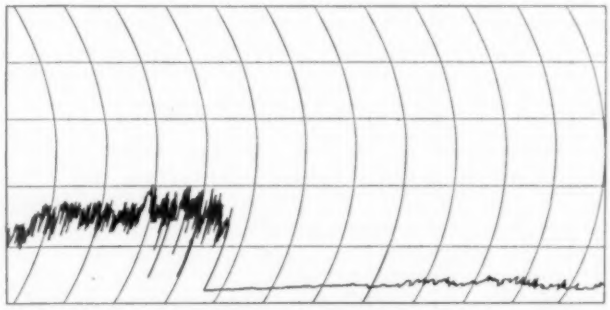


FIG. 29 INSTANTANEOUS RISE FROM 8 M.P.H. UNIFORM WIND TO 50 TO 60 M.P.H. FLUCTUATING WIND

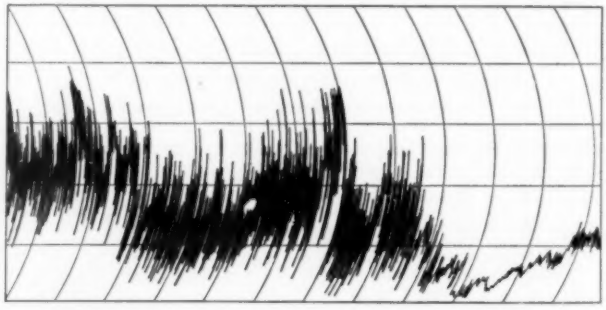


FIG. 30 VIOLENTLY FLUCTUATING WIND (11/30/22)
(With rapidly recurring extreme variations from 4 to 92 m.p.h. with 118 m.p.h. maximum peak continuing for 6½ hours.)

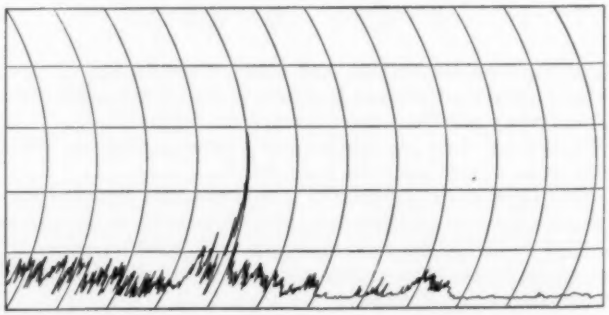


FIG. 31 SINGLE SUDDEN 87½ M.P.H. PEAK OF VERY SHORT DURATION (5/20/25)
(Preceded and followed by low wind.)

correction, or to "corrected" or "true" velocity, while under each of these heads, reference may be intended to a momentary velocity or to average velocity over one of several possible periods.

Wind velocities recorded by the U. S. Weather Bureau are average velocities in miles per hour. Velocities or travel for four selected periods are entered daily in the Weather Bureau records, namely:

(*a*) "Maximum" velocity, which is the highest average "recorded" velocity for any consecutive 5-min. period in a 24-hour day.

(*b*) "Extreme" velocity, which is the highest average "recorded" velocity for 1 mile which occurs in a 24-hour day.

(*c*) The average velocity or travel for each of the 24 hours of the day.

(*d*) The total wind travel during the 24-hour day.

The "maximum" "recorded" velocity is the one ordinarily given out by the Weather Bureau, and is therefore the velocity commonly known or referred to. The "extreme" "recorded" velocity and the other velocity data aforementioned can, however, be ascertained from the Weather Bureau upon request. the "maximum" velocity can be determined from observation of the chart while still in the instrument, while the "extreme" velocity can be obtained only by scaling after the chart has been removed at the close of the day.

Inasmuch as the ratio between the linear velocity of the cup centers of the Weather Bureau type of the Robinson anemometer and that of the wind is not constant, but increases at an accelerated rate as the velocities increase, the "recorded" velocities as registered by the anemometer are not "true" velocities, for the determination of which a logarithmic formula derived by the Weather Bureau from experiments is required.

The commonly used wind pressure formula is based on "true" and not "recorded" or "indicated" velocity, although it is the "recorded" velocity to which reference is commonly made. A corrected velocity curve and table are shown in Fig. 23.

The formula for "corrected" velocity, with slight change in symbols, is:

$$\log V_c = 0.509 + 0.9012 \log \frac{V_i}{3}$$

in which

V_c = "corrected" or "true" velocity of wind in miles per hour

V_i = "recorded" or "indicated" velocity of wind in miles per hour

$\left(\frac{V_i}{3} = v \text{ of the original formula}\right)$ = linear velocity of cup centers of anemometer in miles per hour

$= \frac{1}{3}$ "recorded" or "indicated" wind velocity in miles per hour.

The writer wishes to acknowledge his indebtedness for data used to the several works listed under Bibliography, to which the index numbers in the text refer.

Bibliography

(1) "Wind Pressures and Wind Velocities—Results of Experiments on Mt. Washington," by C. F. Marvin, since 1913 Director of the U. S. Weather Bureau, *Engineering News*, Dec. 13, 1890, vol. 24, pp. 520 and 521.

(2) "Wind Pressure on Engineering Construction," by W. H. Bixby, Captain (later Brigadier General) of Engineers, U. S. A., *Engineering News*, vol. 33, March 14, 1895, pp. 175 to 184; compiled by the author as Appendix C of the Report of September 29, 1894, of the Special Army Engineers Board as to the "Maximum Span Practicable for Suspension Bridges."

(3) Memorial de l'Artillerie, No. V, French War Department

(4) "Experiments in Aerodynamics," by S. P. Langley, Smithsonian Contribution to Knowledge 801, vol. 27, no. 1; 1st edition, 1891; 2nd edition, 1902.

(5) "The Resistance of the Air and Aviation, and Supplement," by A. G. Eiffel. "Nouvelle Recherches," 1914, A. G. Eiffel.

(6) "Hand Book of Modern Aerodynamics."

(7) "Descriptive Meteorology," by Willis L. Moore, Chief U. S. Weather Bureau, 1895 to 1913.

(8) Circular D, Instrument Division, U. S. Weather Bureau Bulletin 811, Appendix 2, 1910.

(9) Scientific Papers of the Bureau of Standards, vol. 20, 1924 to 1926, paper S-523, "Wind Pressures on Structures," by Hugh L. Dryden and George C. Hill.

(10) National Physical Laboratory, Report for 1929.

Anemometer charts are shown herewith (Figs. 24 to 31, inclusive), that were selected to show high peaks and characteristics of fluctuating winds of several types at Superior, Wis., at the head of the Great Lakes.

These charts were made by a Burton recording and indicating anemometer mounted on the mast of the North Western Fuel Company's No. 5, 551-ft. clear-span coal-handling bridge. Each chart shows a curve for about a 3-hour period. The duration of similar wind condition of reduced intensity is indicated by notation for the individual charts. The chart travel is 3 in. per hour. "Recorded" velocities are shown and specified.

Author's Closure

The author is pleased to note the favorable comments of various discussors of the paper presented. The aim in its preparation was to cover the whole field as well as possible without making the presentation unduly long. Many of the features of construction, control, or protective equipment might have been greatly elaborated upon, and many individual or special cases discussed, but only a general discussion was considered possible. Several of the discussors have made comments which undoubtedly add to the completeness or clarity of certain points, and the author appreciates their interest. In some cases, questions have been raised to which the writer wishes to reply.

Mr. D. C. Wright has requested information as to the adjustment of the anemometer relay used by the author. This relay can be adjusted to trip at a given velocity and to reset at a wide range of lower velocities. Assuming a tripping point of 40 m.p.h., the contacts may reclose at 38 miles or 35 miles, or 30 miles, or any point intermediate or below these values. In general, a three- or four-mile differential is desirable, since better protection is thus obtained, and any tendency to rapid opening and closing of control circuits or "fluttering" when the wind velocity fluctuates a little above and below the tripping value is eliminated. This relay has another operating characteristic which also tends to promote increased safety in operation. Magnetic hysteresis of the relay frame causes the tripping and resetting point to become temporarily lowered after exposure to high wind velocities. In other words, with a normal tripping adjustment of 40 miles and a reset point of 37 miles, a wind velocity peak of 60 to 70 miles may reduce the tripping points, as the wind decreases, to 38 and 35 miles, respectively. Higher peaks may further reduce the settings. This temporary effect is partially erased as the wind drops down to 10 or 15 miles, and completely disappears when a calm occurs. To put it in other words, the relay "remembers" the high peak velocities, and enforces greater operating caution until the storm is over.

Mr. Heinselman emphasizes the importance of proper training and experience of bridge operators, and the author heartily concurs in these remarks. Cautious and well-trained operators constitute in themselves a great safety asset. The instruction and education of dock superintendents and their forces in the effective use and maintenance of all safety appliances through

inspection bureau bulletins and reports also contributes a great deal toward the prevention of losses.

The author is convinced that the methods of testing the effectiveness of rail clamps and similar devices are in many cases too indefinite and uncertain. There are so many types of motor and control equipment, and so many types and percentages of traction, cable or pinion drives, that a general rule to apply "two or three points of power" or "until drive wheels slip," is quite useless. Undoubtedly each clamp should be tested separately, with an ammeter connected in the motor circuit so that the pressure against the clamp under test may definitely be determined. A certain current value should be set for each individual bridge by the inspector or bridge builder, under which the clamps must hold. The author has found that good clamps in good condition will hold singly up to 150 to 200 per cent or more of full load current applied to both series-wound motors on each leg of a four-motor bridge. The torque developed by shunt-wound motors at 150 to 200 per cent load current would naturally be considerably less than in case of series motors, and in case of induction motors 150 to 200 per cent load would represent a still lower torque or propelling force. The bridge builder or designer would seem to be the logical authority to specify the current value which will constitute an adequate test. A very little oil or grease on the rail may cause the wheels to slip at 100 per cent load or less, while oil or grease on the clamping surfaces may cut the holding power of the clamp in half. A monthly test of each clamp backed up by a graphic recording meter chart showing the actual test current might be suggested as a most effective system. Similar metered load tests might also be made against the solenoid brakes applied to the travel motors.

In the testing of rail clamps, on steam-propelled rigs, the author wonders if suitable and effective tests could not be made with temporarily reduced boiler pressure applied direct to the propelling engine. This would seem to eliminate the possible danger of full-pressure tests.

To insure 100 per cent effective operation of rail clamps, it is important that worn or defective parts be replaced promptly with genuine replacement parts supplied by the manufacturer, or with parts which are known in every way to duplicate the physical and metallurgical properties of the original parts. For instance, certain highly stressed parts of a cam-operated rail clamp are made of heat-treated alloy steels to safely withstand the enormous gripping pressures exerted under power tests or under wind pressure. The ill-considered substitution of steel parts of low elastic limit and low tensile or compressive strength may almost completely ruin the effectiveness of the clamp.

The effectiveness of wood or steel wedges of proper form when applied to *idler* wheels can hardly be questioned. However, there is some doubt as to their value on 100 per cent wheel-driven bridges. If the solenoid brakes are effective, the wheels should slide on the runway rails without turning. The author is familiar with several cases where at least one end of a bridge was moved by a storm for some distance along the runway with the wheels locked and sliding. The flat spot on every wheel furnished incontestable evidence that the wheels did not rotate. If the wheels do not turn, they will simply push the wedges along ahead of them without any appreciable retarding effect.

The collection of wind-pressure and wind-velocity data, formulas, and tables arranged by Mr. Hutchinson brings together a great deal of scattered information, frequently difficult of access, and this section ought to be of particular interest to engineers whose work may include wind-pressure problems. The remarks and explanation covering the proper understanding and interpretation of the various terms used in referring to wind velocities ("recorded," "observed," "indicated," "corrected," "maximum," "extreme," etc.) are also pertinent and valuable.

Mr. Hutchinson does not mention the recent change (1928) of the Weather Bureau from the Robinson four-cup anemometer to the Ferguson three-cup model as the standard instrument of the bureau. The new three-cup instrument has a much smaller error than the older four-cup instrument, and at all ordinary velocities may be considered substantially correct.

The following condensed correction tables show the comparative correctness of the indicated velocity values of the two instruments:

Old Four-Cup Anemometer

Indicated velocity, miles.....	5	10	20	30	40	60	80	100	120	140
Corrected velocity, miles.....	5	9	17	25	32	47	62	76	91	106

New Three-Cup Anemometer

Indicated velocity, miles...	5	10	20	30	40	60	80	100	120	140
Corrected velocity, miles...	5	10	20	29	39	58	77	95	114	133

The author may point out that the velocity record charts, Figs. 3 and 4 of the presentation, as well as the charts, Figs. 24, 26, 27, 28, 29, 30, and 31, furnished by Mr. Hutchinson, are all made on the basis of "indicated" velocities of the Robinson Scale, and are subject to correction for true velocities according to the Robinson Correction Table. Current anemometers of the Burton type are now calibrated according to the new standard of the Weather Bureau.

Manufacture of Large Weldless Forged-Steel Pressure Vessels

By JOHN L. COX,[1] PHILADELPHIA, PA.

This paper describes in detail the various steps taken in the manufacture of large weldless forged-steel cylinders for high-pressure boiler drums and for various uses in the oil-refining and chemical industries. It covers the melting, casting, and heating of the ingot; cutting-off, piercing, and expanding the billet; forging, treating, and machining the cylinder; and illustrates a number of ways in which the ends of the cylinder can be closed.



DURING the last few years a great change has taken place in several important industries in the United States. Formerly almost entirely dependent on Europe for our dyes or the intermediates from which they were made, the years since the Great War have witnessed an astounding growth in our manufacture of chemicals, which has resulted in our practical emancipation from this dependence.

Once our only source of nitrogen in quantity was Chilean saltpeter, sodium nitrate, of which the War Department carried an enormous stock to provide for contingencies. The substitution of manufactured for natural ice and the more extensive use of fertilizers as our virgin lands lost their fertility have called for ammonia and nitrates in great quantities, a demand now largely met by the fixation of atmospheric nitrogen.

Gasoline, once a troublesome by-product in the production of kerosene, through the perfecting of the internal-combustion motor has become the most valuable petroleum product. Every means that could increase its yield has been eagerly sought, and it has been discovered that destructive distillation under pressure would in some cases increase the yield as much as thirteenfold.

The necessity for the more economical generation of power has led to a great increase in boiler pressures and dimensions. Today no one comments on a pressure of 1400 lb., and much higher pressures are in actual use.

These advances have been made possible largely through the development and general introduction of steel vessels sufficiently capacious for large-scale operation and strong enough to withstand high pressures at temperatures generally elevated. When the requirements are not too severe, riveted or welded vessels may answer satisfactorily, but the most exacting service requires the use of weldless steel vessels, forged entire in one piece. It is with the manufacture of such vessels that this paper deals.

The making of a good cylinder begins in the open-hearth furnace. The steel may be either acid or basic open hearth, the acid process being preferred for steels of high quality as it affords better opportunity thoroughly to settle up the metal. If very low phosphorus and sulphur are demanded, it may be necessary to use the basic process because of the difficulty in securing suitable raw materials for the acid furnace, or the impurities may be reduced by the basic process and the metal finished on the acid hearth.

In both cases the steel is carefully melted, brought to good condition in the furnace, where practically all the finals are added, and a very moderate addition is made of deoxidizers to basic metal, none to acid. Generally the service requires only a carbon steel, but sometimes a stronger alloy steel must be used, and at times special conditions may require the use of special compositions. Thus for the synthesis of ammonia it has been found that a steel with carbon below 0.30 per cent and containing about 2 per cent or more of chromium and 0.18 per cent of vanadium is especially suited, being very resistant to intergranular penetration by hydrogen.

Since it is important that the steel be as free as possible from sonims, good practice dictates that it be melted hot, and instead of being poured at once into the ingot mold from the ladle into which the furnace is tapped, that it be held in the ladle as long as is considered prudent for the size and temperature of the heat, so as to afford ample time for the particles of slag to coalesce and, by virtue of their less specific gravity, to rise to the surface of the steel. For a 70-ton heat of 0.25 per cent carbon steel this time of holding may be as much as half an hour.

For very large ingots it is necessary to combine the heats of several furnaces. Arrangements must be made to permit the practically continuous flow of metal into the mold, for otherwise a transverse fissure, termed a "stop pour," may readily result.

Pouring may be either from the bottom (through a refractory-lined conduit alongside the mold, communicating with similar passages in the mold bottom and its cover plate, on top of which the ingot is cast) or from the top (usually through a small intermediate ladle called a head box, the function of which is to remove the velocity head of the stream due to the height of the metal in the ladle). The first method exposes the metal less to oxidation and gives cleaner surfaces, but may introduce slag if the runner bricks are eroded by the steel. The second method exposes the metal more and gives poorer surfaces because there will inevitably be some splash congealed on the sides of the mold, but it introduces the minimum of fluxed refractories.

The ingot mold may be round and of sand, as was that for this emergency 114-in. ingot of 375,000 lb., but for most purposes this is not desirable because the slow cooling of an ingot cast in sand induces the formation of very large crystals, requiring

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[1] Assistant to President, The Midvale Company. Mem. A.S.M.E. Mr. Cox was born in Philadelphia in 1866. He was educated in private schools, receiving finally the degree of Mechanical Engineer from the Stevens Institute of Technology in 1887. In 1888 he entered the service of the Midvale Steel Company, with which company or its successors he has continued, having been in charge of the forge, armor, projectile, and research departments, and is now Assistant to the President. Mr. Cox has taken out patents on armor plate, projectiles, and alloy steels, and is the author of various technical papers, as well as the article on recommended practice for large forgings in the National Metal Handbook of the American Society for Steel Treating.

Presented at the Fourth National Iron and Steel Meeting, Chicago, Ill., September 24 to 26, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

more heat, or more time at heat, or more forging, or treatment, to produce as fine a final grain as would have resulted had the ingot been cast in an iron mold, as was this 108-in. octagon ingot of 455,000 lb.

Generally the mold is of cast iron and is an octagon or higher polygon in cross-section, rarely round. To increase the ratio of perimeter to area in the ingot, with the object of preventing its cracking through cooling contraction of the ingot skin, the sides of the ingot polygon are usually made concave, with small corner radii, or are crenate. The walls of the mold are thick, so as to chill a thick skin on the ingot, which will not be burst by ferrostatic pressure from above when the ingot shrinks from the mold and loses its support.

Sufficient taper is given the mold to enable the ingot readily to be stripped, the amount varying from about $^1/_8$ to $^3/_8$ in. in

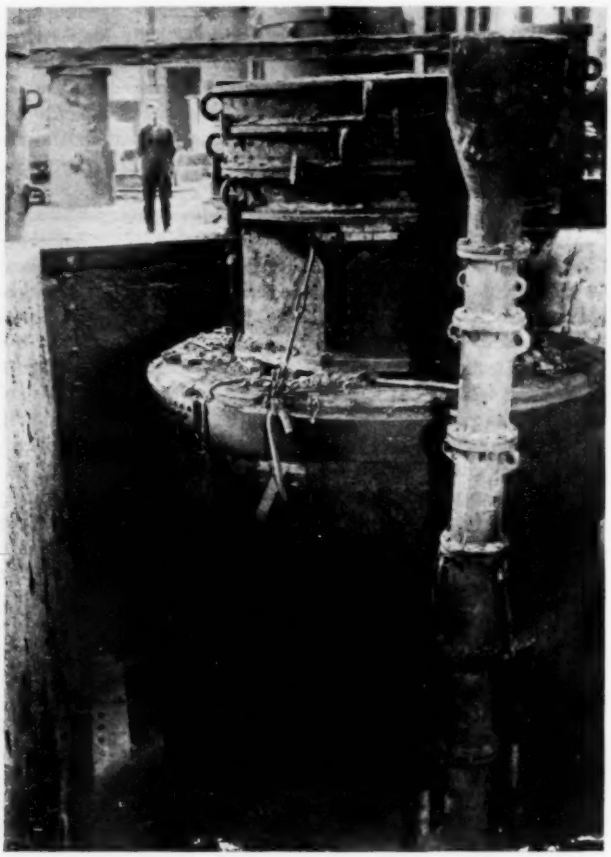


FIG. 1 MOLD FOR BOTTOM POURING

diameter per foot of length, and the inner surface is carefully coated with a mold wash.

The ingot may be cast with the large end up or down, but the best practice is to cast it large end up, as there is then a natural tendency for solidification to proceed regularly from the bottom upward. At the upper end of the mold is fitted a refractory-lined sink head in which the metal cools more slowly than in the chilling mold and remains fluid to feed the pipe caused by the contraction accompanying solidification of the steel in the ingot body. Toward it also gravitate the non-metallic impurities unavoidably present in the steel. We have tested scores of heats while in the electric furnace and found them perfectly clean, but every one showed foreign matter when examined in the ingot.

The proportion of the total ingot weight contained in the sinkhead varies with the design of the mold, the character of the metal, and its ultimate use. For the usual run of cylinder ingots, cast in heavy iron molds, a sinkhead of 15 per cent will suffice.

If the ingot be cast small end up, every precaution must be taken to feed the shrinkage or there will be in the upper end a



FIG. 2 POURING A LARGE INGOT

central internal void called a pipe. It may result from too small or too light a sinkhead or from an improperly designed mold. For a hollow forging this might be thought unobjectionable but it is not, because a concentration of impurities occurs around and immediately below the pipe which might not be entirely removed subsequently and should therefore be avoided.

Much judgment is required in pouring large ingots, to avoid cracking by too fast pouring or bad surfaces by too slow pouring.

When the mold has been filled, the upper surface of the metal in the sinkhead is covered with an insulating material, usually sand, to delay solidification, and under certain circumstances it may be necessary, after some hours, to bring up fresh metal to fill the shrinkage which has taken place.

After casting, the ingot may be stripped in a few hours if poured small end up, but must solidify much more thoroughly before it can be lifted from the mold when poured big end up. In both cases it must remain vertical until completely solidified throughout, or a pipe will result. It is our custom to allow a 63-in. ingot to stand from 12 to 15 hours before throwing it down, an 82-in. ingot about 20 hours, and a 108-in. ingot from 40 to 48 hours.

Very large ingots contain such great internal stresses, and the initial heat is so valuable, that it is highly inadvisable to allow them to cool unduly before charging them into the forge furnace. Even to anneal them for decapitating and boring when cold is dangerous, although for medium size ingots it is often done, particularly abroad. In such cases it is generally advisable to anneal the ingots before allowing them to go cold.

At first sight it might be thought a good plan to cast hollow ingots for hollow forgings, such as the 78-in. one shown, but practical experience has not been satisfactory with large hollow ingots. The cylindrical shrinkage weakness is always present in the wall, and it is most difficult to avoid scabs on the interior

surface; yet this system has been followed in some cases, as in medium-size ingots for hoops for the Swedish artillery, because of its cheapness.

The forge furnace is generally of the movable-hearth type for very heavy ingots intended for cylinders. It may be oil- or gas-fired or may use powdered coal for fuel. In it the ingot is slowly and carefully brought up to a forging heat in the course of many hours, approximately an hour to the inch of diameter,

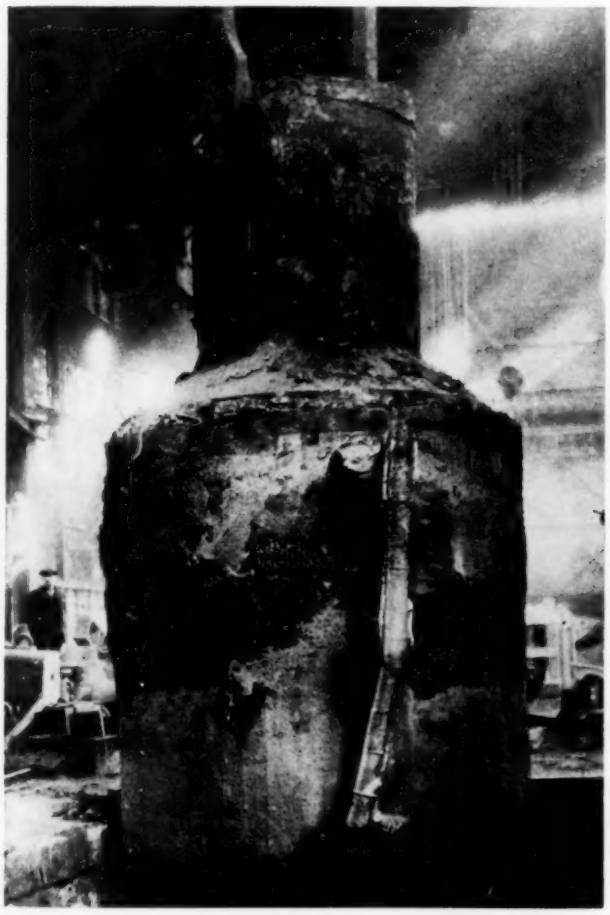


FIG. 3 SAND-CAST 114-IN. INGOT OF 375,000 LB.

and is then thoroughly soaked until every part is fully up to the desired temperature. It is advisable to heat the metal at first as highly as its composition will allow without injury, for a high temperature hastens the solution of the ingot structure—a much finer grain resulting when the metal finally cools than would follow were the temperature too low.

After a sufficient heating, the usual temperature for low-carbon steel being about 2200 to 2250 deg. fahr., the ingot is removed from the furnace, placed under a powerful hydraulic press, and the sinkhead sliced off with a large cutter.

The block is then reheated thoroughly, stood on end under the press, and upset to a convenient length for punching, the extent depending chiefly on the vertical clearance of the press. This upsetting requires more power than any other press operation except forging thin plates. Thus, to upset a block averaging 7 ft. $7^3/_8$ in. in diameter and 11 ft. 1 in. long to 8 ft. 11 in. in median diameter by 7 ft. 3 in. long (i.e., about $7^1/_2$ ft. by 11 ft. to 9 ft. by 7 ft.), requires 7000 tons, an average pressure of 1560 lb per sq. in. on the median area. The pressure required rises rapidly with decrease in thickness, the internal friction being greater the thinner the piece. Thus to reduce a block 30 in. in diameter by $7^3/_{16}$ in. thick to 42 in. diameter by 4 in. thick requires a pressure of 8100 lb. per sq. in. over five times as much.

After reheating, the upset block is next punched or trephined with a hole about 24 in. in diameter. This hole is generally enlarged by a drift to about 27 in. in diameter, to admit using a stiffer mandrel for the expending operation later on.

If at this stage the diameter of the piece be considerable compared with its length, it is usual to draw out the billet on a mandrel between plain dies, or by the press crossheads if too large to admit of using dies, until of a length that is slightly shorter than the bottom crosshead, and of a diameter small enough to permit expansion of the bore to 3 or 4 in. larger than the largest mandrel to be used in the final forging, for the longer the piece the more easily it is kept from going long-cornered.

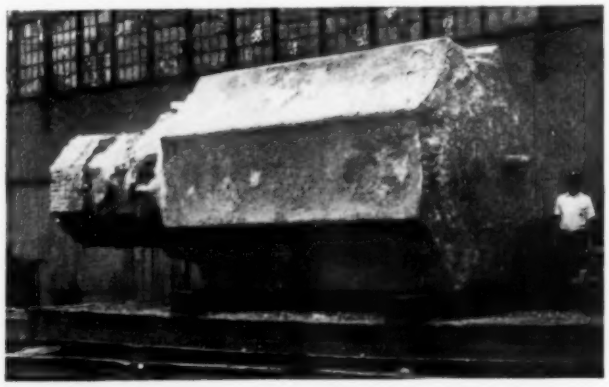


FIG. 4 AN 108-IN. OCTAGON INGOT OF 455,000 LB.

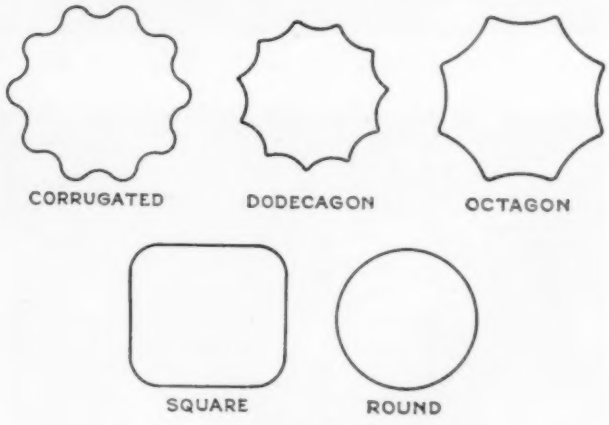


FIG. 5 INGOT CROSS-SECTIONS

The largest mandrel in use in the United States today is about 72 in. in diameter.

Drawn to a sufficient length, the billet is swung on a mandrel supported at each end close to it upon the bottom crosshead of the press, and the hole is enlarged by a narrow-faced top die parallel to the mandrel, which thins the walls without sensibly increasing their length, the billet being revolved on its axis through a small angle after each press stroke. Since there is almost no longitudinal flow of metal, the diameter of the hole rapidly increases as the walls are thinned, until the desired internal diameter has been reached.

After the press dies have been changed to a bottom vee die

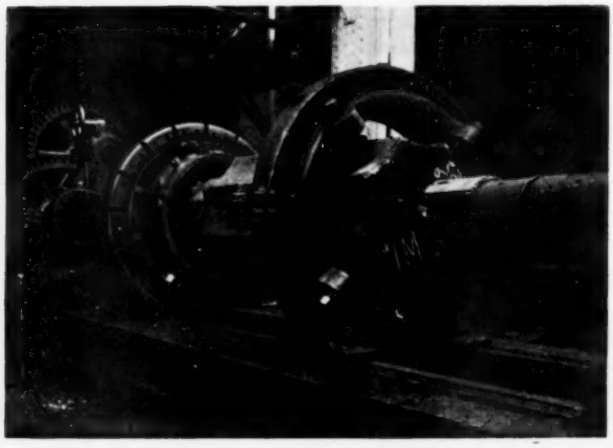


FIG. 6 BORING AN INGOT

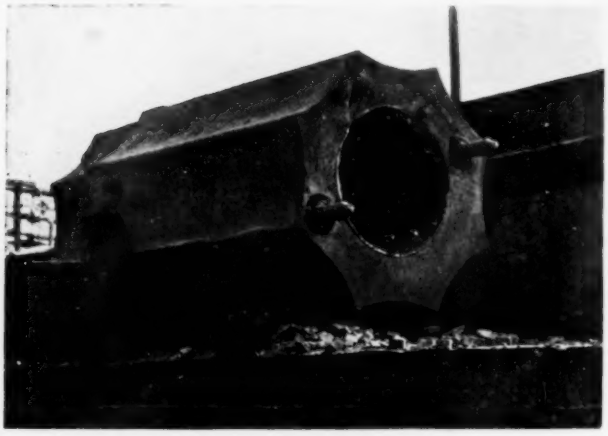


FIG. 7 HOLLOW-CAST INGOT, 78 IN. OCTAGON



FIG. 8 CAR-BOTTOM FORGE FURNACE WITH 108-IN. INGOT

and a short plain-top die of the same width, a slightly tapered mandrel is inserted in the reheated billet and the bore is slightly reduced by forging, so as to crack the scale, which if left in the bore would tend to key the mandrel fast and prevent its withdrawal. After the scale has been raked out, the mandrel is reinserted and forging is begun with heavy pressure, which closes the walls of the billet down upon the mandrel and then reduces their thickness. Confined circumferentially between the three die faces and the mandrel, the metal flows longitudinally as the exterior diameter is reduced. The whole forging is thus roughed down until it is of such diameter that it is possible to bring a short section at mid-length to the final forged diameter in one heat. This having been done, first one end and then the other is brought to final sizes, as many heats being taken as the length of the piece may require.

For the finishing operations a furnace with fixed bottom is



FIG. 9 PUNCHING A LARGE BLOCK

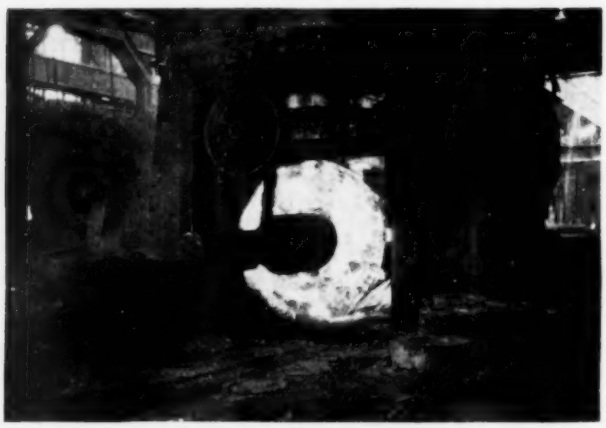


FIG. 10 ROUGHING ON MANDREL ON PLAIN DIES

desirable. Into it, after each heat, there is inserted as much of the end of the forging as remains unfinished. As in every heating, before a piece is removed from the furnace for forging it is turned upside down and soaked for a while, to allow the colder bottom side to equalize in temperature. Were this not done, the hole would quickly get out of center. It is usual, also, when the piece is inverted, to push into the furnace a little of the finished section. It is not there long enough to be injuriously affected by the heat, and the step avoids too sudden changes of temperature in the forging.

In certain cases of especially sensitive steels it may be necessary to avoid complete cooling of any part of a forging before it has received its preliminary annealing.

The angle between the faces of bottom vee dies varies from about 105 to about 120 deg. A large angle makes easier the removal of the mandrel after each heat and permits the making of a greater range of diameters in one die, but has the disadvantage of increasing the arcs between the top die and the sides of the vee. The longer these arcs the greater is the tendency of the unsupported wall of the forging not to shorten but to spring away from the mandrel and form a peak which, in turn, has to be pressed back into contact with the mandrel before the next reduction in wall thickness can be made. It is for this reason that, as the internal diameters of forged cylinders increase, the wall thickness must be increased, to control the "breathing" in the vee dies. An attempt to forge a cylinder

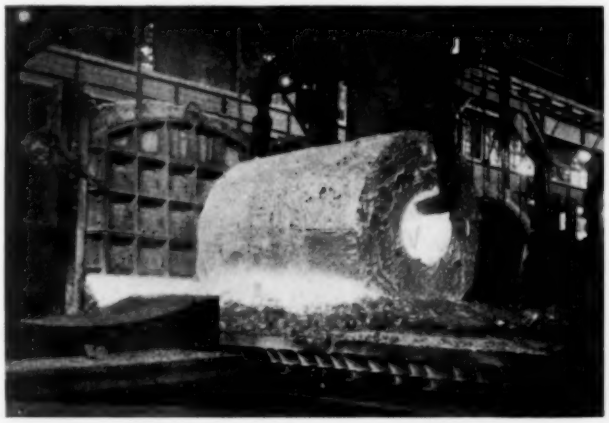


FIG. 11 ROUGHED BILLET ON WAY TO PRESS

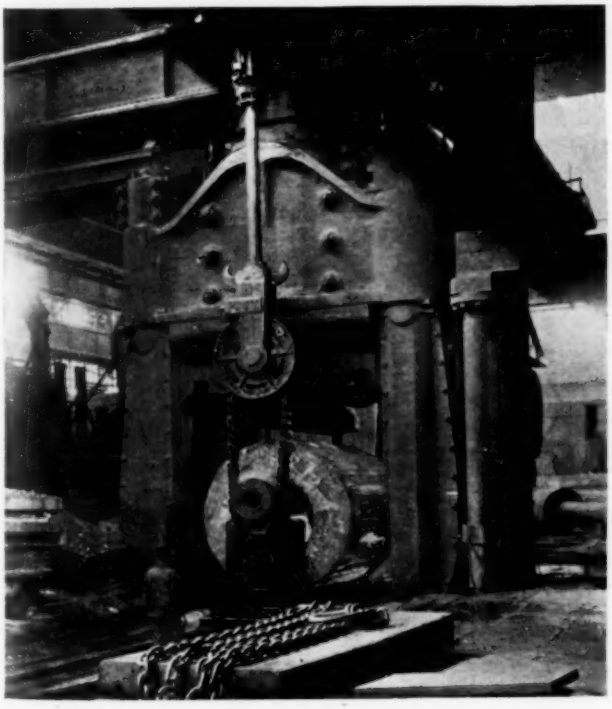


FIG. 12 EXPANDING A BILLET

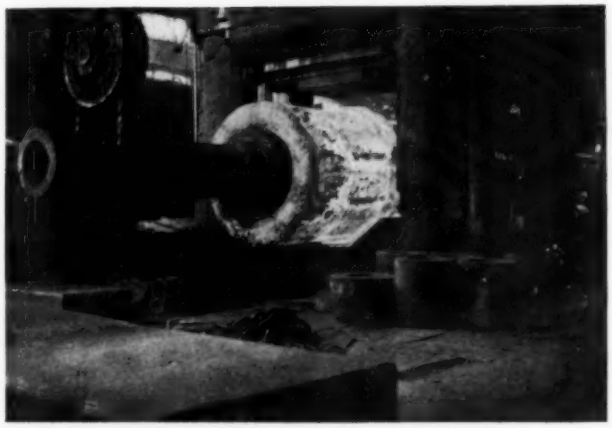


FIG. 13 ROUGHING IN THE VEE DIES

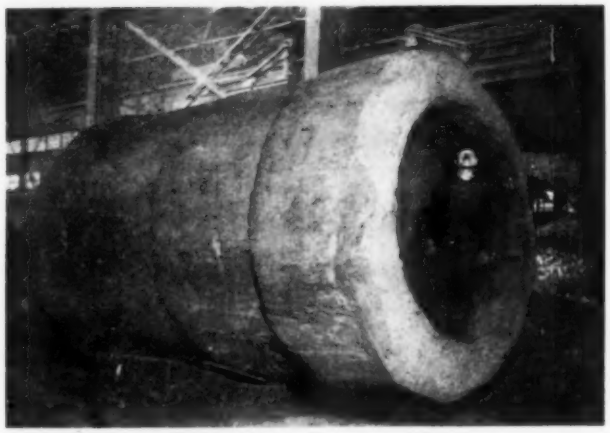


FIG. 14 PARTLY ROUGHED BILLET

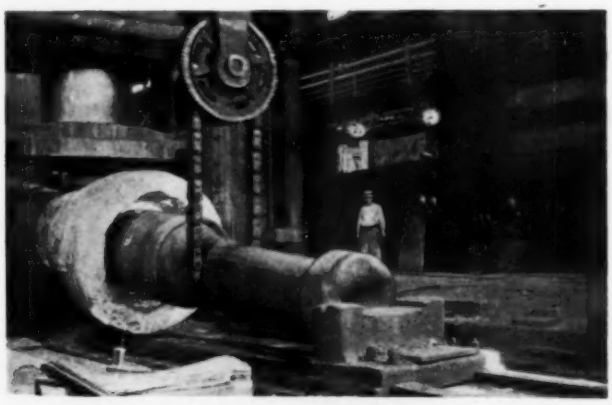


FIG. 15 PUSHING IN THE MANDREL

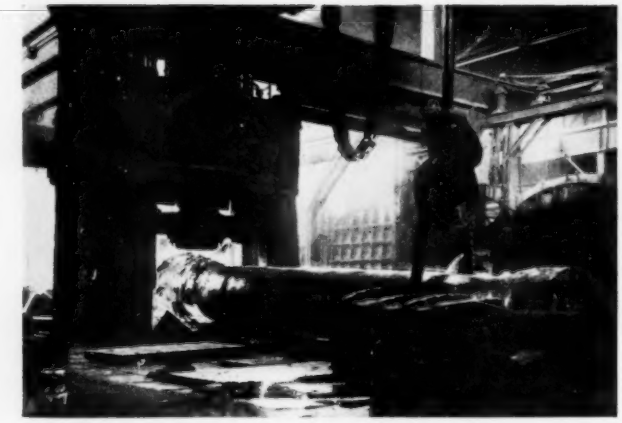


FIG. 16 FORGING IN THE VEE DIES



FIG. 17 CYLINDER OF 52 IN. AS FORGED



FIG. 18 ROUGH-BORING THE CYLINDER



FIG. 19 CLOSING-IN OPERATION

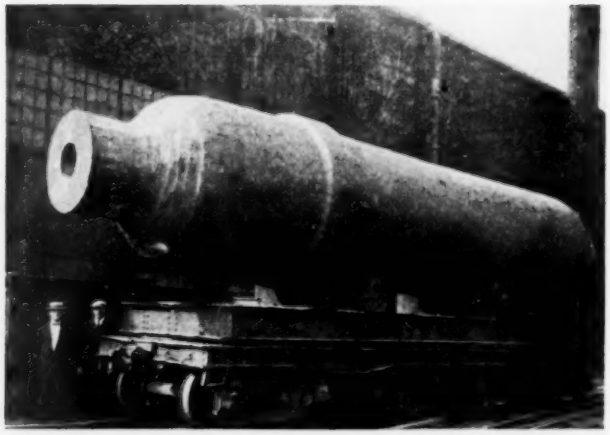


FIG. 20 CYLINDER FINISH-FORGED, WEIGHT 265,000 LB.



FIG. 21 TREPHINED TEST

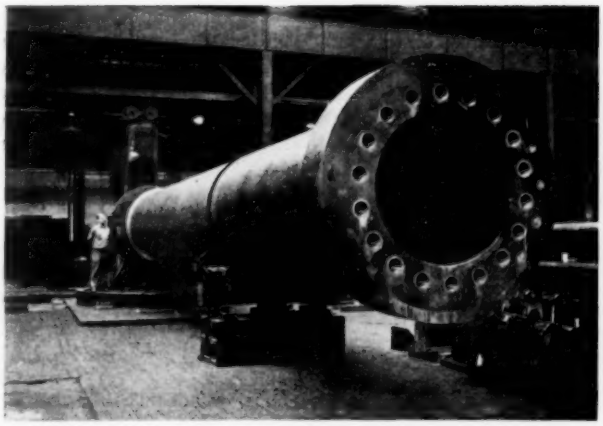


FIG. 22 HEAD TO BE ATTACHED BY STUD BOLTS TO 220,000-LB. CYLINDER

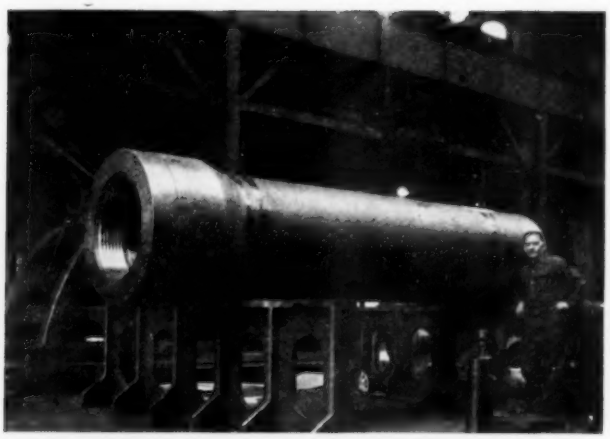


FIG. 25 THREADED CYLINDER



FIG. 23 FLAT HEAD ON CYLINDER 60 IN. INSIDE DIAMETER

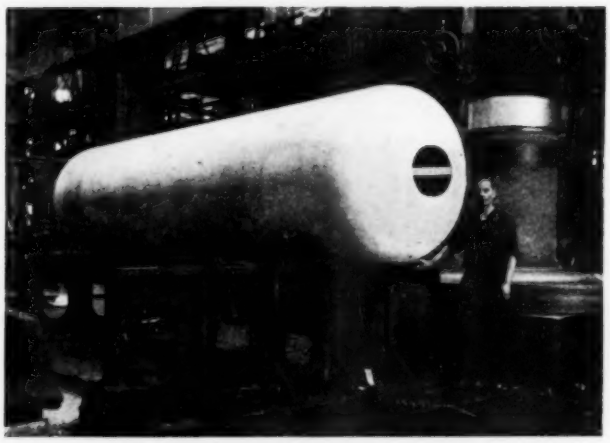


FIG. 26 BOILER DRUM WITH ELLIPTICAL MANHOLE

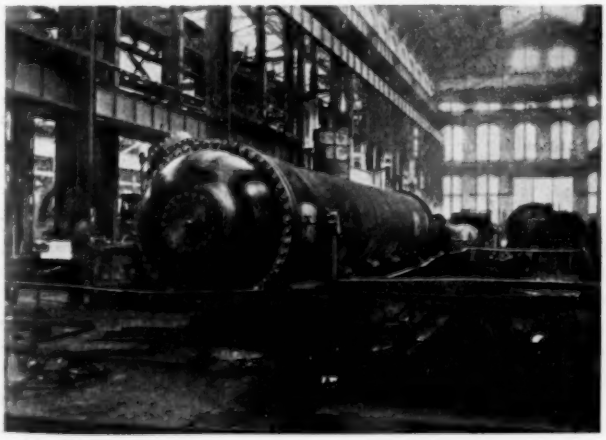


FIG. 24 DOMED HEAD ON CYLINDER

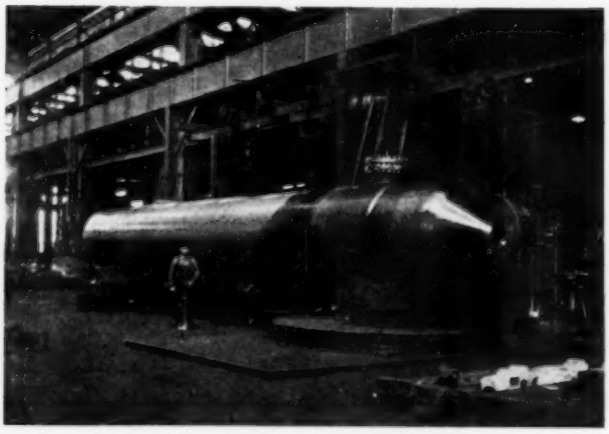


FIG. 27 CLOSURE WITH CONICAL FITTING

too thin for its diameter might require so many reheatings that the loss by scaling would exceed the saving in weight of thinner walls, and a short forging would result.

At each end of the forging there is left a greater or less length of enlarged outer diameter, to provide metal for the closure of the finished vessel, the length and diameter depending on the type of closure. A head to be attached by studs, or by bolts through a flange, requires but a short section of sufficient diameter, but if the vessel is to be closed in under the press there must be provided enough metal to allow for the final thickness required

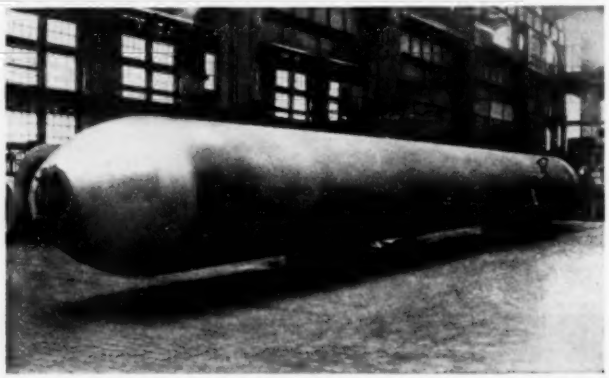


FIG. 28 BOILER DRUM WITH CIRCULAR MANHOLE

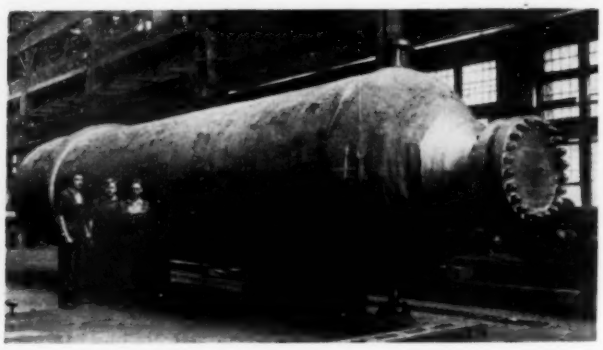


FIG. 29 FLANGED END WITH BOLTED COVER

as well as for the unavoidable elongation accompanying the operation.

This stage of the forging finished, the cylinder is generally given an annealing above the upper critical temperature of its composition, to refine the grain and relieve stresses or to secure the physical properties desired.

From the annealing furnace the cylinder goes next to the machine shop, where, if required, test pieces are cut from it to determine its physical properties. If so ordered, it is rough- or finish-bored, the first operation being performed by tools set in the ends of the arms of a spider fed along a heavy stationary boring bar, the forging being revolved in a lathe. Finish-boring is generally done by two cutters at opposite diameters of a wood- or babbitt-packed skeleton steel bit carried in a socket in the end of a heavy boring bar which is fed into the forging as it revolves in a lathe.

Such exterior work as is required on the cylinder is done now. The body may be rough- or smooth-turned, supporting rings cut from thicker bands left on the exterior at the press, and if the ends of the cylinder are to be closed in, the outside of the heavy collars left at the ends is contoured to give the amount and distribution of metal that calculation, guided by experience, shows to be necessary to secure the final shape and dimensions desired. In this work, using heavy tools of high-speed steel, one can take a cut of from 1 in. to $1^1/_4$ in. depth, with a feed of $^1/_4$ in., at about 20 to 25 ft. per minute on a simple mild steel.

The allowance for machining depends somewhat upon the internal diameter. Since it is impossible to make the work hug a large mandrel as snugly as a small one, there is, in large cylinders, a greater probability of pockets existing in the bore, places where the metal has stood off the mandrel, leaving the bore large. Generally, 1 in. on a side, inside and out, is sufficient allowance for machining a large cylinder.

After the ends have been contoured they are reheated and closed in to measure or to templet, in suitable dies. The operation requires much skill and may be very troublesome. If conditions are not exactly right, as the bore decreases, wrinkles first form in the mouth of the forging, and then the sides of the wrinkles close together, forming galls. If closing-in continues, the galls quickly turn into deep splits. If these galls occur, before they have progressed too far they must be removed by boring after the forging has cooled, and the work of closing in is then resumed.

If the cylinder be a boiler drum, after closing-in it is required

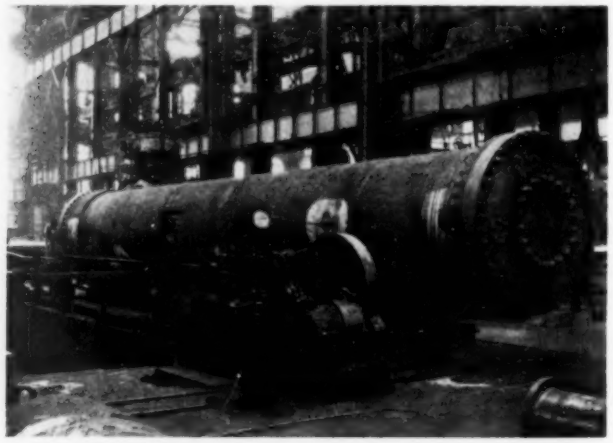


FIG. 30 HYDROSTATIC TEST

to be reannealed all over at a temperature above the upper critical point, but below the temperature of its first annealing, so as not to affect the physical properties already determined. If the physical properties demanded require quenching and tempering of the forging, the necessary treatment is now given, but it is generally considered better to attain these properties by annealing or by free-cooling in air, from a quenching temperature, followed, or not, by a light tempering to release stresses.

Very large vessels are beyond the capacity of any quenching plant in existence, which limits the possibilities of that process; slower cooling avoids a certain amount of risk always present in quenching; and, finally, as operating temperatures rise, the increased hot strength gained by quenching gradually disappears, beginning at about 750 deg. fahr. (400 deg. cent.), and is gone at about 1050 deg. fahr. (570 deg. cent.) for all but special alloy steels.

If it be a chemical vessel, it is usual to take the tests at this stage, when all heating steps have ended. Tests are generally longitudinal tensile bars, of standard size, 0.505 in. in diameter by 2 in. gage length, which must show the physical properties required by the specifications. Occasionally, transverse bending tests are desired, rectangular bars usually 1 in. by $^1/_2$ in. which are customarily required to bend cold through 180 deg. around a pin of diameter equal to twice the thickness of the



FIG. 31 SHIPPING THE LARGE CYLINDERS

bar without cracking of the convex surface of the bent portion.

The closed-in ends are now machined on the exterior to the extent that may be necessary, the throat is opened to final diameter, and always there is the machining of the closure.

The percentage of the shipped weight to ingot weight in large cylinders depends upon the design and the amount of machining required, varying from about 55 per cent for cylinders rough-forged in bore and exterior to about 35 per cent for those finished all over.

Cylinders may have their ends closed by a variety of methods. A row of studs may be implanted in the end face to attach a head with a joint-making gasket, generally a copper ring, or a flange may be turned up at the end of the cylinder so as to allow fitting of through-bolts to hold the head, which may be of many shapes. Sometimes a screw thread is cut in the interior of the cylinder mouth, either continuous or interrupted, and a corresponding plug fitted which carried an obturating device selected from a number available. The choice of continuous or interrupted thread depends chiefly on the frequency with which it will be necessary to open the cylinder when put in service.

If a full-size or approximately full-size opening into the cylinder be not necessary or desirable, a reducing section can be attached or made integrally by closing in the mouth of the cylinder. Boiler drums of this type may have the ends closed in to take the usual elliptical manhole cover, or, as has been done frequently of late, one or both of the covers may be circular—a much cheaper form. In case both covers are circular it is necessary to finish them completely and build them into a wall in the middle of the drum where they can be protected from oxidation when the drum is heated at each end for closing in, and heated all over for final annealing, after which the seats are finished. Sometimes an end of the closed-in cylinder is sealed by a conical fitting, or the neck may be flanged and provided with a bolted cover fitted over a ground-steel gasket plate.

Any lateral fittings desired are attached, and after a thorough inspection for dimensions a final pressure test is applied to the sealed cylinder, water being pumped into it until the desired test pressure is reached, usually $1^1/_2$ times the working pressure. Search is then made for any leaks that may require tightening of bolts or refitting of gaskets, adjustments being made until all points are watertight. If required, air may be pumped in up to specified pressure and any fall of pressure over a given time be noted. Where this is not necessary it is to be avoided, being a much more costly test than the hydrostatic.

With damageable parts protected by lagging, the finished cylinder is loaded in cradles on one of the special cars now provided by the railroads for this type of heavy service, secured in position by holding-down bolts, and sent out to perform the service for which it was designed and built.

Discussion

P. E. MCKINNEY.[2] This excellent paper is so complete that it is difficult to add anything of interest to what has been presented. The author touches on the use of riveted or welded vessels for certain applications. The production of large pressure vessels from plates formed and welded, both of carbon and alloy steels, is developing at an astounding rate, and it can safely be predicted that within the next few years this type of construction will find ever-widening application.

While the fact remains that fusion welds are essentially cast metal joints, improvements in welding materials and standardization of welding practice have eliminated many of the objectionable characteristics of welding which formerly were considered inherent.

Notwithstanding the broadening of applications for vessels of welded construction, the ever-increasing demand of industries for vessels to operate at higher temperatures and pressures creates a very wide field in which the use of vessels of weldless forged-steel construction is absolutely essential. One can look forward to these more exacting service conditions requiring the development of large alloy-steel forgings with characteristics which are beyond anything that has been contemplated in the past.

The author refers to the importance of steel-making practice in the production of good cylinders, which is a point that cannot be too strongly emphasized. In the melting of steel for these large forgings, refinement must be carried to a point which would not be

[2] Bethlehem Steel Company, Bethlehem, Pa.

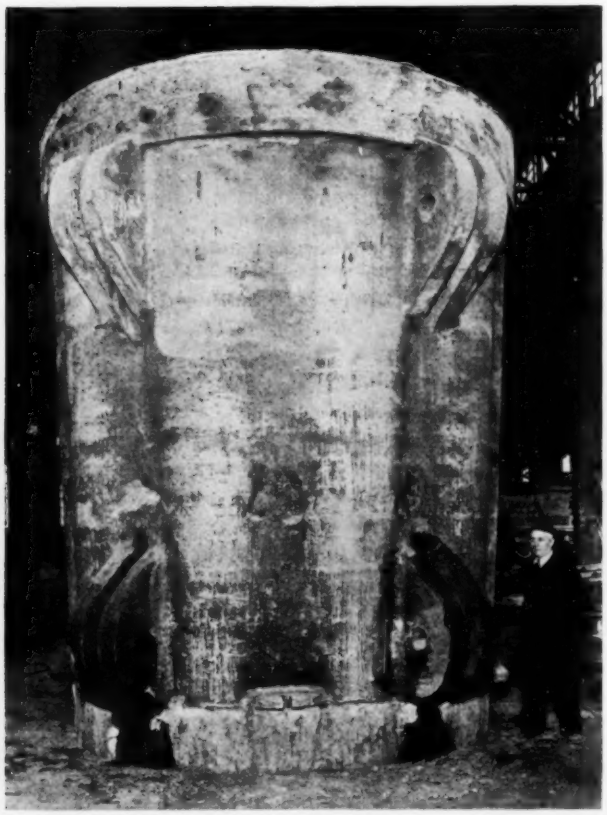


FIG. 32 MOLD FOR 108-IN. INGOT STANDING ON STOOL PLATE

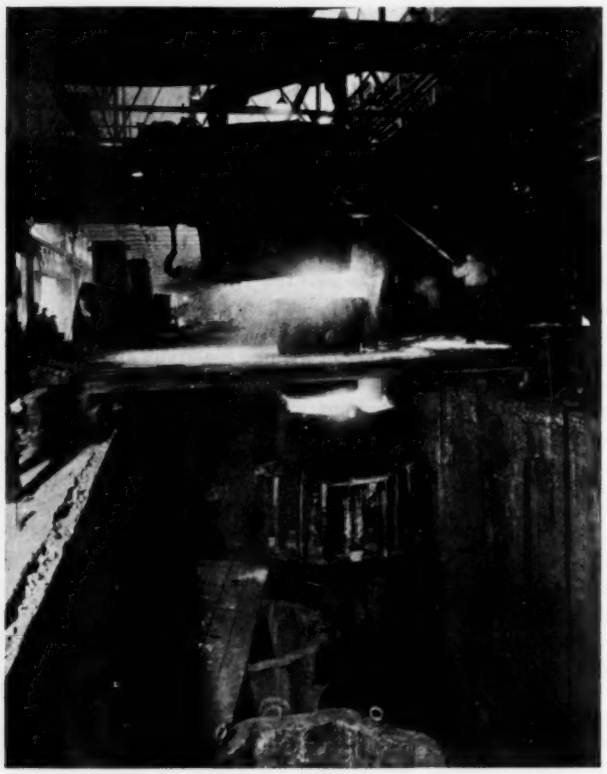


FIG. 33 CASTING 108-IN. INGOT, SHOWING SPECIAL POURING-BOX EQUIPMENT AND SINK-HEAD FLASK PLACED ON INGOT

justifiable or necessary in the production of small or medium size forgings. Large masses of molten metal, slowly solidifying, would react in the mold unless the steel is in a condition approximating equilibrium. Refining and finishing to such a degree that the steel can be left in the furnace for at least an hour



FIG. 34 POURING OF 108-IN. INGOT MOLD, SHOWING THE THREE LADLES IN PLACE, SPECIAL RUNNERS, AND LIFTING EQUIPMENT

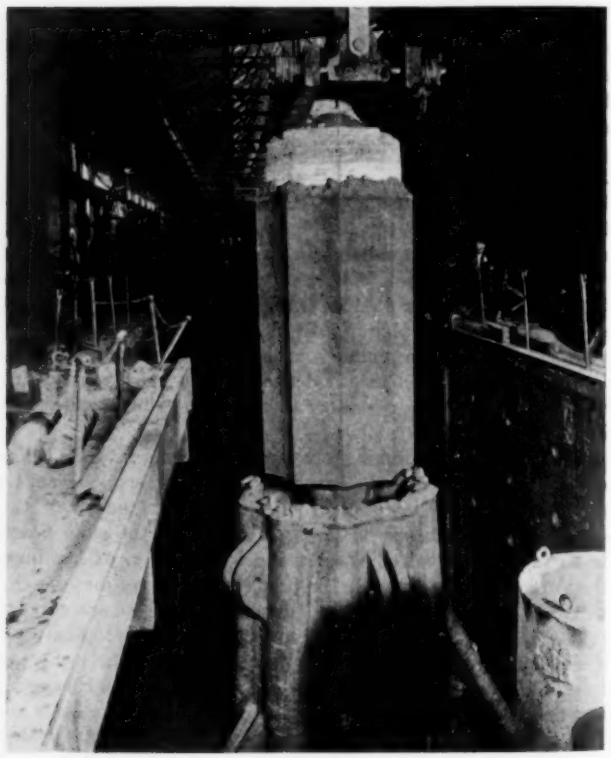


FIG. 35 REMOVING 108-IN. INGOT FROM MOLD; WEIGHT OF INGOT, 427,000 LB.

with practically no action or change in composition, is necessary for the successful production of this type of material. This involves a most careful study of slags, temperatures, and all other conditions surrounding the production of the molten steel.

One outstanding advantage of seamless forged vessels is that during the forging operation the material gets a very thorough proof test as to its soundness and malleability. The forging operations, involving as they do the upsetting, enlarging, piercing, and, finally, drawing down from a wall thickness of approximately 50 in. to approximately $6^1/_2$ in., would uncover any unsound areas in the material, and it is reasonably certain that a mass of metal which will stand these operations without breaking up or showing serious tears is sound and homogeneous throughout.

In the handling of these massive forgings, many extraordinary hazards are encountered, and the entire forging may be spoiled by a mistake in any one of the many operations, such as:

(1) Pouring of the ingot, involving the getting of a number of open-hearth heats in perfect condition for pouring at one time

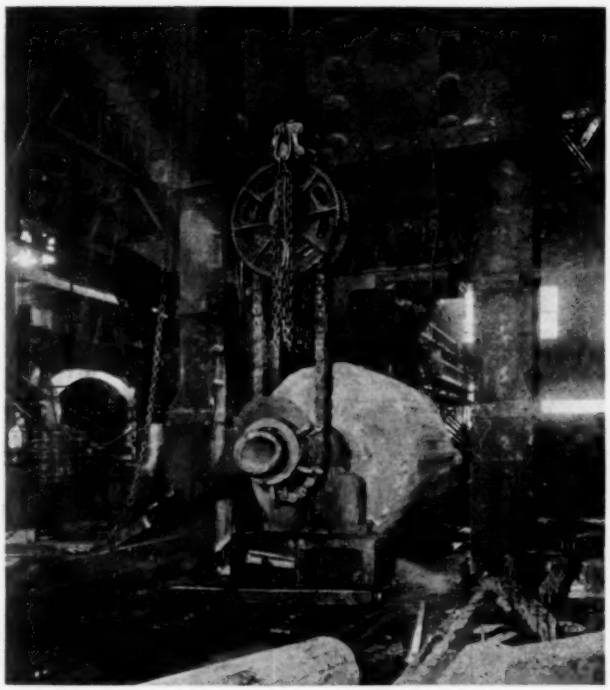


FIG. 36 ENLARGING OPERATION, SHOWING RIGGING FOR REVOLVING THE MANDREL

(2) Any one of the many forging operations involving holding this mass of metal at a forging temperature for from 14 to 16 days
(3) The heat treatment of such a large mass of steel
(4) Machining operations in which the improper drilling of holes or machining of header seats may render the forging unsatisfactory for the purpose intended.

A matter which should be of considerable interest to mechanical engineers is the magnitude of the special equipment, other than furnaces, machinery, and cranes, required in the production of these large forgings.

With the admitted superiority of chilled cast ingots over sand cast ingots, the first problem presented to the steelmaker is the production of a large enough mold to cast these massive ingots. The casting of these large molds requires, in turn, special heavy flask equipment, special clamping devices, patterns, and core-box equipment. The large molds are cast either from molten metal transferred from the blast furnace in heavy mixer ladles or from pig iron remelted in the open-hearth furnace. The mold required for casting a 108-in. corrugated ingot weighs approximately 360,000 lb., the base stool for the mold weighs approximately 86,000 lb., and the sink-head flask weighs approximately 40,000 lb. without the refractory lining. This mold equipment will usually not withstand the pouring of more than 15 ingots without renewal. Innumerable heavy mandrels and special tools are

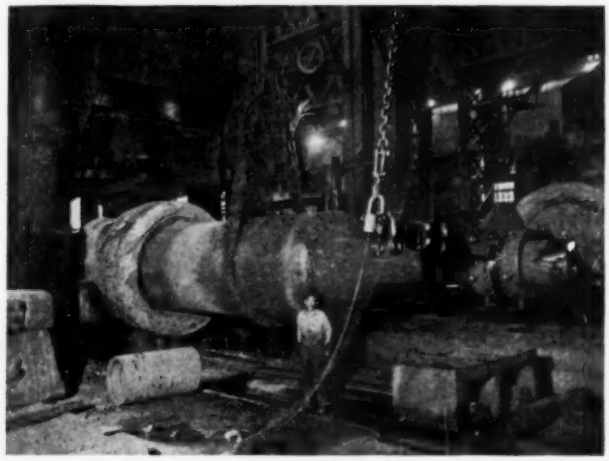


FIG. 37 OPERATION OF DRAWING TO LENGTH ON MANDREL



FIG. 38 ROUGH FORGING, SHOWING LIFTING GEAR



FIG. 39 FORGINGS IN COMPLETED FORM AS TANKS ON HELIUM TANK CAR

required for the various forging operations other than the conventional tool equipment required for hydraulic presses. As an instance, one of the large mandrels required in the production of oil-still forgings is approximately 58 in. diameter by 22 ft. long and weighs 210,000 lb. For this forging, there is required a 91-in. ingot weighing 300,000 lb. In many cases these mandrels will not withstand more than 8 to 10 forging operations.

Notwithstanding the massive sizes of these forgings and the many hazards involved in their production, the steel industry has so surrounded the various operations with safeguards and precautions to assure quality that it is quite apparent that, for the most exacting applications, this type of construction now holds a position which cannot justify substitution for it of any other process.

Continuous Reheating Furnaces for Rolling Mills

By A. L. CULBERTSON,[1] PITTSBURGH, PA.

The subject of this paper is one which is indeed much broader than could be expected from a casual glance, due to the many types of furnaces used for the various rolling-mill operations. The subject matter is confined to those furnaces used for reheating purposes, necessary for serving, rolling, and piercing mills. Even this reduced field leaves it an extremely broad one when considering the types of furnaces required for the different operations.

The author discusses the various classes, using only broad subdivisions of each, and with this in mind a chart is given to show this classification, in order to permit of some common points of comparison, both as to design and operation.



CONTINUOUS reheating furnaces for rolling mills as shown in Fig. 1 are first divided into three general classifications: (1) Wide furnaces, 25 ft. to 60 ft. wide inside, (2) medium-width furnaces, 8 ft. to 25 ft. inside, and (3) narrow furnaces, 4 ft. to 8 ft. wide inside. The majority of furnaces will be found in the medium class. The three divisions as to width have been made so that general comparisons of fuel consumptions and tonnage outputs among a number of furnaces may be confined to the class in which they are given. This is desirable as a means of maintaining accuracy in comparison.

It is known that the wide furnace operates on much less fuel per ton than a medium-width furnace of the same design, and, likewise, a medium-width furnace uses less fuel than a similar narrow furnace. Heating capacity is likewise a function of the width of the furnace.

Wide Furnaces

These are generally used for heating light sections, such as flats or squares with a thickness of material very seldom more than 3 in. and frequently as thin as $1\frac{1}{2}$ in. Inasmuch as the distance is limited through which these light sections may be pushed without having buckling, the length is limited. Should these furnaces be provided with a horizontal hearth, the friction would be greatest and the length a minimum. In an effort to overcome friction and make it possible to lengthen the furnace, a sloping hearth is used with the material being pushed downhill. Due to the lightness of the material, it is not desirable to use underfiring. This class of furnace, therefore, generally employs top-firing only with the material passing through the furnace either on solid billets used as skids embedded in the hearth or being pushed over the refractory hearth without any skids whatsoever.

The use of solid skids in top-fired furnaces is desirable to reduce the wearing away of refractories. Since the steel completely covers the hearth, it prevents the direct action of the flame upon the skids, and a fairly long life is obtained. Solid skids, when used, extend from the charging door of the furnace down to that point where the temperatures become high enough that rapid wearing away of the skids is evidenced. From this point, the usual construction of the hearth calls for a chrome-ore facing which is quite effective in withstanding the abrasive action of the material passing over it.

With the length of the furnace limited, the next method employed in increasing the tonnage output is widening of the furnace that longer billets may be charged.

The tonnage output of any furnace is dependent upon the heat-absorbing surface of the steel exposed to the heat. By widening the furnace, the billets may be lengthened and the heat-absorbing surface increased, thus increasing the tonnage obtained from the furnace. The most common size of furnace in this class employs billets 30 ft. in length. Depending upon the thickness of material, its physical shape, i.e., squareness of corners and straightness, tonnages have been obtained of from 25 to 75 tons per hour, depending upon the length possible to push the billets and the temperatures carried throughout. The wide furnace is usually used in connection with a continuous mill with but one furnace serving the mill. The small section of material heated eliminates the possibility of using it as an end-discharged furnace. These furnaces are, therefore, discharged through the side directly into the first pass of the mill, the material being pushed out by means of pinch roll and pusher bar operated from the side opposite the mill. This type of furnace could be end charged, but to do so would reduce the tonnage output, since the distance the material can be pushed is the limiting factor. As a result, they are charged through the side, employing cantilevered water-cooled rolls within the furnace proper, extending through the back wall through which the pusher also works, or else by means of a skid trough on which the material is pushed in a similar manner to that employed in discharging the furnace.

The fuel consumption of wide furnaces is extremely low per ton of output. This is true whether preheated air for combustion is employed or not, the reason for this low fuel consumption being:

1 Continuity of operation. The furnace is usually being pushed at top speed serving a continuous mill. Yearly figures on operation of a number of these furnaces show an operating time of 90 per cent to 98 per cent of that possible.

2 The furnaces are side charged and side discharged. This

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

method of charging and discharging makes it possible to close up the entire furnace excepting a small opening at the charging and discharging points. This effectively eliminates the influx of cold air usually found at both ends of an end-discharged furnace. Cold-air infiltration not only chills the furnace but likewise chills the flame, eliminating the high temperature radiation heat-transfer possibilities. It also causes oxidation and higher fuel cost. Eliminating these disadvantages results in the saving of from 30 to 50 lb. of coal or its equivalent per ton of steel heated. Unfortunately, all furnaces cannot be constructed of side-discharge design, due to the operation requiring more than one furnace.

3 A wide furnace makes possible a reduced fuel cost by its very width. In comparing two furnaces heating 15-ft. billets with one furnace heating 30-ft. billets, it is readily seen that two side walls and several feet of hearth and roof together with their radiation losses, have been eliminated. In other words, the wider the furnace the less the fuel consumption due to the reduced amount of radiating surface which loses heat to the atmosphere. This results in a saving of from 10 to 30 lb. of coal per ton according to the width of the furnace.

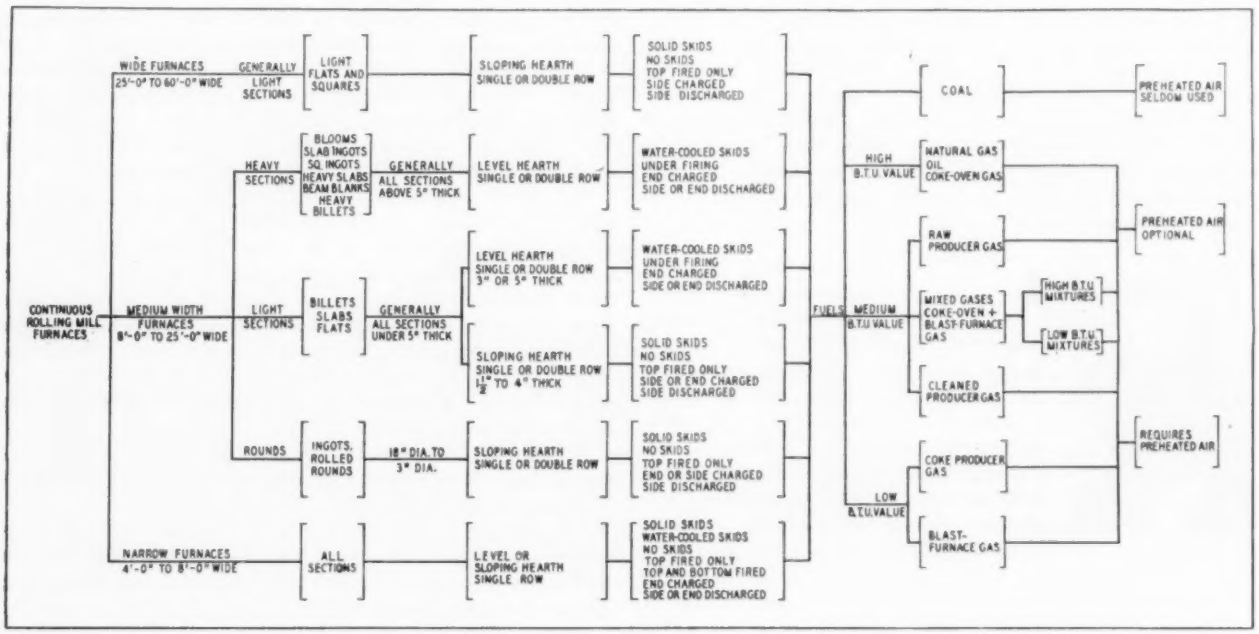


FIG. 1 CLASSIFICATION OF ROLLING-MILL FURNACES

MEDIUM-WIDTH FURNACES

As will be seen on the chart, this class of furnaces is used for more purposes than either the wide or narrow classes. Medium-width furnaces are subdivided into three classifications, those heating (1) heavy sections, (2) light sections, and (3) rounds.

Discussing them in the above order, it is found that the heavy section classification contains furnaces that heat blooms, slab ingots, square ingots, heavy slabs, beam blanks, and heavy billets. The material to be heated in this classification is 5 in. in thickness and over. Continuous furnaces operate successfully heating blooms 13 in. square; slab ingots 12 in. by 30 in. by 48 in; square ingots up to 18 in.; heavy slabs as used for plate mills and strip sheet mills; large beam blanks; and heavy billets for any general use.

In general, furnaces heating these heavy sections require "underheating" as well as "topheating." This necessitates the use of watercooled skid pipes in order that underfiring may be accomplished. Ordinarily, the hearth is level and the furnace is provided with a refractory hearth at the discharge end in order to eliminate the effect of the watercooled skid pipes and permit equalization of temperature before discharging. Occasionally, no hearth is used and some form of skid shoe is used on top of the watercooled skids to raise the material far enough from the skid to likewise eliminate the cooling effect. Both of these methods are in use and the object of their use is identical; viz., to make it possible that the material being discharged leave the furnace at an uniform temperature throughout with no objectionable black spots.

Heavy sections make it possible to increase the pushing length. This type furnace is, therefore, end charged. Either side or end discharging may be employed, although end discharging is the more common. When furnaces are side discharged, it is usual to employ a discharging machine which may be equipped either with tongs or peel. This machine makes it possible to space the heavy pieces on the refractory hearth, if desired, and possibly turn them over for a further equalization of temperature if the design of furnace makes it essential. It is desirable to lift the material before discharging in order to reduce hearth maintenance when side discharging.

Either single or double row may be charged. The average size for this type of furnace approximates 15 to 20 ft. in width and 50 to 60 ft. in length.

Due to the heavy sections, the use of watercooled skids is required. This, of necessity, increases the fuel consumption since the cooling water will usually take away in heat units the B.t.u. equivalent of from 30 to 50 lb. of coal per ton.

It has been found that end discharging saves an appreciable amount of labor and equipment. It is also a saver of space in rolling mills. End discharging makes it possible to serve a mill with a number of furnaces spaced closely together since the necessity of a manipulating machine for discharging is eliminated. This feature makes it possible to keep the size of the furnaces within reasonable constructional limits and vary the tonnage of the mill by the number of furnaces being used. Often a mill is called upon to roll a variety of sizes and with its capacity being mainly determined as to the number of pieces per hour that it can handle, the number of furnaces employed to serve it will vary in accordance with the size of the material being heated, so that a mill may be kept busy as much of the time as is possible.

End discharging saves in labor and space, but in itself usually requires more fuel than the side-discharge type of furnace. Referring to the old standard type of end-discharge furnace equipped for firing above the steel only at the discharge point, this difference in fuel consumption amounts to 40 to 50 lb. of coal per ton over the old side-discharge type. This is due partly to the fact that the discharge door is usually of the flapper type which does not make a tight closure. Since gravity discharge is employed, a natural chimney effect is found at the discharge slope and causes infiltration of a large quantity of cold air. This cold air not only causes oxidation but chills the material in the furnace at its point of discharge. It furthermore chills the flame and reduces the rate of heat transfer which is so beneficial if properly applied.

Rounds

The furnaces used for heating rounds for piercing mills usually employ a sloping hearth of varying degree in accordance with the size of material to be heated, the method of working the material through the furnace, and its nature. The usual slope employed is 1 in. to $1^1/_2$ in. per foot. Rounds are heated from 3 in. to 18 in. in diameter, the larger sizes being in the form of cast ingots, with the smaller sizes being rolled and hot sawed. The smaller size rounds are usually rolled through the furnace by hand with a large number of doors either rolling type or lift type, being available so that the operators may reach the entire hearth.

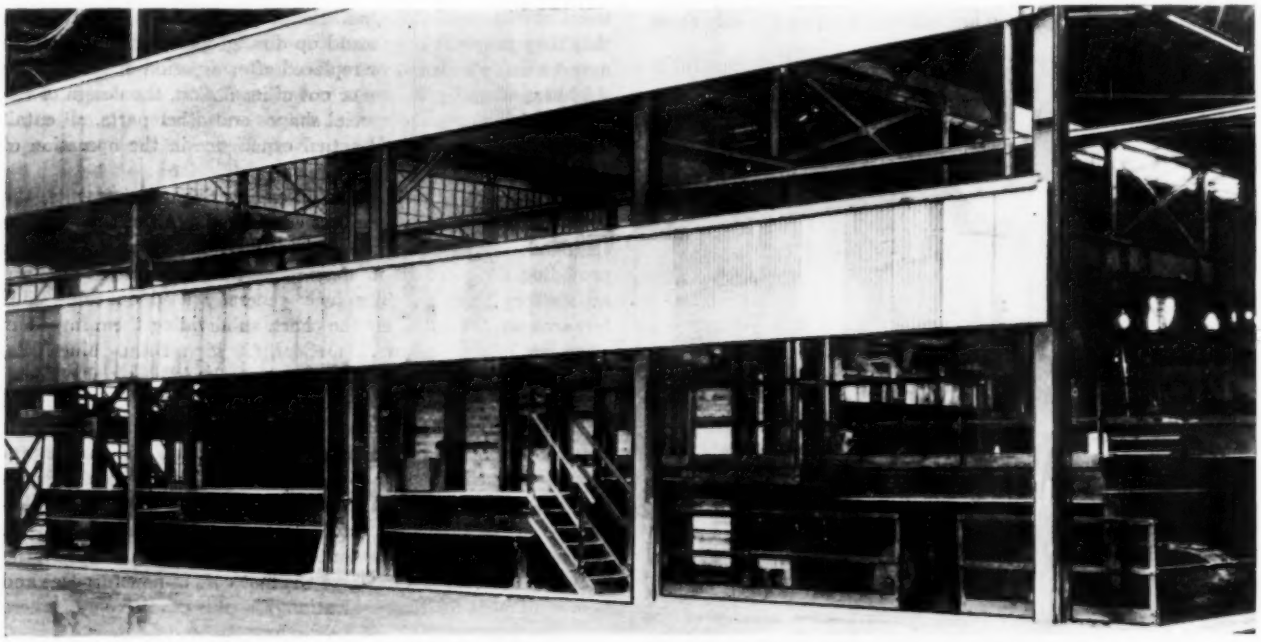


Fig. 2 Three Furnaces at the Republic Steel Corporation, Youngstown, Ohio, Using Producer Gas or Coke-Oven Gas

Light Sections

Under this classification come those furnaces of medium width which are confined to heating sections up to 5 in. in thickness, generally billets, thin slabs, and flats. A level hearth is usually employed for the slabs and billets from 3 in. to 5 in. thickness with a sloping hearth employed for the thinner sections, for the same reasons as mentioned under the wide furnaces above. Those furnaces with a level hearth usually employ watercooled skids and should be underfired. They are end charged and may be either side or end discharged. If single rows are used and the tonnage of the mill is not excessive, side discharging is generally employed. When double rows are employed and the capacity of the mill requires more than one furnace, end discharging is usual.

For the sloping-hearth type, solid skids or no skids at all are the custom, with topfiring only employed, since underfiring is undesirable with the thinner sections. Either side or end-charging may be used, side discharging, however, being the usual practice.

As in the case with furnaces for heavy sections, watercooled skid pipes and end discharging requires additional fuel, but less labor, whereas the side-discharged furnace with sloping hearth and no watercooled skids is economical from a fuel standpoint but higher in labor.

For the larger size rounds, a manipulating machine usually works the rounds through the furnace, thus saving labor.

Furnaces heating rounds are usually side discharged. This is desirable in order to maintain the discharging end of the furnace as nearly uniform in temperature as possible, since properly to pierce a billet concentrically, necessitates absolute uniformity of heating throughout the material. Even with side discharging, this class of furnace has a high fuel consumption due mainly to the low temperatures carried, the amount of hearth which is not covered, and last, but quite important, the number of openings required in the furnace side walls for rolling the material through the furnace. Very seldom does the furnace operator take care that these doors be closed when not being used. As a result the general chilling of the furnace by infiltrated air increases the fuel consumption. Figures of 200 to 250 lb. of coal per ton are the rule rather than exception, and often it has been known to go as high as 350 to 400 lb. of coal.

Narrow Furnaces

Under this classification come furnaces 4 to 8 ft. in width. They are used to heat all kinds of sections; rounds, flats, billets, blooms, etc. This type furnace may have level or sloping hearth, but usually only takes one row. The method of charging is determined the same as the medium-width furnaces, as likewise are the skid requirements and the methods of firing. This type of furnace is usually used for small mills where production is not

high or where a variety of products rule, in which case a number of the single-row furnaces are used to separate orders.

When serving a piercing mill with the larger sizes, a single row is desirable from the manipulating-machine standpoint. Furnaces 130 ft. in length have been constructed, although the usual size would be from 50 to 70 ft.

Combustion Engineering and Fuel Requirements

In the preceding description of wide, medium, and narrow furnaces, only general characteristics have been described. No reference was made to the fuel to be employed nor was reference made to preheated air. The reason for this is that mill requirements and the material to be heated should determine only those

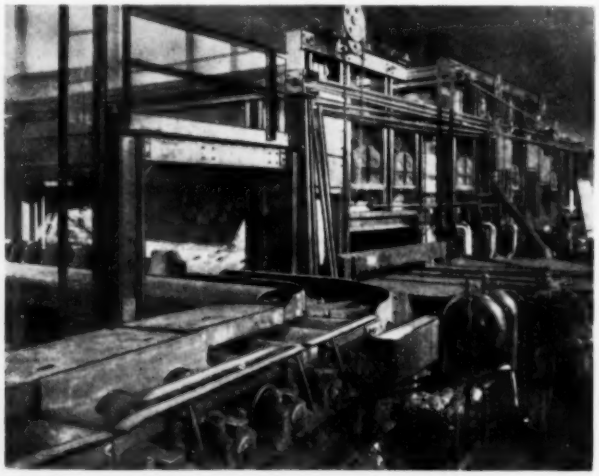


Fig. 3 Furnace at the Washburn Wire Company Fired With Fuel Oil and Using Cold Air for Combustion

characteristics already described, which consist mainly of the general shape of the furnace, the method of charging and discharging, the tonnage required, space available, and the shape of the hearth. This much of the furnace can be determined with no reference to the combustion-engineering side of the question. From this point on, however, the combustion engineer enters the picture and his part is an extremely important one. It might be well at this point to express generally a thought regarding the proper design of the furnace from the furnace-builders' standpoint. In the first place, it is realized that once the shape of the furnace and general characteristics are determined in accordance with mill requirements, the remaining problem is much the greater one. The true combustion engineer looks at a furnace primarily without reference to anything excepting the holes or voids therein.

The holes in a furnace consist of ports, combustion chamber, channels, downtakes, flues, doors, etc. The areas and shapes of these holes are the most important part of a furnace and determine first of all whether the furnace will work or not, and in the second place, they determine how well it will work, both as to quality of heating and as to fuel efficiency. The combustion engineer determines the holes of the furnace and his ability is measured, not entirely with the amount of theory employed, but by the proper blending together of theory and actual operating practice in such a manner as to give best results for the problem at hand. How well he has done his work is never known until the furnace has been put into operation. If he has analyzed the problem from all the standpoints that should be taken into consideration, he has a good furnace. If some have been overlooked, it is desirable that these may be corrected with little trouble. If he has overlooked too many, the furnace is considered a failure. This is the case no matter how well the furnace has been built. It is for this reason that the holes of a furnace are looked upon as of major importance.

After the combustion engineer has determined the holes, then once more a highly important part of the furnace design has to be carried through. This entails the proper application of brick and steel work to surround the holes in such a manner as to make a furnace which will stay in operation once it has been constructed. This part of the furnace designing requires likewise a mixture of theory and practice with the proper consideration given for the temperatures carried throughout the furnace, the proper material to withstand them, and the proper method of construction that they may not only stand up during operation, but likewise may be easily repaired or replaced after a period of time. The thickness of walls, the use or not of insulation, the design of the watercooled parts, the special shapes and other parts, all entail careful consideration and actual experience in the operation of furnaces.

In considering the combustion-engineering requirements, the fuels that may be used are first considered. All of the furnaces classified in Fig. 1 operate satisfactorily on a variety of fuels providing the combustion engineering has been carried out in a satisfactory manner. The fuels generally used on continuous furnaces are tabulated on the chart, subdividing them into four classifications as follows: (1) Coal, (2) high B.t.u.-value fuels, (3) medium B.t.u.-value fuels, and (4) low B.t.u.-value fuels.

First, there is coal. It is classified by itself and may be used either by direct firing or by using stokers, although each year finds fewer continuous furnaces using this fuel. Seldom is preheated air used when coal is direct fired. This type of firing is being eliminated due to the excessive labor costs entailed which require so many points for distribution of coal and the subsequent removal of ashes, with constant attention being required from the heater. In small plants where the first cost of new furnaces and new equipment for the gasification of coal is excessive and where



Fig. 4 Double-Fired Furnace at the Steel Company of Canada Fired With Low B.T.U.-Value Mixture of Coke-Oven and Blast-Furnace Gas

no other fuel is available, coal-fired furnaces may find their place.

Second, under high B.t.u.-value fuels are natural gas, oil, and coke-oven gas.

Third, under medium B.t.u.-value fuels, are raw-producer gas, mixed gases, such as coke-oven and blast-furnace gas, and cleaned producer gas.

Fourth, low B.t.u.-value fuels contain coke producer gas and blast-furnace gas.

The reason for listing these fuels in this manner is to get a subdivision and classification, so that consideration may be given to the various groups with regard to the use of preheated air. With

coal, preheated air is seldom used. If it is used at all, it is applied above the fire bed and the fire bed is being carried in a condition approaching a semigas producer effect. Application of air beneath the grates would result in clinkering.

Preheated air is optional with natural gas, oil, coke-oven gas, raw-producer gas, or the high B.t.u. mixtures of coke-oven and blast-furnace gas. Preheated air is essential when using low B.t.u. mixtures of coke-oven and blast-furnace gas, cleaned producer gas, coke-producer gas, or blast-furnace gas.

With the higher B.t.u. fuels consideration must be given to a number of things to determine whether preheated air should be used or not. Of primary importance is the cost of the fuel delivered to the furnace per million B.t.u. In addition, consideration must be given to plant conditions, available depth, available space for the preheating equipment as well as available space for the furnace. Should a limited space only be available for the furnace, it is desirable to use preheated air to increase the temperatures and, accordingly, the tonnage output. It all sets itself up into an analysis of the existing conditions and the return which will be obtained on the added investment covering equipment to preheat the air. In recent years, a most important step has been taken in regard to the use of preheated air, and that is that the tendency is more pronounced to consider the air-preheating equipment as a separate consideration, which should stand on its own merits and be used or not in accordance with the investment value, rather than to be considered as an essential part of a furnace. This is the obvious viewpoint and the type and general design of a furnace should be decided upon which will provide maximum efficiency without regard to type of air heater to be used, and after the type of furnace is decided upon, then the preheating equipment can be properly applied whether refractory-tile recuperator, metal recuperator, or regenerative air heater, according to which one is best suited for the particular needs of the definite installation at hand. Often, the make or type may be optional. In other cases, definite types will be required.

Under the above classification of fuels and the preheated-air requirements, it will be noted that the division is made on the basis that preheated air is optional only with those fuels whose flame temperature when burned with cold air is sufficiently high to enable the proper furnace conditions to be easily obtained. The lower B.t.u.-value fuels, in some cases, will give a flame temperature which is possible to be used in continuous furnaces. As a rule they are too slow while using cold air to be a desirable fuel for satisfactory operation. The preheated-air requirements in this case are merely to obtain a B.t.u. input in the combustible mixture sufficiently high to give the desirable flame temperature and flexibility needed for proper operation in continuous-furnace work.

Continuous-Furnace Operating Requirements

After the shape and other characteristics of the furnace as described have been determined and, in addition, the fuel and its preheated-air requirements are known, the design of the furnace should be considered from the operating standpoint. In this regard, the furnace should be designed to perform the following functions which are listed in the order of their importance:

1 Uniform heat distribution throughout the mass of the discharged product at a predetermined temperature.

2 Flexibility as to tonnage output without any appreciable drop in efficiency or change in quality of heating.

3 Low oxidation losses.

4 Low furnace maintenance.

5 Low fuel consumption.

6 Ease of control.

Uniformity of Heating

The function of a continuous furnace which is of paramount importance is that heading the above list which demands "uniform heat distribution throughout the mass of the discharged product."

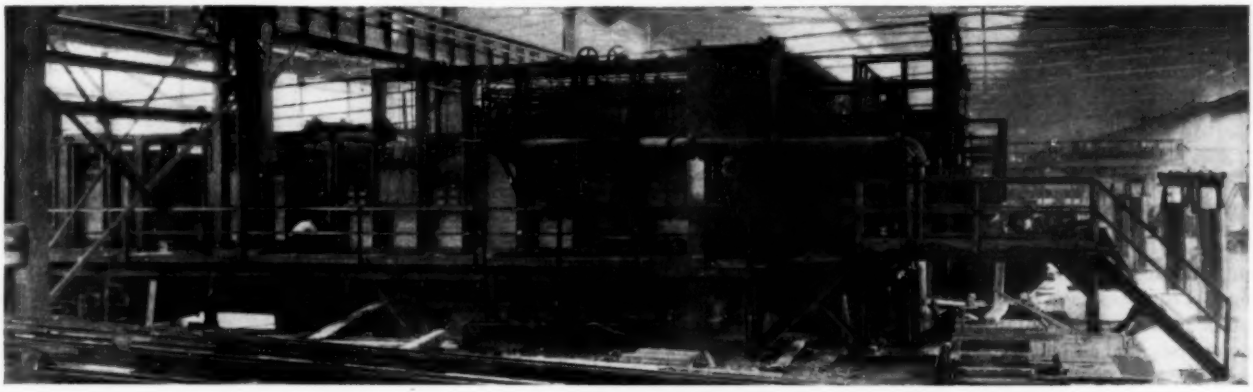


Fig. 5 Two Triple-Fired Furnaces at the Timken Roller Bearing Company Fired With Natural Gas

This may be spoken of, as the "quality of heating." The mill operator knows how important it is to have quality heating if the mill is continually to roll quality product with low delay periods and maintenance cost. To the steel plant executive it also means much. Low maintenance means lower production costs. Quality product finds a ready market.

Fundamentally, the way to obtain uniform heat distribution is the same in all furnaces; viz., first, that a uniform temperature be carried in the furnace at the discharge end, which temperature should be but slightly above that of the product to be discharged; second, this temperature condition should be maintained over a sufficient length of hearth to provide sufficient time for the material which is passing over the hearth to become thoroughly soaked and equalize its temperature throughout.

In other words, it requires the proper temperature and the proper time.

In the usual form of continuous furnace which has been used in the steel plants for many years, the furnace is fired above the steel at the discharge end. It may have underfiring or not, in accordance with the size of material to be heated, but the most of the fuel is fired at the discharge end immediately above the material waiting to be discharged. This is true in split flame-fired furnaces, the solid-hearth type of furnaces, and box-roof furnaces, in both end-discharged and side-discharged types. The main requirement of uniform heating necessarily imposes on all these furnaces the condition that the temperature carried be not much above that wanted in the steel at the point of discharge.

Many furnaces have been built which cannot fulfil this requirement, and as a result the material is melted, washed, ununiformly heated, and hearth maintenance is made excessive.

This temperature condition may be obtained more or less satisfactorily in end-fired furnaces as follows: (*a*) By using cold air for combustion, (*b*) by imperfect mixing of fuel and preheated air, causing delayed combustion, and (*c*) by the use of moderately heated air and semi-mixing. This is in effect a combination of (*a*) and (*b*). A fourth method and a positive one which has been developed in the last few years, is: (*d*) By constructing the soaking-hearth portion of the furnace as a separately fired independent unit and supplying only sufficient fuel to this chamber to compensate for the radiation losses of the chamber itself.

As already mentioned, it is exceedingly important that billets heated for piercing-mill use be heated uniformly throughout for

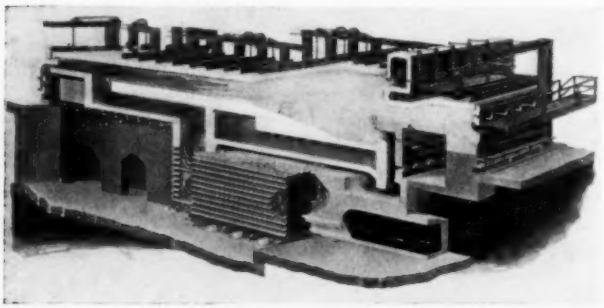


FIG. 6 ANOTHER FURNACE AT THE TIMKEN ROLLER BEARING COMPANY

proper piercing. It is worthy of note that for many years piercing-mill furnaces have been constructed in the main without the use of preheated air. Many companies would not consider the use of preheated air on this type furnace and, until recently, have been exceeding cautious even with regard to the kind of fuel they used. The reasons, although not always expressed are obvious. With end-fired furnaces, preheated air raises the temperature at the discharging point above that desirable in the steel, excepting with the lower B.t.u.-value fuels. Even with cold air some of the more rapid burning high B.t.u. fuels will give an excessive flame temperature unless carefully watched. The operators realized that the conditions were wrong for proper heating when using high B.t.u. fuels and preheated air, particularly if high flame temperatures, which are available, were permitted to develop. End-fired furnaces used for other types of mills frequently permit a greater range of temperature differential to be present in the discharged product, than is possible with piercing mills. It was for this reason that frequently recuperators and air heaters were indiscriminately applied to continuous furnaces which, in some cases, gave satisfactory operation while in others caused failures until revisions were made in the method of firing, which as shown in (*a*), (*b*), and (*c*) preceding, overcame the objections of hot air for combustion in the end-fired furnaces and minimized its excessive flame temperature.

It was learned rather quickly that highly preheated air caused overheating at the discharge end of the furnace. Many recuperator manufacturers then argued that 500 deg. to 600 deg. air was as high as was permissible and built the recuperator accordingly. Many people were against recuperators of any kind due to the failure they may have had from too high a temperature, excessive hearth maintenance, and poorly heated steel.

In looking once more at the methods listed for obtaining a satisfactory temperature surrounding the steel at the discharge point of the furnace, it is felt that the various means should be expanded somewhat. (*a*) Use of cold air for combustion results in high fuel cost, since the entire furnace temperature is low and no heat salvage is employed; (*b*) results in difficult control of the hot spot or point of thorough mixing, wherever it may be; fluctuating temperature conditions, high fuel consumption due to unburned fuel passing through the furnace in stratified layers, heavy scale losses due to free air, and low average temperature throughout the furnace which lessens the output; (*c*) gives more control and better efficiency than (*b*) but reduces the output and is uneconomical due to low preheat in the air. Each of the preceding methods are being used and are giving fairly good results, but to combustion engineers, high furnace efficiency and good quality of heating should go side by side, and the furnace which is obtaining one at the expense of the other is not a properly designed furnace.

Under the method explained in (*d*) in which the soaking hearth is a separately fired independent furnace unit, it is possible to obtain quality heating and at the same time use as highly preheated air for combustion as can be obtained, thus making the furnace an extremely efficient one as well as having one which is delivering properly heated steel.

Before going further into the desirable features of a continuous furnace, it is felt that it would be instructive to point out the analogy between the heating operation of a continuous furnace

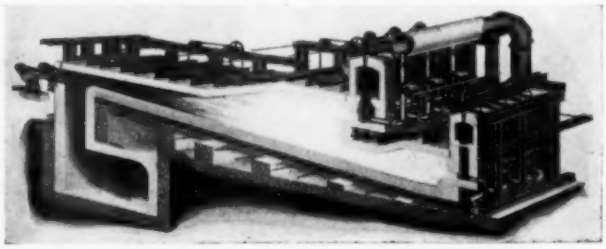


FIG. 7 DOUBLE-FIRED FURNACE AT THE WEST LEECHBURG STEEL COMPANY FIRED WITH PRODUCER GAS

and that occurring in the soaking pit. At first soaking pits were used, as their name implies, to perform only the operation of soaking. The hot ingot was charged while still containing molten metal and sufficient heat present to equalize the ingot at a temperature satisfactory for rolling. No additional fuel was supplied and a true soak was obtained. As production increased, the ingots arrived at the soaking pits too long after pouring to contain the required amount of heat and many times they were charged cold. The form of the soaking pit gradually changed until it was made a heating furnace as well as a soaking pit. Fuel-saving features were added, such as regenerators and recuperators. Still, the soaking operation was exactly the same as that in the original soaking pit; it still was a necessary part of the heating cycle; i.e., the soaking period is essential in order that the temperature of the entire mass may be equalized throughout before it is rolled. Modern pits first have a period of heating and end up as originally, with a period for soaking. During the heating period while the ingots are cold, it is desirable to apply a maximum flame temperature, and a high fuel rate is used in order to obtain rapid heating and, accordingly, higher production. In this period of the cycle, a high B.t.u. fuel and highly preheated air are desirable. As the ingots approach rolling temperature, a high flame temperature is not so desirable. The heater then may do one of several things. He may shut his pit entirely down and obtain preliminary soaking to equalize the ingots and prevent overheating, or he may reduce his firing rate greatly, or reduce the flame temperature of the combustible mixture for the same reason. Common practice calls for the heater to use an excess of gas. This not only lowers the flame temperature but provides a reduc-

ing atmosphere and reduces the oxidation losses. Such a practice is wasteful of fuel but is essential unless other means are provided to obtain the same results. If a pit had two fuels available, such as natural gas and blast-furnace gas, such a combination would permit obtaining ideal heating conditions. The natural gas, with highly preheated air, would give a maximum temperature during the heating portion of the cycle. When the ingots approached a finishing condition if the natural gas could be turned off and blast-furnace gas turned on, a wonderful finishing condition would be found in the soaking pit and maximum production and minimum time of heating would be required. The important points in the analogy of soaking-pit heating and continuous-furnace heating are that a rapid heating rate may be used until the steel approaches rolling temperature; from that point on the heating condition should be carried out in an atmosphere which is protective to the steel and still has a temperature which is but slightly above that required for rolling. The soaking-pit operator has to obtain these varying conditions throughout the heating cycle in a single chamber by the manipulation of his valves and dampers. These same conditions may be obtained in a continuous furnace by having the material pass through the furnace at a speed which will pass it through the various portions of the furnace so that it will be kept the proper amount of time in each section, with time allowed for a suitable period of equalization.

Any end-fired furnace should have its maximum temperature but slightly above that desired in the steel if it is to have a soaking condition at all, and a period of soaking is considered as important for quality heating in continuous furnace operation as it is necessary for soaking-pit use.

The analogy between pit-furnace operation and that desired in continuous-furnace quality heating is responsible for the actual design and construction of the Rust Zone-Controlled Double and Triple Fired furnaces. These furnaces are all provided with a soaking chamber and refractory hearth where uniform heat distribution throughout the mass of the product being heated, may be obtained at any predetermined temperature. This soaking chamber is supplied with only sufficient fuel to provide for the radiation losses from the chamber itself. No heating is attempted nor desired. Approximately 15 per cent of the total fuel is introduced into this chamber and this chamber is responsible only for the equalizing of the heat throughout the material passing through it before discharging, and is maintained at a temperature which is that desired in the product to be rolled. The fuel rate being only that required for supplying radiation losses, is a constant one. The remainder of the fuel, or 85 per cent of the total, is introduced into the heating chamber, either partially above and below the steel to be heated, or in the case of the double-fired furnace, all of it above. The fuel rate in the heating chamber is not a constant one. Since the heating chamber is responsible only for transferring heat to the steel at the rate desired by the mill, it fluctuates up and down in accordance with the mill requirements.

Flexibility

"Flexibility" listed as the second requirement for a good continuous furnace is obtained by varying the fuel rate of the heating-chamber burners. It is evident that this firing rate will, therefore, fluctuate with the mill and, in fact, keep step with it.

When considering end-fired furnaces from a flexibility standpoint, if they are designed correctly for a given tonnage, the firing ports will be correct for the firing rate desirable for the predetermined tonnage output. The flame characteristics will likewise be predetermined for a definite fuel rate and tonnage. Any changes from this fuel rate will result in a change in the flame characteristics and the resultant heating conditions in the furnace which are undesirable. Pushing an end-fired furnace beyond its rated capacity requires an increase in the average temperature throughout its length that heat may be transmitted to the steel more rapidly. Higher tonnage and increased furnace temperature requires increased firing rate. With the same rate of mixing, more heat will be liberated at the discharge point of the furnace than previously. This results in an increased temperature where it is undesirable. Overheating frequently occurs with its resultant cindering, slagging, and increased maintenance for the hearth and refractories as a result. At lowered tonnages than that for which the furnace is designed, a decreased fuel rate again changes conditions which are objectionable in the discharged material. In all cases, the slightest ununiformity in firing or improper adjustment of burners is immediately reflected in the discharged steel, since the burners at the discharge end have a double duty to perform: first, the heating of the material in the charging end of the furnace and, second, the proper conditioning of the material about to be discharged at the opposite end of the furnace. These two operations are completely isolated in the Rust zone-controlled continuous furnace; each being independent makes it possible to have a much more flexible furnace.

Low Oxidation Losses

The third requirement for a continuous furnace is listed as low oxidation losses. It is to be noted that the first two requirements have to do with actual furnace operation with reference to the mill. The third, low oxidation losses, takes its place in importance due to the actual amount of money that may be saved by a furnace having this requirement. This item is one too often overlooked and proper consideration has not been given it excepting in special cases where oxidation affects the resultant product as to quality. Oxidation is undesirable not only from a decarbonization standpoint, but, in addition, due to the large amount of money lost in scaling which occurs in the furnace that could be reduced on all kinds of steel. It is not uncommon for a mill to roll 50 tons per hour. On the basis of 20 hr. per day and 300 days per year, this amounts to 300,000 tons yearly. A scale loss of 1 per cent means 3000 tons which, at $40 per ton, amounts to $120,000 yearly loss. Many furnaces have a scale loss of from 2 per cent to 4 per cent. A furnace properly designed should not have oxidation losses of more than 1 per cent to 1½ per cent.

Oxidation in the furnace is dependent upon two things: first, reaction of iron with the oxidizing atmosphere whether it be free oxygen due to excess air, or its reaction with water vapor formed in the combustion of hydrogen containing fuels. The second important part of oxidation is the time the steel takes in passing through the furnace. It is well known that oxidation is a function of the time material is exposed to an oxidizing atmosphere, and since water vapor is always present in the furnace atmosphere using hydrogen containing fuels, this is an important cause of scale losses.

The zone-controlled double- and triple-fired furnaces make possible a minimum of oxidation losses. While the steel is passing through the soaking chamber, it is contained in a reducing atmosphere. The rate of heating in passing through the heating chamber is exceedingly high, and, therefore, the time the material is hot is kept to a minimum all of which tends to reduce oxidation. During mill delays there is no necessity of protecting the steel at the discharge point from overheating, since the temperature carried in the soaking chamber is approximately the rolling temperature desired. It is, therefore, unnecessary for the heater to check the air and lengthen the flame to protect this steel. It is only necessary to reduce the firing rate in the heating chamber. In this manner, cold steel at the charging end of the furnace is maintained in its original condition and is not heated up during mill delays. This also reduces scaling.

Low Furnace Maintenance

Any furnace which has high flame temperatures where the temperature cannot be dissipated toward doing useful work will have high maintenance on the refractories. This is found in the end-fired furnace when excessive temperatures are used at the discharge point. The steel is hottest just before being discharged and its absorbing power is least. This puts the load on the furnace refractories which become overheated and cannot give up the heat by radiation to the steel rapidly enough to afford its own protection. The zone-controlled furnace has its highest temperatures developed in the heating chamber which is of sufficient size to permit the heat to radiate rapidly to the cold steel entering from the charging end. The highest temperatures are not where the steel is hottest, but are where the steel is relatively cold, and the ability of the steel to absorb heat rapidly due to the high differential of temperatures protects the refractories and reduces furnace maintenance.

The elimination of cindering, washing, and melting of steel in a continuous furnace reduces furnace maintenance to a minimum, particularly with regard to the soaking hearth where the temperature prevents cindering and the chrome-ore facing is provided to withstand the abrasion of the material passing over it.

Low Fuel Consumption

The fifth requirement for proper continuous furnace design calls for low fuel consumption. End-fired furnaces cannot use highly preheated air and gain the benefit of the high flame temperature which is possible of development. The zone-controlled furnace, on the other hand, welcomes the highest preheat that can be obtained and uses the preheated air in the heating chamber where the steel will rapidly absorb the heat given up from the resultant high-temperature flame with the corresponding beneficial effect on increased efficiency.

Since radiation is a function of the difference of the fourth powers of the absolute temperatures of the material being heated and the radiating body, a few simple calculations will effectively show the desirability of using the highest flame temperatures which can be obtained, with the subsequent benefit of reduced fuel consumption and increased efficiency.

Any continuous furnace will show a reduced fuel consumption per ton as the tonnage output increases. The zone-controlled furnace can be pushed beyond its rated capacity as far as 50 per cent. Since the heating operation is separate from the quality of heating contained in the discharged product, the heating-chamber firing rate may be speeded up with a corresponding increase in downtake temperatures which is required when the tonnage is increased. In spite of the increased downtake temperatures, the fuel consumption per ton is reduced due to the higher capacity output. This overload capacity feature is one entirely absent in the end-fired furnace, although some additional capacity is frequently obtained from this type furnace, the amount of which, however, is entirely dependent upon how poor a quality of heating the rolling operation will put up with. In actual operation the zone-controlled furnace of the same length as an end-fired furnace, will surpass the end-fired furnace at least 25 per cent in capacity and still give the same quality of heating as at normal rating. When it is considered that the heating is done in the zone-controlled furnace only in the heating chamber, it is readily seen that this high rate of heat transfer is an important one since the zone-controlled furnace is, in effect, from 10 ft. to 15 ft. shorter than the end-fired furnace, although the overall length of each may be the same.

Ease of Control

The sixth requirement of a properly designed continuous furnace calls for ease of control. It is felt that great progress has been made in the past few years with regard to control features on these furnaces. At the present time, it is possible to install automatic temperature control in order to maintain a constant temperature condition in the discharge end of the furnace. Gas and air proportioning controls can be satisfactorily employed. By a combination of the two working in conjunction with an automatic draft regulator, the heater's work is made quite easy. The zone-controlled furnace, to be completely equipped, should have an automatic temperature control in the soaking chamber, together with proportional control equipment. The draft regulator would maintain a constant pressure condition in the furnace. The heating-chamber burners should be equipped with proportional control. Thus the heater would have but one valve to regulate, that controlling the gas flow to the heating chamber burners. This he would operate in accordance with the mill demand.

A combination of heating and soaking chamber eliminates to a large extent the requirement for delicate adjustment of firing rates and fine adjustments of burner equipment which is extremely difficult with high-capacity burners. It does this by providing a period of time for the steel to obtain the proper heating conditions after it leaves the heating chamber while passing over the soaking hearth.

An important feature of the zone-controlled furnace is the restriction which is provided by the suspended-roof construction between the heating and soaking chambers. This restriction accomplishes two beneficial results. In the first place, it effectively prevents heat from being radiated into the soaking chamber from the high-temperature heating chamber, by effectively shielding the material on the hearth. Without this restriction, the soaking operation could not be fully accomplished. The second beneficial result is obtained by throttling the flow of waste gases from the soaking chamber. In so doing, a positive pressure is maintained in the soaking chamber which prevents the infiltration of air up the discharge slope. Not only does the restriction prevent infiltration, but it likewise causes the soaking chamber to become independent of the pressure conditions carried in the heating chamber at different rates of firing. The height of the restricted roof portion is calculated to be that correct for the fuel rate of the soaking chamber which, by its constancy, makes unnecessary any further control.

Preheated Air

With the quantity and quality of the heating operation separated, it will be readily seen that the heat-salvage device, whether tile recuperator, alloy recuperator, or regenerative preheater, should be considered on its merits alone and from the return given by the added investment. The zone-controlled furnace places no limitations upon the degree of preheat in the air. For this reason, any heat-salvage device may be considered as operating at its greatest possible efficiency and its corresponding return can be fairly calculated.

Rust zone-controlled furnaces are operating with Rust refractory-tile recuperators, Mantle recuperators, and Blaw-Knox regenerative air preheaters, furnishing preheated air for combustion. Others are installed with cold air where conditions warrant it. In many cases, any one of the above heat-salvage devices may be used satisfactorily. The one exception which should be mentioned here consists of high combustion-chamber furnaces which require the use of producer gas as fuel. Since producer gas has but 1 in. or so of pressure available, it is found desirable to use some form of air heater other than the tile recuperator in order to deliver the hot air under an appreciable pressure. It is always desirable that either the fuel or the air be under sufficient pressure to provide proper mixing through a burner block and at the same time provide proper flame directional

characteristics. With the refractory-tile recuperator, the pressure of the air delivered to the top heating burners may be as high as 0.3 in. of water. This is not sufficient with producer gas as fuel due to the static pressure found in the high heating chamber.

A number of end-fired furnaces having high-combustion chambers, have been constructed with refractory-tile recuperators and producer gas for fuel. Auxiliary cold-air pressure jets are employed to furnish the medium by which preheated air and low-pressure gas are forced into the furnace chamber. Cold-air jets have a double objection. Cold air reduces the temperature of the preheated air for combustion which it is helping into the furnace and, accordingly, reduces the efficiency of the heat salvage apparatus and the furnace as a whole. At the same time, it necessitates the use of mechanical equipment to furnish the second source of air supply and complicates the heater's work by having additional valves and regulation dampers to handle. The reduced air temperature of the mixture of hot and cold air lowers the flame temperature of the furnace. In one way of speaking, this is a desirable feature for end-fired furnaces, as a lowered flame temperature would more nearly approximate the temperature of the steel at the discharging point. However, the benefit of the heat-salvage device is reduced, and in many cases would not be a paying proposition when operating under these conditions, and is, accordingly, contrary to proper combustion-engineering principles.

On any type of continuous furnace having a high-combustion chamber, any type of air-heating equipment may be used satisfactorily by using any fuel other than producer gas. The type of heat-salvage device to be used depends entirely upon existing plant conditions and the entire mill layout.

Summary

The author has attempted:

1 To prepare a chart containing a classification of rolling-mill furnaces which may assist the steel-plant operator in the determination of the correct type of furnace for his specific requirements, and at the same time divide furnaces into classes which make possible a fair means of comparison with regard to fuel consumption, operating efficiency and tonnage output.

2 To point out that the main requirement for all types of continuous furnaces is to maintain a temperature at the point of discharge approximately the same as that desired in the discharged material, that quality heating may be obtained.

3 To point out that if quality heating is obtained from an end-fired furnace where the maximum temperature is carried at the discharging point, many undesirable conditions must be put up with, such as slow heating, a long furnace, high oxidation losses, and the elimination of the full benefits possible with preheated air.

4 To explain that if only sufficient fuel is introduced at the discharge point to care for radiation losses and the remainder is fired in a separate heating chamber, it is possible separately to control the quantity of heating from the quality of heating. This condition makes possible a rapid-heating furnace, one flexible of control and capacity output, reduced radiation losses, and the use of air-preheating equipment at the greatest efficiency.

Discussion

E. A. Hawk.[2] The author has brought out many interesting points in regard to continuous reheating furnaces. In this day of higher efficiencies and greater economies, complete and thorough studies of the different heating problems are necessary.

[2] Fuel Engineer, Department of Research and Technology, Tennessee Coal, Iron & R. R. Co., Ensley, Ala.

Recent installations of triple-fired continuous furnaces with soaking hearths in the Birmingham district include:

(1) Two 15-ton high-pressure natural-gas-fired furnaces, without recuperators, heating 4-in. by 4-in. billets.

(2) One 30-ton low-pressure by-product gas-fired furnace, with tile recuperators, heating structural shapes including 5-in. by 5-in. blooms up to 18-in. beam blanks, the soaking hearth being fired with high-pressure by-product gas under automatic temperature control.

The performance of these furnaces in regard to tonnage and product heated is satisfactory.

In order to show the difference in the rate of heat absorption by steel in passing through a continuous furnace, the chart

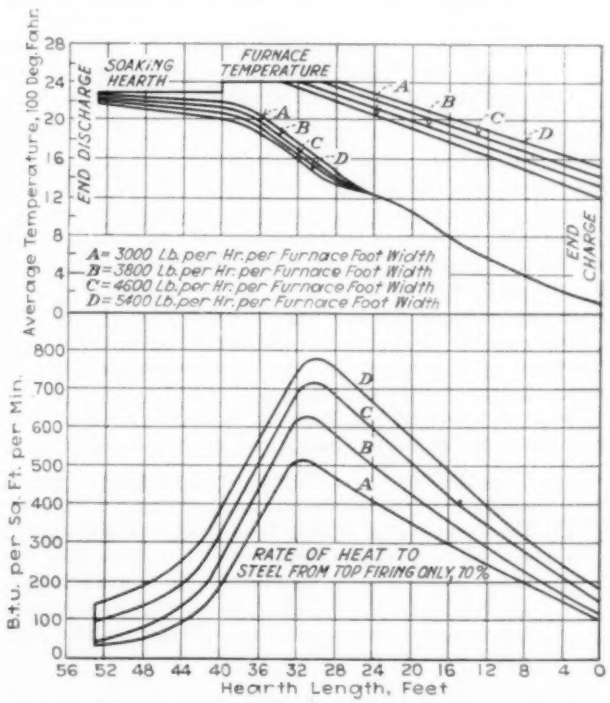


Fig. 8 Continuous Furnace, Triple Fired, Heating 8-In. by 8-In. Blooms

(Fig. 8) is shown. Consider a triple-fired furnace having a soaking zone and top and bottom firing in the main body of the furnace, heating 8-in. by 8-in. blooms at the following rates:

A = 3000 lb. per hr. per furnace foot width = 56.6 lb. per sq. ft. per hr.
B = 3800 lb. per hr. per furnace foot width = 71.7 lb. per sq. ft. per hr.
C = 4600 lb. per hr. per furnace foot width = 86.7 lb. per sq. ft. per hr.
D = 5400 lb. per hr. per furnace foot width = 101.8 lb. per sq. ft. per hr.

As the tonnage rate increases, it is necessary to extend the high-flame temperature zone in the direction of charging, resulting in a slight increase in waste-gas temperature. After passing through the 1150 to 1400 deg. fahr. critical range, the rate of temperature increase becomes greater, and overheating results unless the furnace temperature is closely regulated. By maintaining the soaking zone at constant temperature and varying the fuel rate in the main body of the furnace according to the demand, this type of furnace serves the same purpose as a flywheel on a rolling mill, storing up energy during light loads, to be relieved again during periods of peak loads.

When this type of continuous furnace is used in conjunction with in-and-out furnaces supplying steel to a mill where the maximum rolling rate is 100 tons per hour and where one in-and-out furnace can discharge 50 tons in half an hour, the continuous

furnace must discharge at a similar rate; for example, 25 tons during a 15-min. period of each hour. This high rate of discharge is made possible by use of the soaking zone designed to hold sufficient tonnage at rolling temperature.

When heating 2-in. by 2-in. billets, the advantages of the two-zone furnace are not as apparent, as:

(1) The temperature difference between surface and center of the billet is small
(2) Heating through the 1150 to 1400 deg. fahr. range is not so pronounced
(3) As bottom firing is not required, there would be no skid marks to remove with a soaking hearth
(4) With side discharge, the furnaces are less subject to cold-air infiltration, except when too steep a sloping hearth is used
(5) Extending the high-temperature zone too far back will cause 2-in. by 2-in. billets in a sloping hearth to slip past the discharge door if the slope is greater than 2 in. per foot.

Even for the large-size billets, such as 3 in. by 3 in. and 4 in. by 4 in., the advantages of the two-zone furnace may not be so apparent, especially when the rate of heating is fairly uniform. For example, a merchant mill furnace in the Birmingham district has given satisfactory results heating 4-in. by 4-in. billets, 20 ft. long, this furnace being top-fired only, having level hearth and dry skids and using proportioned by-product gas and cold air.

The use of preheated air is not necessarily limited to the two-zone furnace. Higher preheats are being used successfully by proper regulation of the rate of mixing air and fuel.

The writer considers the two-zone furnace as primarily intended for large sections requiring end discharge, as:

(1) Large sections require top and bottom firing and skid marks are removed in the soaking zone
(2) The soaking zone is made long enough to provide the mill for high rates of rolling
(3) Large sections must preferably be end-discharged; air infiltration is reduced considerably by employing a soaking hearth operated under a slight positive pressure.

Too liberal size of water-cooled skid pipes and skid supports increases water-cooling losses very rapidly. Cooling-water losses may amount to as high as 25 per cent of the total fuel supply.

For some operations, the in-and-out type of furnace may be preferred to the continuous furnace, as:

(1) When charging hot steel
(2) Slabs or flats requiring a wash heat to remove scale
(3) To eliminate buckling which may occur with light slabs or flats in a continuous furnace
(4) On mills rolling high- and low-carbon steels
(5) Miscellaneous small tonnages.

Quality of product, of course, is essential. The type of fuel available, mill conditions, and labor costs make each heating problem an individual study. The advent of automatic control of temperature, of air and fuel ratios, and of furnace draft is receiving considerable attention, not only in the application to new furnaces, but to existing furnaces as well.

G. T. Hollett.[3] The subject of furnace design, with actual authentic data thereon, is noticeable on account of its absence. Until very recently the subject of furnace design received little scientific study in this country. Fuel costs were only a small part of the total cost of the manufacture, and the quality of product was not so exacting, so that most reheating furnaces were designed and constructed by the mason foreman or other local talent. A great deal of credit must be given to those practical men who perhaps could not actually calculate what the areas should be or what size of port to use, but through trial could observe results, and at each rebuilding of the furnace would approach their ideals. This school of practice is one quite often necessary even at the present time, as usually minor changes are necessary on a supposedly fine, technically constructed furnace, due to the designer not securing all the facts or because of some alibi not recorded in the contract and hence being at the purchaser's expense.

The author has given a classification of furnaces which enables us to divide the furnaces into groups. These divisions are arbitrary, and while no definite line separates a wide furnace from a medium furnace or a narrow furnace, this classification does give a division point. We may not all agree on the exact dimensions, but our arguments for division points may not be any more weighty than the author's. It is unfortunate that all wide furnaces cannot be constructed with side discharge. With proper forethought this can be accomplished in most cases and represents a neat return on fuel costs. A side discharge is impractical, and end discharge must be used as is necessary when handling heavy sections. A proper door-opening mechanism should be provided so as to keep the end of the furnace closed except when the steel is being discharged. This has been worked out in several cases and has proved practicable.

The author has emphasized uniform heating, particularly for piercing furnaces. This is becoming more and more true of other products as well. The open-hearth or bessemer may make a good quality of steel only to have it ruined by improper heating in the reheating furnaces. With the quality of product the trade is now demanding, it is not permissible to subject the steel to tremendous heat shocks and then expect to produce a steel that will not show structural fractures. If we can deliver steel to the mills after it has been heated up by a proper cycle from end to end and center to outside, the chipping expense and rejections would be less than the average practice. Proper furnace design will help to accomplish this result. Quality heating requires a definite cycle, and this cycle should not be exceeded even in a zone-controlled furnace. If a furnace is properly rated, then the capacity cannot be increased 50 per cent without definitely decreasing the quality of heating, regardless of type of furnace. As soon as a zone-controlled furnace is pushed above the point where the steel is entering the soaking zone below the proper temperature, the purpose of the soaking zone is defeated and it becomes an end-fired furnace.

The six continuous furnace-operating requirements are well taken. Uniform heat distribution throughout the mass of discharged product at a predetermined temperature is what operators are striving for. Flexibility as to tonnage output without any appreciable loss in efficiency or change in quality of heating may be obtained if the output of the mill does not vary over too wide a range. Where this variation is over a wider range, the radiation and fixed losses are such a large percentage of the total input at the low ratings that this will not be true. Fortunately most of the continuous-heating furnaces are operating on mills that have a fairly steady rate of reheating. The low oxidation loss is a very much desired item and good results are being obtained. However, after observation of various zone-controlled furnaces, we are making provisions to remove scale from below the hot portion of the heating zone on the three continuous-heating furnaces we are installing. The low furnace maintenance is very desirable. We often sacrifice this point and install a

[3] Illinois Steel Company, South Chicago, Ill.

furnace for, say, 25 tons per hour and then expect the operator to produce 30 to 35 tons without increasing the maintenance cost. The rating of the furnace cannot be increased materially if the original rating of the furnace was properly calculated unless one defeats the zone principle and makes the soaking zone a part of the heating chamber. One should not try to operate a furnace at 50 per cent over its true rating and expect results. The writer has found this to be true in some furnaces of this type recently inspected. With a fixed height of throat between the soaking chamber and the heating chamber it is impossible to adequately throttle the flow of gases from the soaking chamber to the heating chamber unless the furnace is operated on one definite thickness of material heated. In some instances the variation in thickness of steel being heated may affect the area of the throat as much as 25 to 30 per cent, which will materially change the draft condition in the soaking chamber. This perhaps is the exception to the rule, but it is something that the writer is facing in actual practice.

Author's Closure

The author appreciates the frank discussion, which has brought out many interesting points.

It is particularly evident that all think alike with regard to the importance of "quality heating," the first requisite of a well-designed continuous furnace.

Mr. Hawk apparently believes that zone-fired furnaces are mainly advantageous for use with heavy sections. This is incorrect, as will be quickly seen if one thinks of the prime requisite of a properly designed continuous furnace—namely, "quality heating" or uniformity of heat distribution.

The design of the zone-fired furnace calls for a soaking chamber which has the sole duty of discharging steel uniformly heated throughout and at the desired temperature. This can be obtained only if the temperature at the discharge point of the furnace is maintained at or slightly above the predetermined temperature desired.

This is self-evident in the case of furnaces heating thin, wide slabs. Zone-fired furnaces at the plant of the Republic Steel Corporation, Youngstown, Ohio, are heating slabs $2^3/_4$ in. thick up to 35 in. in width and 12 ft. long. The soaking chamber proves its worth in delivering these slabs so uniform in temperature from front to back that a maximum of a few thousandths variation is found in the finished plate from end to end and from side to side.

Should an end-fired furnace use delayed combustion or regulated rate of mixing to prevent excessive temperatures when using preheated air, it is possible that for one definite tonnage rate correct temperature conditions in the discharged product might be realized. However, it would be impossible to maintain these conditions for fluctuating rates of output, and it also would be impossible to have a higher temperature in the furnace than that carried at the discharge end. This would result in a lowered efficiency, which condition is overcome by the use of zone firing, since the heating chamber employs as high a temperature as can be realized, with the amount of heat liberated or fuel fired varying with the tonnage rate desired.

The author has in mind a second installation which is used to heat billets and flats from 2 to 4 in. in thickness and 15 ft. in length. This is a double-fired furnace having no underfiring nor watercooling. It is located at the plant of the Steel Company of Canada. Due to the varying nature of orders oftentimes the tonnage demands vary from 10 tons per hour to 35 tons per hour, although the furnace is normally rated at 25 tons per hour.

Obviously when heating at the low rates it would be foolish to use the soaking chamber as anything but a soaking chamber. In this case the heating-chamber burners are turned almost entirely out; in fact, for the first week of operation they were not used at all. This resulted in the soaking chamber being carried only at a soaking temperature, with the entering portion of it employed for heating. The time of travel through the soaking chamber is regulated by the tonnage demand. This permits the same length of time in the soaking heat as would be obtained at the higher tonnages; however, the distance of travel is much less. In other words, if a soaking hearth 12 ft. long is required to give the proper time of soaking at 30 tons per hour, then a soaking hearth 2 ft. long would give the same time for soaking when operating at 5 tons per hour.

During low-tonnage operation practically 85 per cent of the fuel is shut off by cutting out the heating-chamber burners.

The author agrees with Mr. Hawk that flexibility as to tonnage output without any appreciable drop in efficiency is impossible over extra-wide ranges of tonnage. However, he wishes to point out that since the soaking chamber utilizes only approximately 15 per cent of the total fuel when operating at the normal rating of the furnace, a zone-fired furnace makes possible a much greater flexibility and a wider range of tonnage, since it is possible to vary the 85 per cent downward to zero and still for overload capacities increase it to change the proportions to approximately 90 to 10.

The author believes that a proper normal tonnage rating for a furnace should take into consideration furnace maintenance. Every one knows that furnace maintenance goes up with excessive overloads. If the furnace is properly rated at 20 tons per hour, such as the installation at Republic Steel Corporation, it should be able to be pushed without excessive maintenance increase to 25 tons per hour. If desired, it also should be able to be pushed beyond this, with the operator well knowing that increased maintenance will result. Tonnages of 31 tons per hour have been obtained over an appreciable length of time without excessive maintenance having yet been noticed. However, undoubtedly such overloads will shorten the life of any furnace.

The Steel Company of Canada double-fired furnace with a 25-ton normal rating has delivered from 5 to 38 tons per hour.

The Timken Roller Bearing triple-fired installation with a normal rating of 40 tons per hour operates nicely at 50 tons per hour and has turned out $61^1/_2$ tons per hour at various times.

It is to be borne in mind that a zone-fired furnace obtains its higher tonnage and greater overload capacity, not by using the soaking chamber as a heating chamber (except when extra thin stock is being heated, which of course requires a shorter time for soaking), but by increasing the high-temperature portion of the furnace and permitting the steel to more rapidly enter the high-temperature zone and thus heat more rapidly in the heating chamber. It is also to be remembered that at higher tonnages an increase in temperature of the waste gases increases the temperature of the preheated air, which in turn increases the heating-chamber flame temperature. This is more evident in the zone-fired furnace using cold air for combustion on the soaking chamber than on the ordinary end-fired furnace, since only approximately 85 per cent of the air for combustion is preheated, yet 100 per cent of the waste gases is available for this heating.

With regard to the observations on scale losses in Mr. Hollett's discussion, the author wishes to point out that overheating may occur with regard to time as well as temperature.

Every one knows that a piece of steel which is hot too long, as occurs during delay periods, has more scale on it than when it is heated speedily as in normal operation of the furnace.

The tendency of a heater is to insure himself of plenty of hot steel. A zone-fired type of furnace makes it possible for him to play extra-safe in his heating. This sometimes results in

heating the steel too far back in the heating chamber, with the formation of more scale than is necessary. If the operator properly controls the heating-chamber burners so that the billets are permitted only their proper time for soaking, which is dependent entirely upon their thickness, both fuel and scale can be saved. It therefore behooves the users of zone-fired furnaces to have careful, intelligent heaters if the highest efficiency possible is to be obtained.

Furnace maintenance, particularly with regard to refractories, is noticeably high whenever more heat is liberated in any portion of the furnace than can be absorbed rapidly by the steel. This is noticed at the discharge point in ordinary end-fired furnaces where the steel and flame are hottest, the differential being so reduced as to permit the refractories to become overheated frequently.

It is interesting to note the temperature condition in a zone-fired furnace. It will be found that temperatures of the refractories in the downtakes are much higher than the temperatures of the roof in the channel as far as 30 ft. from the downtakes. This should explain the reason for high maintenance. The low roof temperatures are due to the ability of the steel to absorb heat more rapidly than the refractories, and hence the refractories radiate this heat rather than maintain it. It is for this reason that the heating chamber of the zone-fired furnace has shown such a low maintenance requirement.

The heating chamber is large and contains relatively cool steel as compared to the temperature condition found in the ordinary end-fired furnace at the discharge point, where in each case the highest temperatures are carried.

The cooled steel rapidly absorbs the heat and protects refractories in the one case, where this is not possible in the other due to the higher temperature of the steel.

German and American Steel-Plant Practice

By F. L. ESTEP,[1] NEW YORK, N. Y.

The United States leads the world in the production of steel and Germany has again become second. The greater percentage of the total amount of steel made in the United States is used domestically, whereas, on the other hand, a very large percentage made in Germany is exported. This in itself has had a certain influence upon the processes developed, methods pursued in the two countries, and the supply and cost of raw materials. Another factor which has affected the methods and the equipment used is the quantity and quality of labor available and the wages which must be paid.

In Germany today the greater percentage of ore used comes from foreign countries, while in the United States the great bulk of the ore is mined at home. Coal is plentiful and of good quality in Germany, but its price for metallurgical purposes is higher today than the coals commonly used in the United States for the same purpose. This paper shows to some extent where the processes differ.

Coke Plants

THERE is little to choose between the latest coke plants of the United States and Germany. Ovens have been designed and built of sizes to suit the particular coals. Both countries have gone to high ovens, and widths have been reduced in order to lower the coking time. In both countries the heating of the ovens is under automatic control; all mechanism for the control of temperature and the handling of gases and of by-products has been made accessible. Means have been provided for quenching and handling the coke at the ovens, and delivering either directly to the points of use or to central points for distribution or shipment.

It is quite common in Germany to find coke ovens built at the coal mines, so that coal delivered from the tipple passes through a preparation and washing plant directly to the ovens. In the United States it is the practice to build ovens at the point nearest to the use of either the coke or gas, and haul the coal to the ovens.

Blast Furnaces

In the United States a blast-furnace shell, supported by a mantel and columns, carries the entire weight of the top and superstructure, and encloses the brickwork. In Germany the furnace lining is usually built and held together by bands, and separate columns extend from the ground to the top of the furnace and carry all the weight above.

The United States leads in the number and percentage of large furnaces. There are perhaps six furnaces in the United States that have averaged over 1000 tons per day for at least one month and one furnace which averaged over 1000 tons per day for the year 1928.

The methods of filling the furnaces are again different. The practice in the United States is to use double skips under automatic hoisting control, and also to lower and close both large and small bells automatically. In Germany, the almost general practice is to fill the furnaces by means of large circular buckets, which, when set on the furnace top, automatically discharge their contents into the furnace.

With the bin systems in common use in the United States, the amount of labor required for filling the furnaces has been reduced to the absolute minimum, whereas with the system in use in Germany the labor is greater. Several of the larger furnaces in the United States are today being charged with one man per shift for eight hours.

Because of the fact that since the war Germany has had to obtain ore from many different countries abroad, much of the ore received contains a very high percentage of fines which must be sintered. The percentage of sinter used in the burden is much higher than in the United States. This has enabled them to increase their productions very materially, lower their coke consumption per ton of pig, and lessen the pressure required on the furnaces.

The United States has advanced farther than Germany in the preparation of raw materials, other than the question of sinter. Ore is being crushed and sized, as is also the limestone. The same holds in regard to the crushing of coke, and it has become common practice to crush and screen the coke so that furnace coke will not exceed $3^1/_2$ in. maximum or 1 in. minimum in size.

In Germany blowing of blast furnaces and converters is almost always done by gas engines, whereas in the United States comparatively few gas engines are used and none has been installed for several years in any steel plants. Blowing in the United States in the plants with the larger furnaces is done almost exclusively with turbo-blowers. These have been developed to a very high degree of reliability and with very great reserve power, enabling one blower to deliver sufficient air for the largest furnace under all operating conditions of fluctuations in pressure up to as high as 35 lb. maximum.

In Germany a number of gas-blowing engines deliver air into a common cold-blast main leading to the stoves and blast furnaces. From this common supply of air each furnace receives its air through a valve, the pressure being reduced to that required for each individual furnace. In other words, the pressure in the system as a whole must at all times be in excess of the highest pressure on any particular furnace. From the standpoint of operation, to the mind of American blast-furnace men and engineers, this system is entirely wrong. Blast furnaces have reached such sizes and volumes that the requirements for air have surpassed the blowing capacity of any one single gas-engine unit which has been built to date.

Because of the comparatively high price of coal, it has always been essential that a steel plant in Germany operate at the highest possible thermal efficiency. This has necessarily involved the use of gas engines for blowing and for the generation of power. If and when Germany uses individual blowing units on each furnace, and when they size the raw materials along the lines mentioned, then, in conjunction with the 20 per cent to 30 per cent of sintered ore in the burden, it is the opinion of many Americans that Germany may again take the lead in output per blast furnace.

The difference in the over-all thermal efficiency of blowing equipment for a steel plant, between an installation using gas engines with waste-heat boilers and a modern high-pressure boiler plant burning blast-furnace gas, is steadily decreasing. Al

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

though the efficiencies will never be the same, the difference is becoming so small that, all things considered, including interest, depreciation, and maintenance, turbo-blowers will always hold their present supremacy for blast-furnace operation in the United States, and may have first consideration in the near future in Germany.

Because of the necessity during the past years for the cleaning of blast-furnace gas to a degree suitable for gas-engine purposes, German steel plants have been in a position to try this very clean gas for many different purposes, such as heating of stoves, general reheating purposes, and the mixing of blast-furnace gas with coke-oven gas for metallurgical and reheating purposes. In this respect they are at the present time far ahead of the United States, where there are only two or three installations using a mixture of blast-furnace and coke-oven gases.

In Germany it is common practice to granulate blast-furnace slag for adjacent cement plants, where properly prepared granulated slag is used in quantities varying from 60 per cent to 85 per cent in the manufacture of cement. In the United States, on account of the comparatively low cost of quarrying, crushing, and preparing limestone, very little blast-furnace slag has been used in the manufacture of cement.

Steel-Making

In Germany about 45 per cent of the total amount of steel made is produced by the Thomas process. The remainder is made in open-hearth furnaces, usually of the stationary type, where practically all of the scrap made in a plant is melted. Thomas steel is used for ordinary products, such as rails, structurals, merchant material, rods, and wire. Open-hearth steel is used for seamless tubing, sheets, tinplate, and other products where the open-hearth grade is better suited.

In the United States the percentage of the total production of steel ingots by the bessemer process has steadily decreased over the last 30 years. Whereas in 1900, 6,684,700 tons, or 65.5 per cent of the total, was made by this process, in 1928 there was made 6,591,745 tons; but this was only 13.1 per cent of the whole.

Because of the predominance of Thomas steel in Germany, an arrangement of plant different from United States plants has been possible. Thomas steel can be poured and stripped very quickly, and for many products can be brought to a rolling temperature without fuel by merely enclosing in a single ingot pit for about two hours.

In practically all plants in Germany the slag made by the Thomas process is made into fertilizer containing approximately 15 per cent to 16 per cent of P_2O_5 soluble in citric acid. In one plant in the United States where the duplex process is used, the primary or first slag is poured off and made into fertilizer containing about 18 per cent P_2O_5.

The later open-hearth furnaces in the United States are larger than those in Germany. They range from 100 to 150 tons rated capacity, with a few furnaces of 250 and one of 400 tons, while in Germany the later furnaces range from 80 to 100 tons capacity, with a few up to 200 tons. In the United States practically all furnaces, except those originally designed for the duplex process, are of the stationary type. In Germany there are a few tilting furnaces.

The construction of stationary open-hearth furnaces in the two countries is generally the same, but there are some points where they differ more or less radically. In the United States furnaces are equipped with very efficient water-cooled doors, frames, and ports to insure the longest life and the minimum delay of production throughout the year. Water cooling is not done on German furnaces to any marked extent. American furnaces are also built with sloping back walls which very greatly increase the furnace life and minimize delays in production.

In Germany checkers have been developed to a much higher degree of efficiency in heat transfer than is common practice in the United States. They are building checkers of large volume and extreme height, making it possible to preheat the air and gas to a very high temperature. This means also that the temperature of the stack gases is very much less than here, so low in fact that it is an uncommon thing to find a waste-heat boiler installed on an open-hearth furnace in Germany. The claim is made that the proper place to secure the highest overall efficiency in open-hearth operation is in the heat transfer in the checkers, and not by the absorption of excess heat in waste-heat boilers. The latter are used to a very marked extent in the United States. Open-hearth design in the United States must therefore take this point into very serious consideration and develop checkers of proper dimensions, with checker openings suitable to secure maximum economy without waste-heat boilers.

Fuel consumption per ton of ingots in Germany is generally considerably better. This is partially due to the use of a mixture of blast-furnace and coke-oven gas in their open-hearth furnaces, this having been developed to a very high degree of efficiency and reliability. Their late furnaces are equipped with mixing and regulating devices and a full complement of instruments per furnace, whereby it is possible to know at all times the exact calorific value of the mixture, the percentage of CO and CO_2 in the waste gases, the stack temperature, and the general over-all efficiency of the furnace. Furthermore, they have become very skilled in the operation of a furnace with this fuel, using a rich mixture for initial melting, and during the latter portion of the melting time reducing the calorific value of the mixture by the addition of more blast-furnace gas, and thereby effecting an over-all economy of operation that is quite remarkable.

In the United States the percentage of scrap and hot metal used at the open hearth depends largely upon the price and quantity of scrap available. It usually fluctuates from 40 per cent scrap and 60 per cent hot metal to 55 per cent scrap and 45 per cent hot metal. It has been found that operations are slowed down materially if the percentage of hot metal is much over 60 per cent. In Germany the percentage of scrap is, as a rule, much higher, because if a plant has both the Thomas and open-hearth processes, practically all of the scrap is used in the open hearth, a small percentage occasionally being charged into the converters.

In the United States there are eight plants with equipment of mixers, converters, and furnaces for the manufacture of steel by the duplex process. One plant has always operated by this process and is running at full capacity, making about 70 per cent of rail steel and 30 per cent of soft steel. In the remaining plants the equipment is used in various ways. In one or two plants the furnaces are used as straight open hearth and the converters for straight bessemer; in others some bessemer steel is made and duplex furnaces are used by a modified process, and in others some of the furnaces are used as straight open hearth and the remainder run on a modified duplex process. However, in one plant where there are three duplex furnaces, one is used as a straight open hearth to melt practically all of the plant scrap, and the other two are operated by the (strictly speaking) duplex process and make high-quality steel, including high-carbon steel for seamless tubes.

The design of open-hearth furnaces in the United States has not yet reached the limit of proper proportions and design for long life, combined with greatest economy. As has been mentioned previously the design, size, and proportion of checkers, the size and laying of the checker brick, together with the use

of fuel in the form of a mixture of blast-furnace and coke-oven gas under automatic control, will undoubtedly greatly reduce the time required per heat for the manufacture of steel of any given quality.

Mills

The first rolling operation of finished steel products in Germany, as in the United States for the great majority, must necessarily be that of converting an ingot into a bloom or a slab for finishing into some specific product on another mill.

Soaking Pits

Because of the fact that in Germany a large percentage of ingots is made by the Thomas process where heats varying from 25 tons to 40 tons are usually blown, it is possible to strip the ingots very quickly after being poured, and get them very quickly into single-hole ingot pits, where they are allowed to soak. For the soft steel made by the Thomas process, the ingots reach a rolling temperature without application of heat. On rail steel and soft steel for some products, heat is applied usually in the form of blast-furnace gas to the soaking pits for a short length of time after the ingot is first soaked and has equalized its own temperature. In a plant making Thomas steel about 40 per cent to 60 per cent of the total number of pits are without gas.

In the United States, where the great percentage of steel is made by the open-hearth process, it becomes essential that the soaking pits and blooming mill be separated from the open-hearth plant with an intermediate stripping point. This means that it is impossible to put ingots into the soaking pits at as high a temperature as is possible in a plant making steel by the Thomas process.

The soaking pits therefore in the United States have been developed along different lines. Heating is done with producer gas, coke-oven gas, a mixture of coke-oven gas and producer gas, tar, a mixture of tar and coke-oven gas, and, infrequently, with oil. Practically all pits are provided with checkers for preheating both air and gas. However, in one or two plants where coke-oven gas is used, a pit has been developed without checkers, using the gas under automatically controlled pressure with a special type of burner. It is claimed that the fuel economy over a year in pits heated in this way is equal to or better than the general average in the United States of pits with checkers. Furthermore, it does not cost more than 40 per cent of that of the ordinary type and the maintenance is very much reduced.

In regard to fuel consumption, there has always been a great deal of confusion and misunderstanding between the results obtained in Germany and the United States. This is due to the fact that, usually, proper allowance is not made for the large percentage of steel made in Germany by the Thomas process and, furthermore, that perhaps 60 per cent of the total steel made by that process in any given plant requires no heat whatever. If they were put on a strictly comparative basis for open-hearth operation only, it will be found that the average fuel consumption per ton of ingot in Germany would be somewhat less than that in the United States, but no better than in some of the later and best-operated soaking pits in the latter country. These pits can be operated on about 100 lb. of coal equivalent per ton of ingots, or expressed in another manner, 4.5 per cent of fuel.

Blooming and Heavy Finishing Mills

Blooming mills in both countries are practically all of the two-high reversing type, the later and larger installations all being motor driven. In each country ingots, usually of 4 to 5.5 tons in weight, are rolled with the minimum number of passes practical into a bloom of a size suitable for transfer to a finishing mill.

The finishing of rails, heavy and medium structural shapes, and some special shapes, is done in Germany almost generally upon two-high reversing mills, consisting of two or three stands adjacent to and usually in line with the blooming mill. Finishing is done from the original ingot heat. The blooming and finishing mills for this class of product in Germany are somewhat larger than those in common use in the United States. Blooming mills will run up to 52 in. pitch diameter of pinions, whereas in the United States the corresponding type of blooming mills range from 40 to 45 in. pinion diameter. German finishing mills, such as those referred to, have rolls of from 800 to 900 mm. in diameter, whereas for the finishing of the same sized structurals in the United States the mills do not exceed 32 in. in diameter.

In the United States a mill to roll rails or structurals up to 24-in. beams is arranged differently than in Germany. Here it is customary to bloom an ingot, deliver it on the back side of the mill, crop it, and send it in blooms of proper lengths to the finishing mill. In some few installations a two-high engine- or motor-driven reversing rougher is interposed between the blooming and finishing mills. In many plants in the United States, sections corresponding to those which would be rolled in Germany direct from the ingot heat are reheated before rolling.

From the viewpoint of American engineers and mill operators, the layout of the German soaking pits, blooming mill, and heavy finishing mills in relation to each other is awkward and wrong. This is not merely a question of difference of opinion. It is practically a certainty that, for the majority of products which can be rolled direct from the original ingot heat, if mills of the two types were installed side by side and given the same materials to roll into the same finished products, the American mill would produce a greater tonnage in a given length of time with less labor. The first cost of the American installation would be little if any higher.

Plate Mills

In the United States practically all plate mills are of the three-high of "Lauth" type, with large-diameter top and bottom rolls and a smaller-diameter middle roll. There is one installation of a very heavy four-high engine-driven, reversing plate mill, the top and bottom backing-up rolls being large in diameter and the two working rolls small. In some plants two stands of three-high rolls make a complete mill unit, one stand being used for roughing to width and rolling to a medium thickness, while the finishing is done on the second stand, thus giving a better quality of surface to the plate.

There are also several two-high, reversing universal mills, and one or two three-high mills. These roll universal plates up to a maximum width of approximately 48 in., and by removal of the vertical rolls can roll sheared plate to a width approximating 50 per cent wider than the maximum universal width. These mills are largely used for skelp for medium diameter of lap-weld pipe.

Merchant Mills

There is a great difference between the types and capacities of mills in the two countries for the rolling of what is termed "merchant products." These include rounds and squares from $^3/_8$ in. minimum to 3 in. maximum; angles, tees, and other shapes from 1 in. by 1 in. minimum to 4 in. by 4 in. maximum; and the corresponding range of rectangles, flats, and special shapes.

Until very recently, practically all the mills in Germany rolling this class of product have been of a type producing a

very limited tonnage per month with a large number of men required for their operation. There were very few mechanical devices to save labor or increase the tonnage, and insufficient facilities for the finishing and handling of large productions.

In the United States there has been a gradual development for many years of special mills to roll these products, which have reached a very high state of perfection of design, workmanship, and flexibility.

These mills operate at extremely high tonnages with very low labor costs, with very low maintenance costs, and practically no interruption due to mechanical defects. There are still a few instances in two or three steel plants in the United States where one new mill of the aforementioned late type, of proper size and with the proper arrangement, can replace three or four existing old mills of the type first described, and roll an equal amount of tonnage with probably 25 per cent of the labor, 60 per cent of the power, and 75 per cent of the fuel now required.

During the last three years there have been a few installations in Germany of merchant mills arranged very much the same as the late American mills. This has resulted from studies by German engineers and operators of American mills and methods for the same type of product.

Rod Mills

As in the case of merchant mills, the United States leads Germany today in production of rods per mill per shift. Most of the plants in the United States operate with fewer men per ton of output than in Germany.

Rod mills here are generally of two types, the "Garrett" and the "continuous." The tonnage per shift and per month of the same size of rods is practically the same on the two types of mills, but the labor costs on the Garrett type is much higher. All of the later installations have been of the continuous type, on which it has been possible to finish No. 5 rods (0.207 in.) at a speed as high as 3600 ft. per min. With the Garrett type mill the limit in speed due to the hand sticking on the last pass is about 1500 ft. per min. Although the delivery speed of the continuous mills is about 2.5 times that of the Garrett mill, the output per unit of time on the two types is the same. On the continuous mill only two strands can be run at one time, but on the Garrett type it is possible to keep four rods in the finishing stand at all times.

In Germany just recently there have been one or two installations of rod mills somewhat different from either the American Garrett or continuous type, in which billets are roughed on a continuous train and are then rolled on a series of independently driven intermediate stands somewhat after the fashion of the Garrett mill. These have proved quite successful and will undoubtedly lead to some modification in layout and design better adapted to German demands for tonnage and labor conditions.

Sheet Mills

The United States leads the world in total production of sheets and very greatly excels in the manufacture of high-grade sheets for automotive construction, deep stamping, and drawing purposes.

Many factors have entered into the process of manufacturing high-grade sheets as of today, among which may be mentioned the quality of steel furnished the sheet mills, the surface of the sheet bars, the automatic control of temperature in pair and sheet furnaces, the operation of the mills on the "loose-screw" system, the methods of cold rolling, the "normalizing" of sheets, and, where necessary, leveling by mechanical means.

In Germany there is but little demand for high-grade sheets, and the production as a whole is comparatively small. Therefore, Germany has not been in a position to follow, only in a modest way, the methods perfected in the United States.

In both countries the roughing and finishing is performed in much the same manner. The number of men per crew is practically the same, but mechanical doublers have been in use in the United States for some few years. This is a device which can be operated by unskilled labor, used to double the sheets from twos into fours, threes into sixes, or fours into eights, and replace a very highly paid doubler with a cheaper operator. This reduces the labor cost per ton and to a certain extent affects the production in favor of the mill equipped with mechanical doublers.

In the United States the demand for sheets for automotive and deep-stamping purposes, heretofore supplied from the ordinary sheet mill by more or less special processes, is gradually being replaced by strip. Whereas until six years ago the widest strip then rolled was approximately 14 in., the widths were increased up to 24 in. In the early part of 1926, and since, mills have been developed for the rolling and coiling of strip up to 48 in. wide.

These mills are usually of the four-high type with driven small-diameter working rolls backed up top and bottom by larger-diameter idle rolls, arranged in continuous formation. It is possible to roll 30 in. wide No. 16 gage, 36 in. wide No. 14 gage, and 48 in. wide No. 12 gage at finishing speeds ranging from 600 ft. to 1000 ft. per min. The same type of four-high single or continuous mills is beginning to be successfully used for cold-rolling strips to final gage. It remains to be learned just how far this type of continuous hot and continuous cold rolling will affect the present methods of manufacturing sheets.

Tin Mills

As was the case in connection with sheets, the United States leads the world in the production of tinplate. In 1928 the production was 1,970,000 tons, being approximately 60 per cent of the total world's production. The accompanying table gives the production of tinplate in Germany, the United Kingdom, and the United States.

TINPLATE PRODUCTION IN TONS

Year	United Kingdom	Germany	U. S. A.	Total 3 countries	Per cent U. S. A.
1891	....	....	1,000	.....	..
1900	....	....	303,000	.....	..
1910	817,000	57,000	773,000	1,647,000	46.9
1920	608,500	31,000	1,437,000	2,076,500	69.2
1926	580,000	97,500	1,782,000	2,459,500	72.4
1927	750,000	130,000	1,657,000	2,537,000	65.3
1928	....	....	1,970,000	.....	..

On account of the high cost of labor compared with that abroad, the United States has been forced to a high production per unit of equipment to reduce the fixed charges and to offset, partially at least, the increased cost of labor. In Wales, Spain, and in some plants in Germany tinplate is rolled today in packs of eights from one bar, doubled four times and heated five times. The latest mills in the United States roll tinplate from a pair of bars doubled on a mechanical doubler into fours after the first operation, drawn in fours and doubled into eights in the second operation, and finished in eights as the last operation with a total of three heats.

Abroad, a mill consists of a rougher and finisher in almost all cases, except in a few instances in Germany, where a single stand of rolls is used for roughing and finishing. In Wales up to 1922 a crew consisted of four men. Later on, two men were added to the crew, with no increase in production whatever. The same sized crew holds for Spain and on most mills in Germany. This crew of six men, on the average range of sizes and gages of tinplate, produces per mill per shift of 8 hours about 60 to 65 boxes or three tons.

In America there are two types of tin mills. The single-roll mills are operated some by the four-part system today where mechanical doublers have not yet been installed, and by the three-part system with mechanical doublers. The two-roll, three-part system has mechanical doublers for both the rougher and the finisher, one stand doing the complete roughing and the other rolls the fours and the eights.

The crew for a one-roll, three- or four-part-system mill in the United States consists of eight men, and a crew for a two-roll three-part-system mill consists of 16 men. The output per mill, per shift of 8 hours on the single mill will vary from approximately 4.5 tons per mill with the four-part system to 5.5 tons per mill with the three-part system. The production on the two-roll, three-part system will vary from 9.5 to 11 tons per shift.

One plant in the United States has recently installed a four-high continuous hot mill for the rolling of strip to replace the roughing of pairs previously required. The plant consists of mills of the two-roll, three-part system with mechanical doublers. The success of operation by this method is being watched very carefully by the tinplate manufacturers in the United States.

Tube Mills

Tube mills can be classified as three kinds, butt weld, lap weld, and seamless. In the manufacture of pipe by the butt-weld process, the United States is far ahead of Germany in tonnage and equipment for its manufacture. Investigation has shown that Germany has very few butt-weld mills and all are of antiquated design and operated very inefficiently. It has only been during the winter of 1928 and 1929 that the first installation of a butt-tube mill equivalent in design, capacity, etc. to those in common use in America has been installed.

In regard to lap-weld pipe, practically none is made in Germany by this process. Lacking the facilities to manufacture seamless tubing the United States necessarily had to develop a method of manufacturing tubes of a size larger than could be made by the butt-weld process. Tubes by the lap-weld process have been manufactured here in diameters up to 24 in. and larger.

In regard to the manufacture of tubes by the seamless process Germany has always led the world. The United States has been very backward in adopting this process. It was first tried out a few years ago for the manufacture of small tubes, principally boiler tubes where the maximum diameter was approximately 4 in. On account of the demands for seamless tubing in the oil fields in the United States, particularly for the deep wells in lower California and in some of the later fields in Texas and Oklahoma, the question of the manufacture of seamless tubing was very seriously considered by different manufacturers in 1924 to 1928.

In Germanv the process of manufacturing seamless tubing consists of first piercing a billet or ingot, and then finishing a tube of the proper outside diameter and wall thickness on what is called a "pilgering" mill. This is a very interesting process calling for very special equipment, but which produces a tube of extremely good quality so far as physical strength is concerned.

In 1926 and 1927, two steel companies in the United States imported German-made equipment for the manufacture of seamless tubes by the piercing and pilgering method. Previous to this date, however, there had been developed a different type of mill for the manufacture of tubes up to 6 in. in diameter. A billet or ingot was pierced, and in some cases given a second piercing, fairly close to the finished outside diameter, and then rolled to outside diameter and to proper wall thickness in a special mill over a mandrel on the inside of the tube. This is called the "automatic" mill.

It is believed that the method of piercing a solid billet or ingot, as the preliminary step for the manufacture of tubes by either the pilgering or automatic process, has not yet reached the limit of possible perfection. There is being installed at the present time in Germany a new design of piercing mill which has been tried out experimentally, and under such conditions has warranted the expenditure for a proper installation. It is claimed that, if successful, this particular type of piercer will be able to pierce in one operation to a much thinner wall thickness than has been heretofore possible by either the Mannesmann or Stiefel process, and that the variation in thickness of wall at any point in the tube will be very greatly reduced.

Discussion

H. J. Freyn.[2] The paper gives a very good and comprehensive comparison of the German and American steel-plant practice. Of course, the author was unable within the compass of the paper to mention every point of difference, but there is one that might be discussed. It is the essential difference in plant layout in the two countries, and while American plants as a rule show a very simple layout which allows for future expansion and simple and efficient transportation, a great many European plants lack a clear conception of the essential requirements of modern production methods. This is partially due to the fact that many of these plants are quite old, and that the very diversified product militates against the establishment of a clear-cut future program.

Another point typical of the general plant layout in America is that, in spite of its greater simplicity, it is more flexible as to tonnage produced. The curtailment of production to 50 per cent in the United States will not handicap the cost of production in as detrimental a manner as would be the case in Germany.

Another point that may be worth mentioning is the much larger diversity of the products that have to be made in the average German plant. This is due to the restricted market and to the earlier development of the European industry, which took place at a time when the change from the small private ownership to a larger concern had not taken place. The complication caused by the diversity of product results in a variety of mills, which cannot be used efficiently. This weakness has been realized, and if there is anything remarkable in the German development of the last years, it is the trend toward specialization in the mills, brought about by mergers. It is the policy of the large companies to concentrate the rolling of rails and heavier structural shapes in certain plants, while others specialize, for instance, in plates and sheets exclusively.

On the question of coke plants, it should be noted that the Germans take less pains in making a high-grade coke than do the Americans. This is caused by the fact that coke plants and blast furnaces are independent, and that the blast-furnace man has no control over the coke plant. The by-product coke plants are very frequently located at the coal mines and are operated for the highest yield of by-products, rather than for quality of metallurgical coke.

In regard to blast furnaces, it may be mentioned that the two latest German plants, those of Mannesmann at Huckingen and Krupp at Essen, have adopted the American skip hoist. Skip filling has also been used for many years at the Rheinstahl plants. There is an unquestionable tendency in Germany to abandon in future the complicated and expensive system of tub filling, and to resort to the much simpler method of double skip filling.

The method of blowing, mentioned in the paper, does not seem to be as common as might be inferred from the author's

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statement. The two latest German plants use steam turbo blowers; moreover, some of the largest plants practice individual blowing exactly as is done over here.

One reason for the use of a common cold-blast main for several furnaces is the use of gas-blowing engines. The present large furnace productions cannot be met with single gas-blowing engine units. Moreover, the application of one single gas-blowing unit on a blast furnace is still somewhat hazardous. In addition, it is of course desired to operate the gas-blowing engines at the highest rating, and consequently at the best efficiency. Owing to the variable blast pressures, this can be done much more easily by having a minimum number of gas-blowing engines produce the blast for a number of blast furnaces, than to have each furnace blown individually by engines which may be operating at only two-thirds, or less, of their maximum capacity.

In regard to open-hearth practice, an essential difference between the two countries is the different layout of the charging and casting facilities. The superiority of the American arrangement is now being realized by a number of operators in Germany. The very keen interest in new open-hearth furnace designs in Germany is significant.

Whether with the advent of more efficient checker chambers the waste-heat boiler will disappear in America might be doubted. In Germany there is an increasing interest in waste-heat boilers, in spite of the low waste-gas temperatures that are being obtained with present efficient checker arrangements. This is due to the fact that a waste-heat boiler always presupposes the application of forced draft; the latter, however, is very desirable from an operating standpoint. It is possible that German development may not lead to the use of waste-heat boilers, but of recuperators, but it is certain that artificial draft with power-driven fans will become general practice.

The author's observations as to fuel economy in open-hearth practice hit the nail on the head. However, he neglected to mention that America is far ahead in the manufacture of high-grade steels in large furnaces.

In regard to the difference in operating practice of the two countries it may be said that open-hearth furnace operation, for instance, is very different in Germany under otherwise equal conditions. Consequently the yield up to the soaking pit is higher in Germany, while in the finishing mills it is higher in America. It is not certain which method is best, but it is to be hoped that these problems will be taken up by the operating men of both countries as having a very important bearing on the design of steel plants.

F. C. Biggert.[3] There is an old saying that comparisons are odious, but the writer feels that the author has made his comparisons with such fairness and allowance for controlling factors that no offense may be reasonably taken.

Having had some experience in designing steel-mill equipment for foreign use and having listened to the frequently ill-considered criticisms of American steel-mill men after they have visited foreign steel plants, the writer has been impressed with the very great care necessary in making such comparisons. In fact, he has come to the conclusion that, given identical conditions of natural resources, markets, and other controlling factors, the engineers of any advanced nation will, in a short while, arrive at solutions identical in all major respects. The paper contains one of very few comparative statements where proper consideration has been given to the diverse operating conditions obtaining.

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Cleaning, Distribution, and Uses of Blast-Furnace Gas

By WILLIAM A. HAVEN[1] and C. B. THORNE,[2] CLEVELAND, OHIO

In this paper the authors discuss blast furnace operation and show the relationship between blast-furnace operation and the amount and quality of blast-furnace gas produced. A comparison is given between methods and costs of gas cleaning, the distributive equipment and cost of distribution, application for various purposes, and the profit that can be expected from blast-furnace gas as a by-product.

THE BLAST FURNACE is the largest producer of by-product gaseous fuel of any of the metallurgical processes. Depending upon the quality of the raw materials, the efficiency of the furnace operation, and the analysis of the pig iron being produced, the quantity of gas produced per ton of pig iron will vary considerably, but for good practice in American blast furnaces, producing basic and bessemer pig iron from lake ores, without excessive use of scrap and burning an average grade of by-product coke, having, say, 11 per cent ash, 1 per cent sulphur, and satisfactory physical characteristics, the production will be in the neighborhood of 135,000 cu. ft. of gas per ton of pig iron.

Under such conditions the thermal quality of the gas will be about 90 B.t.u. per cu. ft. The total fuel value of the gas produced in making a ton of pig iron will be therefore about 12,150,000 B.t.u., which is slightly more than one-half of all the heat which can be developed by complete combustion of the coke charged in the furnace, and approximately one third of the heat value of the coal charged in the ovens. It will have a heat value equivalent to that of about 900 lb. of good bituminous coal.

The daily gas output of a 1000-ton furnace will have a fuel value equivalent to that contained in about 450 tons of coal. The most economical and advantageous utilization or disposal of such a large and valuable amount of by-product fuel has become one of the most important factors in the determination of pig-iron costs, and to this fact may be attributed the keen and widespread interest which has been taken in finding uses for blast-furnace gas which will yield a greater credit against production charges, and a realization of the necessity for greater efficiency in all gas-using devices.

Of the total gas produced a sufficient amount must first be reserved for heating the furnace blast to the highest temperatures which specific conditions will permit the operators to use. These will vary within a range of from 1000 to 1600 deg. fahr., the physical and chemical quality of the raw materials being the chief limiting factors. Herein lies the most common opportunity of gas conservation, for the increasing of stove capacities and stove efficiencies has lagged behind the movement toward larger furnaces with the result that in many cases too much gas is being used for this purpose. Not infrequently as much as 35 per cent of the gas produced is required for the stoves, whereas recent developments in stove design have made it feasible to meet the most exacting demands with a maximum of 25 per cent.

Another 30 per cent, approximately, can always be used for generating steam and power required for the operation of the furnace itself. This is often, however, its least profitable use. If the by-product coking process is employed and there are better uses for coke-oven gas, blast-furnace gas can be burned under the ovens. The furnace-coke requirements alone will absorb about 25 per cent of the blast-furnace gas. If an open-hearth steel plant and rolling mills are included in the picture, from 20 to 45 per cent can be used there, mixed with coke-oven gas as a substitute for producer gas or other fuel. For isolated furnaces the generation and sale of electric power is sometimes the only commercial outlet.

In Table 1 an attempt is made to picture an ideal distribution of the by-product gases incident to the production of a ton of basic pig iron at a blast-furnace and coke plant operated in conjunction with an open-hearth steel works and rolling mill. This is a purely theoretical set-up, but based upon results obtainable in practice with efficient equipment. The following assumptions are used:

TABLE 1 PRODUCTION OF BY-PRODUCT GAS INCIDENT TO THE MAKING OF ONE TON OF BASIC PIG IRON AND A SUGGESTED DISTRIBUTION TO OPEN-HEARTH STEEL WORKS AND MILLS

		B.t.u. content	Percentage of total B.t.u. in—Coal	Coke-oven gas	Coke	Blast-furnace gas
1	Coal charged in ovens	37,950,000	100.0	...	..	...
2	Coke charged in blast furnace	23,100,000	60.8	...	..	...
3	Tar at 9 gal. per ton	2,080,000	5.5	...	..	...
4	Motor benzol at 2.5 gal. per ton	338,000	0.9	...	...	...
5	Coke-oven gas at 11,000 cu. ft. per ton	8,350,000	22.0	100.00	..	...
6	Domestic coke	1,600,000	4.2	...	..	...
7	Coke braize	1,600,000	4.2	...	..	...
8	Unaccounted for losses	882,000	2.4	...	..	...
9	Blast-furnace gas	12,153,000	...	...	52.5	100.00
10	Blast-furnace gas to underfire coke ovens	3,095,000	...	...	13.4	24.48
11	Blast-furnace gas to stoves	3,040,000	...	...	13.2	25.00
12	Unaccounted for coke-oven gas	417,000	1.1	5.00	..	...
13	Unaccounted for blast-furnace gas	608,000	...	...	2.6	5.00
14	Coke-oven gas left for open hearth	6,140,000	16.2	73.50	..	...
15	Blast-furnace gas to open hearth	2,390,000	...	...	10.3	19.70
16	Coke-oven gas to shops and misc	625,000	1.6	7.50	..	...
17	Blast-furnace gas to soaking pits	2,920,000	...	...	12.6	24.02
18	Coke-oven gas to finishing mills	1,168,000	3.1	14.00	..	...
19	Blast-furnace gas to finishing mills	97,000	...	...	0.4	0.80

1800 lb. of furnace coke produced from a charge of 2814 lb. of coal at the ovens corresponding to a yield of 64 per cent.

11,000 cu. ft. of 540 B.t.u. coal gas produced per ton of coal charged.

1800 lb. of coke used per ton of pig iron produced.

135,000 cu. ft. of 90 B.t.u. blast-furnace gas.

Using Table 1 as a basis, a gas-distribution chart (see following page) has been drawn to show more clearly the relationship between the fuel and metal balances for such an operation.

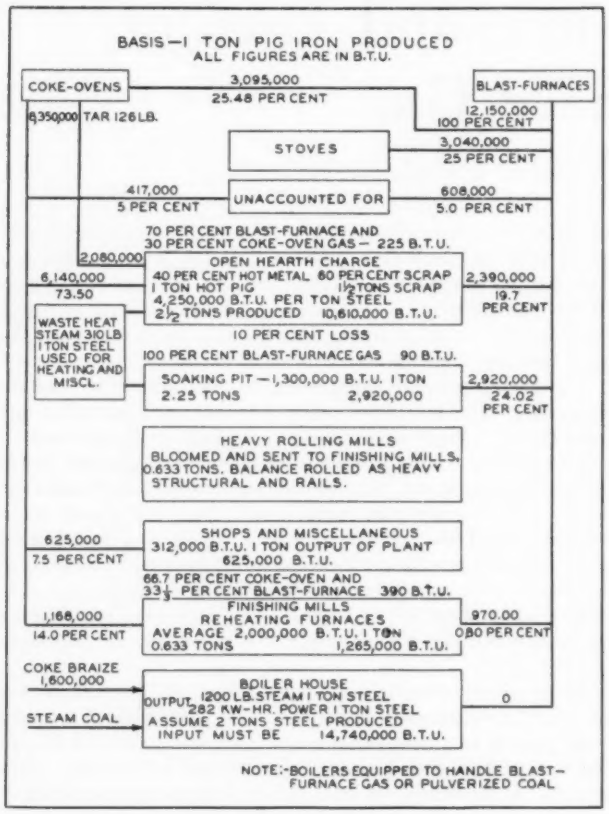
The most profitable disposal of the gas can only be decided upon after a comprehensive study of the commercial possibilities which exist for the sale of coke, coke-oven gas, and power. Against contracts or the possibility of contracts for the sale of these commodities on a profitable basis must be weighed the value of the gas

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.



GAS DISTRIBUTION CHART

as metallurgical fuel in the steel plant. This value in turn will vary greatly. It will be highest when the price of fuels which it may be used to replace is high. If the coal or other fuels are sufficiently low in price it may be very difficult to establish a case for blast-furnace gas from the standpoint of return on investment, notwithstanding the well-justified claims that it is a superior fuel for metallurgical uses.

The expenses involved in the cleaning and distribution of blast-furnace gas are considerable both from the standpoint of the equipment required and its operation. There is a lack of published data bearing on this phase of the subject which deserves the attention of operating men who are in a position to supply it. Reports on the cost of washing gas to various degrees of cleanliness have been anything but uniform and have been meager in detail, due, no doubt, in a large measure to the comparatively small number of installations in this country where "final cleaning" is practiced. This is true to an even greater extent as regards the cost of gas distribution.

With this thought in mind the authors have prepared for this paper, from their own records and from the most reliable information that could be obtained elsewhere, a comparison of the cost of cleaning gas in several types of washers now in use or being developed. They have also combined these costs in another series of figures with gas-distribution costs obtained in a similar manner in an effort to work out in a general way the net credits which will accrue to the furnaces after cleaning and distribution expenses are met. As a basis for the latter determination it has been necessary, of course, to assume an arbitrary range of prices for bituminous coal and of values for coke-oven gas. It is hoped that these calculations will furnish a basis for discussion which may bring out more extensive data along the same line.

In Table 2 is shown the cost of cleaning gas by several different methods giving the cleanliness expected in each case. The costs of final cleaning, i.e., from 0.15 to 0.008 grains per cu. ft. as accomplished either by disintegrators or Cottrell precipitators are shown in the table, but since the wet Cottrell method in this country is yet to be tried out on a large scale, all subsequent uses of the cost figure of final cleaning are on the basis of those obtained from disintegrator operations. No charge has been made for water in these costs other than the expense of boosting the water pressure at the washers. Make-up water for gas-washer pumps is in most cases the cooling water from the furnaces which, it is assumed, has already been charged against the operation of the latter.

All gas used for underfiring coke ovens or at open hearths and steel-heating furnaces must be cleaned to the dust content obtainable in the old-fashioned Theisens, in disintegrators or Cottrell precipitators. The question of the degree of cleanliness most profitable for use in stoves and boilers is a much disputed one, although the majority of operators in this country would probably desire gas for these purposes having not to exceed 0.2 grains per cu. ft. Until recently it was generally considered that the expense of cleaning finer than this was not justified except for gas-engine use.

Assuming that 25 per cent of the total gas produced is used in the stoves, Table 2 will show that the cost of washing gas for stoves to 0.15 grain per cu. ft. amounts to 1.179 cents for each ton of pig iron produced, and for final cleaning of this much gas

TABLE 2 COST OF CLEANING BLAST-FURNACE GAS

Type of cleaner	Tower washer	Two-stage McKee	Feld McKee	Disintegrator	Wet Cottrell	Disintegrator plus tower	Wet Cottrell plus tower
Cleanliness expected, grains per cu. ft.	0.3	0.15	0.10	0.008	0.008	0.008	0.008
No. units and capacity in cu. ft. per min.	1-75,000	1-75,000	1-75,000	1-30,000	1-40,000		
Investment costs							
(a) Cost complete	$50,000	$50,000	$70,000	$30,000	$55,000		
(b) Int. and dep. on above per day at 13 per cent	$17.80	$17.80	$24.90	$10.70	$19.60		
(c) Cost per 1000 cu. ft., cents	0.0165	0.0165	0.0231	0.0248	0.034	0.0413	0.0505
(d) Cost per ton pig iron (135,000 cu. ft. per ton), cents	2.23	2.23	3.11	3.34	4.59	5.58	6.82
Operating costs							
(a) Water g.p.m.	1665	1500	1250	250	70		
(b) Power to run (no boosting), kw.	0	37	90	200	20		
(c) Power required to pump water, kw.	62	54	45	5	1.25		
Total, kw.	62	91	135	205	21.25		
(d) Cost per day at 0.8 cent per kw-hr.	$11.90	$17.47	$25.90	$39.40	$4.08		
(e) Cost per day—lubricants	0.20	0.22	0.66	0.33	0.05		
(f) Cost per day—attendance	1.00	1.00	2.00	4.00	2.00		
(g) Cost per day—repairs and maintenance	1.00	1.00	2.74	2.00	1.00		
Total (items d-e-f-g)	$14.10	$19.69	$31.30	$45.73	$7.13		
Overhead 10 per cent on items (e-f-g)	0.22	0.22	0.54	0.63	0.31		
Total operating cost	$14.32	$19.91	$31.84	$46.36	$7.44		
Cost per 1000 cu. ft. gas, cents	0.0133	0.0184	0.0295	0.107	0.0129	0.1203	0.0262
Cost per ton pig, cents	1.794	2.484	3.98	14.45	1.741	16.24	3.535
Total cost—operating plus investment							
Cost per 1000 cu. ft., cents	0.0298	0.0349	0.0526	0.132	0.0469	0.1618	0.0767
Cost per ton pig, cents	4.02	4.714	7.08	17.80	6.33	21.82	10.35

the charge will be 5.455 cents per ton of pig iron. In other words, it costs 4.276 cents more per ton of pig iron to use final-cleaned gas than it does to use rough-washed gas. Whether savings will accrue from the use of final-cleaned gas in the stoves which will justify this additional expense is debatable. The cleaner gas makes for better operating conditions by the elimination of sludge in the mains and burners, but except for these features satisfactory operation and a high degee of stove efficiency have been obtained with gas of moderate cleanliness.

If rough-washed gas is to be used, a primary cleaner of the two-stage McKee type will supply gas of greater cleanliness than ordinary tower washers, and at the same time reduce the load on the secondary cleaners with a consequent saving in power as compared with that which would be necessary were only tower washers used for primary cleaning.

In explanation of Table 3 it must first of all be pointed out that no attempt has been made to evaluate the fuel economies which may attend the substitution of blast-furnace gas for other fuels. In the application of mixed gases to open-hearth practice this fact is of great importance. Remarkable reductions in fuel consumption are said to have followed this innovation at several plants.

TABLE 3 EFFECT OF CLEANING AND DISTRIBUTION EXPENSES UPON VALUE OF BLAST-FURNACE GAS AS COMPARED WITH STEAM COAL, PRODUCER GAS, AND COKE-OVEN GAS

		(1)	(2)	(3)	(4)	(5)	(6)	(7)	(8)
					Cost of cleaning and distribution				
Purpose for which gas is used		Cost of coal	Cost of coal fired	Gross value gas before cleaning and distribution	Primary cleaning 0.15 grain per cu. ft.	Final cleaning 0.008 grain per cu. ft.	Distribution including boosting	Total	Net comparative value after deduction of cleaning and distribution expenses
		Dollars per ton	Dollars per ton	Cents per M. cu. ft.	Cents per M. cu. ft.	Cents per M. cu. ft.	Cents per M. cu. ft.	Cents per M. cu. ft.	Cents per M. cu. ft.
Primary washed gas as substitute for steam coal of 13,000 B.t.u. value	289,000 cu. ft. blast-furnace gas, 90 B.t.u. = 1 ton coal	2.50	3.00	1.039	0.0349	....	...	0.0349	1.004
		3.50	4.00	1.384	0.0349	....	...	0.0349	1.349
		4.50	5.00	1.729	0.0349	....	...	0.0349	1.694
Final cleaned gas as substitute for steam coal of 13,000 B.t.u. value	289,000 cu. ft. blast-furnace gas, 90 B.t.u. = 1 ton coal fired	2.50	3.00	1.039	....	0.1618	...	0.1618	0.877
		3.50	4.00	1.384	....	0.1618	...	0.1618	1.222
		4.50	5.00	1.729	....	0.1618	...	0.1618	1.567
Final cleaned gas as substitute for producer gas. Gas coal at 13,500 B.t.u.	255,000 cu. ft. blast-furnace gas, 90 B.t.u. = 1 ton coal fired	3.00	4.47	1.753	....	0.1618	0.536	0.6978	1.055
		4.00	5.47	2.140	....	0.1618	0.536	0.6978	1.442
		5.00	6.47	2.537	....	0.1618	0.536	0.6978	1.839
Final cleaned gas as substitute for coke-oven gas	6 cu. ft. blast-furnace gas = 1 cu. ft. of coke-oven gas at 540 B.t.u.	CO gas at $0.10 per M. cu. ft.			1.667	0.1618	0.536	0.6978	0.970
		CO gas at $0.20 per M. cu. ft.			3.333	0.1618	0.536	0.6978	2.630
		CO gas at $0.30 per M. cu. ft.			5.000	0.1618	0.536	0.6978	4.303

The item, designated as "Cost of coal fired" (Item 2), represents the delivered cost of coal plus 50 cents per ton of coal for boiler-house expense in the case of steam coal or plus $1.47 per ton conversion charges in the case of producer coal. The latter charge is made up of operating expense or cost above coal, estimated at $1.255 and $0.215 for interest and depreciation charges. The latter are taken at 13 per cent on an investment of $29,000 for each producer of 4000 lb. coal per hour capacity.

Item 6 is an estimate of interest, depreciation, and operating charges on the distribution system which will include gas mains, gas holders, boosting equipment, mixing devices, etc. The rate is at 13 per cent for interest and depreciation plus 2 per cent for operating expense on an investment which is taken at $17,850 per M. cu. ft. per min. capacity. It also includes a charge for boosting the gas pressure after washing of 0.0256 cent per M. cu. ft.

Although this cost of distribution seems high it should be kept in mind that in many cases the point of consumption is remote from the furnace plant. The estimates given are for a completely equipped installation and are substantiated by actual costs from one or two of the few plants which have been built. As there is likely to be a wide variation in distribution costs among different plants due to their topography, arrangement, and size of units, no exact figures can be given for the cost of distribution at any given plant without first making a careful survey of the property and a study of its requirements.

Table 3 will confirm the generally known fact that by far the most profitable use for blast-furnace gas is for underfiring coke ovens, provided a reasonably good price for coal gas can be obtained. Unfortunately many of the old by-product ovens are not equipped to burn blast-furnace gas and the change-over in such cases will probably await the normal rebuilding period unless the demand and the prices obtainable for coke-oven gas are unusually good.

It will be noted that the cost of final cleaning for boiler gas is 0.127 cent per M. cu. ft. higher than for rough-washed gas. Increased boiler efficiency or decreased boiler operating costs of about 13 per cent would seem necessary to justify this procedure.

As a substitute for producer gas the justification must come largely from the reduction in fuel consumption which has been briefly mentioned before or in the betterment of metallurgical practices. This subject has been covered recently in technical papers by others having first-hand information. The authors will attempt, however, to summarize briefly some of the reports from operators who have used blast-furnace gas in open hearths, heating furnaces, and under coke ovens.

Within the past two years there have been a number of large-scale applications to coke ovens, open hearths, and heating furnaces with uniformly good success from an operating standpoint. In the modern combination regenerative-type coke ovens it is asserted that the longer and less-intensive flame of blast-furnace gas will accomplish more uniform heating of the oven walls. Furnace men using coke from such ovens have testified to its excellent quality. Straight blast-furnace gas has also been used as a direct substitute for producer gas in soaking-pit furnaces. It is also obviously suitable for direct use in box annealing furnaces or other types of furnaces where comparatively low temperatures are sufficient.

Mixed with varying amounts of coke-oven gas, blast-furnace gas has been a pronounced success as a fuel for open-hearth furnaces and for heating furnaces generally. The results obtained in the open-hearth departments of the Steel Company of Canada and the Wisconsin Steel Company have been particularly interesting, inasmuch as in one case the air only is preheated, while in the other both air and gas are passed through the regenerators. There are advantages and disadvantages in both of these methods which have been widely discussed in technical papers, with no lack of advocates for either. Preheating of the gas seems chiefly desirable to produce luminosity of the flame which it accomplishes by the breaking down of the hydrocarbons which are contained in the coke-oven gas. Luminosity is useful in working the heat both as a visible indication to the melter of the furnace temperatures, and also, it is claimed, as an aid in transferring heat to the bath. It is said to be a deterrent of foaming and consequently of particular advantage in working heats in which a high percentage of pig iron and ore are used. The opponents of gas preheating discount the advantages and necessity of luminosity and

contend that they are outweighed by the safety and simplicity of a system in which gas preheating is entirely avoided. If luminosity is desired, it can also be secured by the addition of a small amount of tar.

For the purpose of this paper, it is sufficient and significant to note that remarkably good results have been obtained by both methods, not only in fuel consumption but in tonnage output, quality of steel, and life of refractories. Less than 5,000,000 B.t.u. per ton of steel has been consistently accomplished. The thermal value of the gas may be varied more or less, but generally averages about 225 B.t.u. per cu. ft., corresponding approximately to a mixture of 30 per cent coke-oven gas and 70 per cent blast-furnace gas.

In heating-furnace practice equally satisfactory performance has been reported from the use of mixed gases, and the advantages that were to be expected from having a gaseous fuel adjustable in heating value and flame characteristics to the requirements of each kind of steel to be heated have been borne out in actual practice.

The generally satisfactory results which have attended the use of blast-furnace gas in the steel plant are due largely to the fact that it is ideally fitted for mixing with coke-oven gas; its thermal value being about 90 B.t.u. per cu. ft. and that of coke-oven gas about 540 B.t.u., their combination in varying proportions will cover the range of flame temperatures which is needed for most metallurgical purposes. The large quantity of inert constituents in blast-furnace gas make it particularly useful where mild soaking temperatures are needed, while the admixture of the richer coke-oven gas in the proper amount makes it possible to increase the intensity and cutting quality of the flame to any desirable extent.

This complete or partial substitution for coke-oven gas has the further advantage of conserving the supply of the latter and making it available for a much wider application in the steel plant than has formerly been possible. In the new steel plant which the authors' firm is designing for the Russian government, in which the production of steel will be about equally divided between the Bessemer and open-hearth processes, it is expected that the firing of coal will be unnecessary except for generating about 57 per cent of the plant steam requirements.

Where the extension of the use of blast-furnace gas for existing plants is under consideration, the cost of adapting the coke ovens and furnaces to the use of the new fuel must be taken into account, and, together with the expense of cleaning and distributing the gas carefully, compared with the economies which may be obtained. The latter, as indicated, will depend upon the cost of other fuels, the possibility of profitable sale of coke and coke-oven gas, and the improvement in metallurgical practices that may follow. But it is safe to predict that in many American plants where surplus blast-furnace gas is now being used exclusively in boilers, a careful study of all the conditions will reveal that it can be applied in a large measure to other purposes with a substantial return upon the investment required for making the change.

Discussion

F. H. Willcox.[3] This excellent paper is suggestive of measures that the steel industry will have to put into effect. The industry today is based on production. Since 1910 it has grown from about 49,000,000 tons of pig and ingots to 94,000,000 tons of pig and ingot output. Potential capacity is probably 120,000,000 tons of pig and ingot per annum. It is an industry of small annual turnover and high capital expense. The cost of installing steel-plant capacity is nearly twice as much today as it was 25 years ago. Then, 1.9 tons of ore and 1.7 tons of coal cost $9.30, as against $11.70 today. Pig iron sells for about the same price now as then. However, bars sold for $26 to $28 in that period, as against $36 to $38 today, and the same ratio obtains in rods, structural, plate, billets, etc. The proportion of cost in materials, labor, and supervision has changed little in this period compared with capital charges, which have doubled. It is difficult to escape a feeling that the steel industry in the last 20 years has installed units and capacity geared to the output and prices of boom times, and not too well adapted to the 70 to 85 per cent average production year nor to export of semi-finished material. It is also possible that as the size of units has grown, and as the cost of the plant per ton actually produced per year has doubled, the ratio of money spent for brains in analysis and promotion of better costs and practice, with less capital, material, and labor waste, has not kept pace. In spite of better wages than were paid 20 years ago, the standard of wages and living for men in the steel industry is not what it should be compared with men similarly employed in transportation, building, or general commerce, not to mention luxury trades.

The industry being one with this background of development, of high capitalization, with moderate profit and wages, and finally, of no net result in the lowering of prices that must be charged, it is worth inquiring whether money could not be profitably spent for a greater degree of what one may call "general staff" work. The paper mentions a field earlier suggested by Cutler, Leahy, and others, but few plants on this continent are actually doing work in this line. It concerns not only fuel economy, but faster melting and heating, better practice on yield, and improved uniformity of product and acceptance. Analysis of costs has made surprising disclosures, almost warranting the guess that while total costs, or production costs, are known, too few know the actual detail structure or cause of costs, from ore or coal yard to delivery of steel to customer. Adoption of equipment is not infrequently decided upon on the basis of the best type of sales effort or use elsewhere under different conditions of application.

Transportation, maintenance methods, application of more durable and suitable materials, and even time studies are profitable fields. Undoubtedly too much money has been spent in super-power plants in steel works.

To summarize, the writer is inclined to think that study along the lines of these fields is more in order than the 20-year policy of bigger units, more output, and increased capital costs. The next 20 years, if not 10, should somehow yield more in the direction of higher pay for the men in the industry and lower rather than higher prices for the product, both export and domestic.

T. A. Lewis.[4] The paper presents very clearly the heat balance of a typical steel plant which, in turn, has led on to the discussion of very pertinent problems on blast-furnace gas. Blast-furnace gas, as indicated in the paper, is available in very large quantities and must be handled as quickly as generated, unless tremendous storage capacity is available. In this country, this gas has usually been consumed in blast-furnace stoves, boilers, and gas engines. In recent years we have been giving more thought to further utilization of this fuel for better conversion values. The development of this low-grade fuel has come along only recently in this country, as hitherto the economics of the situation was not sufficiently pressing to force the issue. However, the developments and economies derived in recent applications have proved their justification, with the result that the distribution of this fuel is being extensively made in many plants.

Each plant must be studied carefully to determine the economic

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use. A heat balance must be made, and this in turn must be balanced with the return on the investment cost of the total system required to utilize this fuel. Blast-furnace gas has proved its value in open hearths, coke ovens, soaking pits, and reheating furnaces, as well as in many other types of furnaces; consequently, this fuel can replace fuel that is usually higher priced, as well as eliminating the conversion cost (such as gas producers with high cost of gasification).

The uses of blast-furnace gas involve cleaning and distribution costs. The cost of cleaning is dependent upon the degree of cleanliness desired, which in turn determines the type of cleaner to be used. The cost of distribution is dependent upon the uses, as well as distances of consuming departments from the blast furnace.

In the general distribution, in the authors' paper, approximately 30 per cent was alloted for generating steam and power for the operation of the furnace itself. With the developed uses of blast-furnace gas, this figure is too high, as no more than 14 to 20 per cent should be used. The lower figure of 14 per cent is for the gas-engine blowers and the 20 per cent for the steam (modern) units. Undoubtedly the use of blast-furnace gas in the coke ovens is very desirable, as in turn this releases a rich coke gas which can be distributed easily and cheaply because of its high calorific value and its relative low density. Furthermore, this gas is especially desirable on the higher temperature operations in the steel plant, or it can be sold at a fair profit to the public gas companies. The domestic demand for coke-oven gas is increasing yearly. This is economically sound, and the use of coke-oven gas for such purposes should be encouraged.

After the stoves and blowing units have consumed their requirements, over one-half of the blast-furnace gas is available for other purposes. The desirable units to which it should be applied are open hearths, coke ovens, soaking pits, miscellaneous furnaces, and boilers.

The authors have drawn up, in a concise form, a cost sheet of blast-furnace gas cleaning by various types of cleaning equipment. A pertinent question is brought out as to the justification of going to the additional expense in cleaning gas better than 0.15 grain per cu. ft. From the table presented there is a differential price of 4.276 cents per ton pig iron more for cleaning the gas from 0.15 to 0.008 grain per cu. ft. Study of cost sheets indicates there is justification of this additional expense. We do not have sufficient data in this country as we have not had clean gas long enough to prove the point. However, we can be guided by the practice of Europe, which has had a fairly good backlog of experience with the cleaner gas. The more experience most plants have had with clean gas, the greater is the demand for even cleaner gas. The efficiency of stoves is somewhat higher with narrow checkers than standard checkers with the same heating surface. More heating surface can be stored in unit volume with narrow checkers, which means fewer stoves per furnace and, consequently, added efficiency and less equipment.

For the most economical furnace operation, it is desired to have the highest stove line heats that the charge will permit. Consequently the stove heats will be raised. The life of the stove is influenced by the temperature in the stove and the cleanliness of the gas. The dirt in the gas, particularly the alkaline salts, fluxes with the brick of the stoves, particularly in the combustion zone, dome, and first section of the checker chamber, thereby reducing the fusion point of the brick, which in turn causes disintegration and slagging. This means higher maintenance and more frequent periods of shutdown. Furthermore, dirty gas carries mechanically entrained moisture in excess of saturated clean gas. This means lower efficiency. For stoves with small checkers the gas should be cleaner than 0.02 grains per cu. ft. The higher heats of blast should mean even greater cleanliness. Stove costs for long periods would warrant the advantages of the higher-priced clean gas, particularly where the gas saved could be used to replace fuel elsewhere. Since there is a greater demand for better stove practice, more care is given to better design. With the advent of narrow checkers and clean gas, designs have been developed that will give the desired velocities at the various sections of the stove. There are many ways of accomplishing this. The design with changing sizes and openings is usually termed the zone system. This zone system adapts itself to the physical loss of heat transmission in the upper zone to radiation and to convection in the lower zone.

The radiant-heat transmission increases with the thickness of the gas stream (in B.t.u. per sq. ft. per hr.), which means wide openings at the top; while the convection-heat transmission increases with the velocity due to turbulence, which means a larger number of small openings. Heat penetration is a function of temperature; therefore the thickness of the upper part is greater than the lower colder part. This is comparable with the practice in modern boiler design. In the straight-tube boiler the lower banks of tubes have a large tube diameter with a wide pitch, while the upper cold banks have small diameter tubes with narrow pitch.

As pointed out by the authors, no attempt was made to evaluate the advantages of fuel economies of clean blast-furnace gas over other fuels. This is a very important point with respect to the application of mixed gas for metallurgical furnaces. The operation of a mixed gas furnace or partly mixed gas with producer gas, in comparison to a straight producer gas furnace, is very pronounced. The soot formation in a producer-gas furnace reduces the life of the furnace and makes it considerably less efficient as becomes older. The clean mixed gas application gives longer furnace life and sustained fuel economy. This combined with the elimination of the producer conversion costs will go a long way to justify the use of blast-furnace gas and its attendant cleaning and distribution costs. In the case of underfiring coke ovens, it is essential that this gas be very clean as it is expected to have ten or more years of uninterrupted service.

In the case of soaking pits and other smaller furnaces, the fine degree of cleaning that is needed in coke ovens is not so essential, although the required degree of cleanliness is determined by the layout of the distribution system.

Reference has been made to different methods of application of mixed gases in the open hearth; namely, preheating of both gas and air as one method and only preheating of air for the other scheme. With reference to the type where the coke-oven gas is preheated to a degree sufficiently high for decarbonization of the hydrocarbons for luminosity, great care should be used in cooling down the blast-furnace gas to reduce the water vapors that otherwise would be carried over and mixed with the mixed gases. The effect of luminosity would be neutralized by the dissolving of the C particles by water vapor.

$$C + H_2O = CO + H_2$$

These conditions have existed, and the differential luminosity has been made up with liquid fuel.

Reference has been made to the advantages of the substitution of blast-furnace gas for coke-oven gas. This does enable a plant to reduce the outside purchased fuel. The ideal steel plant, from a fuel viewpoint, is the one which confines to the coke ovens all the fuel necessary to operate the plant. This is done in some plants. However, the character of the plant determines how far this can be accomplished.

FEODORE F. FOSS.[5] It was said that the world does not need to be so much informed as to be reminded. From this point of view the paper is valuable. It seems that American iron-masters should be constantly reminded that the fuel utilization in their plants is far from being ideal. In very many plants at least it has not yet reached a more or less satisfactory level. If we take as a basis of production sheet bar or rail and calculate fuel consumption per one unit of this product, we will find that after having delivered to coke furnaces 1.35 tons of coal (13,000 B.t.u.) we should operate the plant only on gases from coke ovens and blast furnaces, without any additional fuel. Such results have been obtained already on practice by several German plants producing open-hearth steel. Plants equipped with basic Bessemer converters lower fuel consumption to one ton (and even a little less) of coal per one ton of finished product. In its endeavor to eliminate use of coal from all intermediate operations one large steel company near Duisburg went so far as to introduce electric traction on all railroads of their plant, receiving electric current from central power house. Nearly the same ratio of fuel consumption is obtained in best Luxemburg, Belgian, and some French steel plants recently acquired from Germany. Older French plants use from 2.5 to 3.5 units of fuel per ton of finished product. Most up-to-date American plants show fuel consumption of about 2 to $2^1/_2$ units per one unit of finished product, and the average for all plants seems to exceed 3 units. The worst results are observed in Great Britain, where a figure of $4^1/_2$ to 5 units is encountered. We know that there is a great lack of integration in the British steel industry.

All that confirms the rather old idea on what constitutes the most profitable unit in the art of steel making. A long time ago it was shown that a unit of the iron industry should be understood as a plant, which begins with coke ovens, ends with rolling mills, and uses coal only for making coke. The skill of iron-masters and plant designers should be demonstrated in their ability of balancing out productive units in such a way as to achieve the best fuel utilization and an uninterrupted production. An explanation for the lack of attention paid by American iron men to the proper utilization of fuel could be found, possibly, in the fact that coal in this country is so abundant and cheap. On the average it is twice as cheap as in continental Europe. Nevertheless, the time will come, and it seems to be very rapidly approaching, when every component item of steel costs will be scrutinized and analyzed with regard to the possible economies.

Having that in mind, we should pay our utmost attention to blast-furnace gas, its quality, and uses. The authors say, "If the coal or other fuels are sufficiently low in price, it may be very difficult to establish a case for blast-furnace gas from the standpoint of return on investment, notwithstanding the well-justified claims that it is a superior fuel for metallurgical uses." There is no doubt, as they assert further, that "The most profitable disposal of gas can be decided upon only after a comprehensive study of the commercial possibilities which exist for the sale of coke, coke-oven gas, and power." But, as already said, no other fuel other than blast-furnace and coke-oven gas should be present in the modern steel plant. Therefore all our studies should be directed to the ways of utmost utilization of available gases and elimination of all other fuels which have to be bought. There have been several studies made of evaluation of blast-furnace gas based on B.t.u. content of gas and coal. It was shown that if the blast-furnace gas is used in gas engines with an electrical generator attached without waste heat boilers, the value of gas will be from 1.3 to 1.4 higher than the respective coal. If we have a gas engine for electrical generation with waste heat boilers, the value of blast-furnace gas will be raised to 1.5 to 1.6. If we turn blast-furnace gas into the gas-blowing engine with waste-heat boilers, the value of gas will be from 1.15 to 1.5 higher than the coal. In the heating of coke ovens the value of blast-furnace gas is as high as 1.8, and in open-hearth furnace mixed with coke-oven gas it gives the value from 1.5 to 1.8. Even under boilers blast-furnace gas, if supplied constantly, shows the preference above coal to the extent of 1.1 to 1.4.

Blast-furnace gas, as we all know, is dirty, and the cleaning of this gas deserves much more attention than has been given to it heretofore. In olden days and in some plants at the present time it was customary to divide the gas, with reference to dust content, in two different grades—one a clean gas or "machine" gas and another "furnace" or stove gas. It seems that with the view of attaining high efficiency from all the apparatuses of metallurgical plants we have to look at our stoves and furnaces as on precise and delicate machinery and give them fuel in the form most suitable and most profitable for operation. Only then can we expect high efficiency from our equipment. Applying this principle, some blast-furnace managers have achieved very high efficiency of heating stoves and use only from 25 to 20 per cent of blast-furnace gas for one ton of pig iron. In such cases stove checker-work was protected from excessive dust and from slagging of brick surfaces by alkaline particles contained in the blast-furnace gas dust. Very high thermal efficiency of stoves resulted, and the cost of stove cleaning became practically negligible.

Turning now to the cost of cleaning blast-furnace gas as shown on Table 2, we find that this table is rather incomplete. Data relating to some other systems of cleaning are lacking. Results of dry cleaning by using the Halberg-Beth process, and particularly by modern electric cleaning processes, are not given. It should be mentioned also that the figures of the gas cleanliness shown in the table are qualified as "expected" and not as "actually" obtained. We know that these two definitions often differ very markedly. Studying the results of different gas-cleaning methods, we find that by applying different methods of dry cleaning we may achieve a rather low cost with very high degree of gas cleanliness and a very low moisture content in the gas. So far as the writer knows, the electric gas-cleaning plant at Witkowitz Steel plant in Czechoslovakia, reducing the dust content in a cubic foot of gas from 1.92 grains to 0.002 grains, gives the cost of cleaning only as 0.08 cent per 1000 cu. ft. Electric installation costs a little higher than wet disintegrators and is quoted to be about $26 per 1000 cu. ft. per hour, whereas disintegrators cost about $21. It seems that when the demand for the *real* clean gas will come, the attention of blast-furnace men will be turned from the tower washer and the like to the disintegrator and particularly to electric cleaning.

With the results shown in Table 3 the authors contradict what they have said before in reference to the evaluation of gas and show that the most profitable use for blast-furnace gas is for underfiring coke ovens. They actually confirm the fact, which has been explained above, that blast-furnace gas is the real fuel for metallurgical processes and has its definite value in preference to coal and producer gas.

Further on the authors cite several examples of very economical use of blast-furnace gas in combination with coke-oven gas for the open-hearth process, showing that less than 5,000,000 B.t.u. per ton of steel has been constantly obtained. These results, again, confirm the opinion that if proper attention is paid to the quality of blast-furnace gas, ideal results of fuel utilization in the steel plants can be obtained.

J. F. SHADGEN.[6] Particular attention should be called to

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[6] Combustion Engineer, Smoot Engineering Corporation, New York, N. Y. Mem. A.S.M.E.

combustion control. There are two kinds of controls on industrial furnaces: Human control supplied by customers, which has no visible capital charge, and machine control, which has to be installed and paid for.

Machine control (often erroneously called automatic control) has made great strides during the past decade, particularly in boiler rooms. Machine control has also been applied to metallurgical furnaces during the past few years. About 40 open-hearth furnaces and 13 heating furnaces are so far equipped with machine-control apparatus.

The results are most encouraging, as the increased fuel efficiency proves, and greater progress will be made during the next years as the experience accumulates.

Gas mixing is usually done by machine regulation, and 15 installations are operating successfully, mostly mixing blast-furnace gas and coke-oven gas. Recently, the dilution of natural gas with a lean gas broadened the importance of the problem.

The writer would call for investigation of the applications of machine control, as results conclusively prove their economic value.

H. A. Brassert.[7] In planning the distribution of blast-furnace gas for any given plant, it is of first importance to establish accurately how much surplus blast-furnace gas can be made available. The authors state that recent developments in stove design have made it feasible to meet the most exacting demands with a maximum of 25 per cent of the total gas. Our experience with stoves employing the graduated zoning principle shows that this maximum has been reduced to very much lower figures, in some cases as low as 15 per cent. Furthermore, as Mr. Lewis has pointed out, the assumption of 30 per cent of the gas being used to generate the steam and power for the operation of the furnace itself is excessive whether turbo blowers or gas engines are used.

Substituting the lower figures for the stoves and other blast-furnace requirements, nearly twice the amount is available for the other departments and enhances the importance of the subject. In addition we must consider that a minimum of gas should be used under boilers if metallurgical applications are available for its consumption.

Just how far one can go in decreasing the consumption of gas on the hot-blast stoves depends largely on the cleanliness of the gas and the first cost and savings. In any event the saving in gas must be accompanied by the ability to heat the blast to the highest degree which it is possible to use. Fortunately modern stove construction has permitted us to heat the blast to higher temperatures than ever before, in spite of the low consumption of gas, by zoning of the checker column, a subject which has been so well explained by Mr. Lewis. The extent to which the checker openings can be decreased depends on the cleanliness of the gas and the type of impurities in the gas. From results obtained in American operation the writer is of the opinion that it is not necessary to clean the gas to the authors' figure of 0.008 in order to use the most efficient type of stoves, which combines the most effective velocity in the different zones with turbulence. The results of stoves having very small checker openings in the lower zones have shown that the increased velocity maintained therein keeps these narrow passages purged. The extremely fine deposits are thereby swept out, an action assisted by the turbulence created by the staggered fillers. Stoves of this type have been in operation for a period of several years operating on gas which averages 0.02 grain per cu. ft., and they have remained perfectly clean, as revealed by recent inspection of the checkers made possible through banking of the furnaces.

The cleaning of the gas below 0.02 grain requires a large amount of additional power, as it can only be accomplished by machines running at increased speeds, and the power is approximately a function of the cube of the speed. It is questionable whether stove designs should be of a type requiring much additional expenditure for power, the authors have shown, when a gas containing 0.02 grain or below can be delivered to zoned stoves of high efficiency at an expenditure in power of only a fraction of that given in this paper. In Germany stoves containing smaller openings than we have used in this country and so constructed that they cannot be inspected or blown out are considered to require gas of a cleanliness which can only be obtained by the bag-house system, which cleans the gas to 0.003 or less.

A gas with 0.02 grain per cu. ft. is also clean enough to be sent to open hearths and mill heating furnaces, particularly since experience has shown that by the time the gas is carried a considerable distance through pipe lines with bends and water seals, etc., the cleanliness at the delivery point is generally 30 to 40 per cent less than if measured immediately behind the eliminator following the disintegrator. Disintegrators are operating today which deliver from 50,000 to 60,000 cu. ft. of gas per minute with a cleanliness of 0.025 and less at an expenditure of from 50 to 70 hp. From this and from horsepower required by other machines which clean the gas to an average of 0.006, we can safely deduct that to average below 0.02 grain, it will not require over 125 hp. for 50 to 60,000 cu. ft. of gas per minute with Lake Superior ore practice and producing basic iron.

Furthermore, the power cost at the average steel plant runs lower than the figures given in the paper, and if actual costs were substituted, together with the lower horsepower, the comparison as between finely cleaned gas and gas from Feld or McKee washers, as shown by the authors to contain 0.1 to 0.15 grain, brings the cost of the disintegrator gas containing but a small fraction of the dust to only 50 per cent above the costs of the former gas, which cannot be used in high-efficiency stoves or for metallurgical purposes.

For coke-oven heating and gas engines, gas below 0.01 is required, but it is a question whether it is more economical to clean all of the gas to 0.02 and reclean the coke oven proportion separately.

There is no question as to the many advantages of using mixed gas or blast-furnace gas for melting and heating. The advantages are not only expressed in a saving of B.t.u. consumed, but also in the better control of the operations. The ability to regulate the B.t.u. content of the gas is of the greatest advantage, both in open-hearth practice and in heating. The combustion of blast-furnace gas results in a long, mellow flame. It is free from sulphur and its combustion can be proportioned to a much finer degree than that of richer gases requiring larger air-gas ratios. The writer was impressed in Germany with the high development of the art of heating and the much lower amount of scale produced due to the maintenance of neutral flame conditions in the reheating furnaces.

The blast-furnace gas has its greatest value in the finer uses, namely, coke-oven heating, melting and reheating, and its lowest value in raising steam. Boiler plants, even if especially designed for efficient use of blast-furnace gas, cannot obtain as high an efficiency of the same rating by the use of blast-furnace gas as can be easily be obtained from the cheapest solid fuels. At most plants operating conditions will require the bleeding of a certain amount of blast-furnace gas through the boilers, but this amount should be the surplus above all other requirements. The cost of gasifying the coke in the blast furnace has already been borne by the pig-iron production, and a very ideal gas, especially desirable for metallurgical purposes, has resulted. The boilers should be the last place to consume this

[7] President, H. A. Brassert & Co., Chicago, Ill.

gas, and the burning of gas under boilers should be considered only as an equalizing means for the plant system. The development of highly efficient automatic controls has given us practical means for the finest regulation of combustion in metallurgical furnaces, as well as means for automatical substituting of blast-furnace gas for solid fuel under the boilers. The writer is convinced that very large savings can be obtained in the industry with a fuller appreciation of the high value of blast-furnace gas for metallurgical use.

F. H. Willcox. Answering questions, it can be said that open-hearth practice abroad uses a steeper port, a faster velocity, and much more regenerative volume than we use here with producer gas or oil. They run faster heats than we do and have a faster turnover. Of course, open-hearth practice over there is based more on 20 per cent pig and 80 per cent scrap, which is easier than our 50–55 per cent hot-metal practice. One can satisfy one's self of 17 tons per hour rates throughout the month on 100-ton furnaces, of 3,200,000 to 3,600,000 heat consumption on straight runs, and 4,000,000 to 4,400,000 for a month's run.

As to the degree of cleanliness of gas, there is always a tendency to edge too close to what seems to be the toleration limit. That has been true in the case of stove gas, and we know that a toleration based on one blast campaign is a misleading and costly error when applied to a 16- to 25-year stove life. Practice on coke ovens and gas engines has demonstrated the need of the highest degree of cleanliness. The same will be found to be the case in other applications. The difference between 0.01 grain and under and 0.05 grain and over is extremely little in cost of equipment and cleaning, but tremendously great in effect on heating surface.

As to the effect of moisture on radiant properties of flame, it is not uncommon to see Halberger or Cottrell gas used at temperatures of 100 to 140 deg. fahr., without apparent adverse effect. This would be about 35 grains of moisture per cubic foot.

Geo. E. Rose.[8] The chairman having remarked that there had been considerable experimenting done at the Wisconsin Steel Works with the use of mixed gas (blast-furnace and coke-oven gas) on open-hearth furnaces and that there had been developed at this plant a movable port which had been described in a paper by Rose and Washburn of the Wisconsin Steel Works' organization and having asked for a statement of the experience with the use of mixed gas and the movable port construction at the company's open-hearth plant, the writer will give it.

We are quite well pleased with our progress in the use of mixed gas as the fuel on our open-hearth furnaces. Necessity has been the mother of invention in our case in this respect. We were confronted with the problem of making use of the coke-oven gas from an additional battery of coke ovens that we were about to start building in 1928, and which we now have built; therefore, we began to experiment with the use of blast-furnace and coke-oven gas on our No. 1 open-hearth furnace early in 1928. The writer was somewhat opposed to this proposition of trying to operate our open-hearth department with this fuel, and therefore got a trip to Europe out of it. As you know, in Germany they have been using mixed blast-furnace and coke-oven gas as fuel on open-hearth furnaces for the past 12 or 15 years. They have had much more experience with it over there than we have here. They emphasize very strongly the necessity of preheating the mixed gas up to 1000 to 1200 deg. cent., which is approximately 1800 to 2200 deg. fahr., so as to crack up the hydrocarbons in the coke-oven gas.

This cracking up of the hydrocarbons increases the luminosity due to the carbon particles in the flame. It is not the luminosity itself that is so much desired, but rather the transmission of heat by radiation from the carbon particles in the flame. This is a point that is not fully emphasized in open-hearth practice in this country. It is not so much the direction of the flame that is desired in an open-hearth furnace as the quality of the flame. In fact, the writer does not believe that a flame can be directed from the end of the furnace over a bath 40 or 50 ft. long so as to keep that flame hugging the bath, except by means of having the flame what it should be in quality and density. In other words, the important things are the quality and density of the flame rather than the mere directing of it from the end of a furnace 40 to 50 ft. long.

Now, if there can be a flame sufficiently laden with carbon particles (molecular carbon, as it were) derived from the cracking up of the hydrocarbons in the coke-oven gas, we get an increased transfer of heat in the open-hearth furnace proper by means of radiation from the carbon particles in the flame. We must not forget that the intensity of heat radiation from a given source varies inversely as the square of the distance from the source; hence the advantage of the nearness of the luminous flame to the bath as compared with the roof from a radiation point of view.

This explains why the old-time open-hearth operator frequently found it necessary to go back to the gas-producer house and have his gas man put fresh coal into the producers in order to get his furnace to go to work, as he would express it. He needed the soot particles formed by the addition of fresh coal to the producers. He could not get the necessary transfer of heat from the blue gas that the producers were making, even though the gas might have been quite high enough in B.t.u. value.

We feel that our experience confirms that of the German open-hearth operators, that it is very necessary for good results that the preheat of the mixed gas be carried up to around 2100 to 2200 deg. fahr., so as to give the desired luminosity to the flame, which simply indicates its great ability to transfer heat by radiation from the flame.

Now, of course, this does force one to speak of the movable ports, because without the movable ports it is quite impossible to have a port with a sufficiently restricted aperture to give the necessary jet velocity to the flame in the burning end of the furnace, and at the same time give a large enough opening of the gas downtake in the opposite end so as to get sufficient of the products of combustion back through the gas-checker chambers to bring the mixed gas up to the required preheat for the cracking up of the hydrocarbons and thence the luminosity and heat transfer by radiation.

The movable port moves into the furnace over the gas uptake on the burning end of the furnace, giving the flame the necessary jet velocity, but on the other end, the movable port is out of the furnace, leaving the full opening of the gas downtake, so that by manipulation of the reversing slide valves in the flues from the gas and air checker chambers, 50 per cent or more of the products of combustion can be taken down through the gas downtake and out through the gas regenerators, thereby making it possible to get a much higher preheat in the mixed gas than in the air, if desired by the open-hearth operator. At any rate, the preheat in the mixed gas can be brought up to an average of 2100 to 2200 deg. fahr. very readily.

As stated in the first place, the Germans have had fairly long experience with the use of mixed blast-furnace and coke-oven gas on their open-hearth furnaces. They have a high fuel cost, comparatively speaking; that is, a ton of coal costs about as much as three and one-half to four men's daily wages, whereas in our case here in the Chicago district, a ton of coal and a laboring man's daily wages are about equal; hence one can see the emphasis that is put upon fuel economy in a German steel plant

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and how they endeavor to make the best possible use of fuel. As a matter of fact, inasmuch as their major process of steelmaking is that of the basic bessemer (or Thomas), they necessarily make considerably more pig iron per ton of steel than we do here with our basic open-hearth process, in which we use about 50 to 50 pig iron and steel scrap. With the basic bessemer process the yield is about 90 per cent; or, in other words, a ton of pig iron will make about $^9/_{10}$ of a ton of basic bessemer steel, whereas with us, with the basic open-hearth process, a ton of basic iron will make almost two tons of steel, depending on the percentage of steel scrap used in the charge. But inasmuch as they make 60 to 70 per cent of their steel by the basic bessemer process, they necessarily have much more blast-furnace gas per ton of steel than we have in this country at plants where the basic open hearth is the major process of steelmaking. Even though they have much more blast-furnace gas per ton of steel than we have, still they clean their blast-furnace gas to a greater degree and to a finer point than we do and use it much more efficiently and thereby get more out of it, especially in view of the high cost of fuel in their country. However, it would seem as though it always would pay to clean it and use it efficiently rather than to waste it.

Of course, we do not deliberately waste it at many plants in this country, but we do fail to clean it properly at many plants and use it under low-pressure boilers and in stoves of large checker construction, in which cases we use almost twice as much as would be necessary for the same results if the gas were properly cleaned and if the stove and boiler construction was of the type and kind that can be used and should be used with clean gas. Of course, it undoubtedly is a fact that the return on expenditures for cleaning blast-furnace gas and the efficient use of it would not be as great in this country as in Germany, where the fuel cost is much higher, but as against the inefficient use of it in a steel plant where it is otherwise necessary to buy power or additional fuel, the writer believes it will pay in every case to clean the gas thoroughly and use it efficiently.

Authors' Closure

It is very gratifying to note the interest taken in this subject, and the authors are greatly indebted to those who have so generously and capably contributed to the discussion.

The Application of Metallic Recuperators to Industrial Furnaces

By G. D. MANTLE,[1] PITTSBURGH, PA.

Increasing cost of fuels and the growing pressure of competition make apparent the necessity of heat salvage in metallurgical furnaces. The paper asserts that there are few industrial furnaces that cannot be made 20 per cent more efficient by the installation of heat recuperators. A properly applied recuperator may result in an increased efficiency of 50 per cent. The actual value of heat salvaged from the waste gases by recuperation or regeneration is little appreciated by the average engineer.



THE forest of smokestacks in every industrial center would be sufficient evidence, even to those who were not familiar with modern manufacturing processes, that fuel is one of the prime economic factors of today. It is in fact the major cost item in many industries, and its more efficient use is one of the most serious problems facing the largest industries of this country. This is evidenced by the appointment during the last few years in practically every large industrial plant of specially trained engineers with efficient staffs whose sole duty it is to ensure the more economical use of the fuel used in manufacturing processes. The importance of this field can be realized when it is remembered that there are several hundred plants in the United States today whose fuel bill, exclusive of that used for steam-generating purposes, well exceeds $1,000,000 per annum. There are few metallurgical or ceramic plants of any importance whose fuel bill will not reach a minimum of $10,000 per annum.

There are three obvious means of increasing the efficiency of the average industrial furnace: (1) The salvaging of some of the heat in the waste gases by means of either a regenerator or recuperator, (2) the provision of automatic means to assure combustion of the fuel with as nearly as possible the theoretical quantity of air, and (3) the reduction of radiation losses by the use of insulating materials and by strict limitation of the sizes of doors and openings to the minimum required by the character of the operation performed in the furnace. In the general run of industrial furnaces, the magnitude of the saving to be effected will be in the order listed.

In view of the growing interest in combustion and in fuel conservation, it is a little strange that it should be necessary to advance any argument in favor of recuperation. The sheer necessity of heat salvage is made plain by the always increasing cost of fuels and the growing pressure of competition in all industries. Even those best posted in the design and operation of industrial furnaces will, however, often admit their inability to discuss in detail the economics of recuperation, or even to calculate correctly the value of the potential saving.

It will doubtless surprise the majority of engineers and operators to learn that there are few industrial furnaces which cannot be made 20 per cent more efficient by the application of a recuperator and that there are many cases in which an increase in efficiency of 50 per cent is assured by a properly applied recuperator.

The actual value of the heat salvaged from the waste gases by recuperation or regeneration is so little appreciated by the average engineer that no attempt will be made to analyze or discuss the various types of recuperator or regenerator.

Benefits of Recuperation

Combustion is simply the chemical combination of the elements of the fuel with the oxygen in the air. Every fuel, therefore, requires a definite quantity of air, varying with its analysis, for its complete combustion. For instance, the average natural gas requires approximately 10 cu. ft. of air to burn 1 cu. ft. of gas. In fact, the air requirements of the average gaseous fuel approximate 1 cu. ft. of air per 100 B.t.u.

The actual attainable flame temperature of natural gas, with cold combustion air, under practical conditions, is approximately 2800 deg. fahr., and it is obvious that in order to attain this temperature, the 10 parts of air, including the 79 per cent of inert nitrogen, will have to be raised to the flame temperature. The heat required to raise the air for the theoretical combustion of 1 cu. ft. of natural gas to 2800 deg. fahr. is 548 B.t.u., or 54.8 per cent of the calculated heating value of the fuel, assuming a gas with a heating value of 1000 B.t.u. per cu. ft. It is again obvious that if the combustion air be delivered to the burners, or grate, at some temperature exceeding atmospheric, the fuel will develop more B.t.u. per cubic foot, per pound or per gallon, as the case may be, and that proportionately less fuel will be required to develop a given quanity of heat.

To utilize a fuel to full advantage in useful work and radiation would require that the waste gases leave the furnace at atmospheric temperature. In the furnaces under consideration, any such condition is an impossibility, by reason of the temperatures required by the operation performed. It is rarely that the temperature of the gases leaving metallurgical furnaces is less than 1000 deg. fahr., and metallic recuperators are in successful operation on furnaces with waste gases at 2600 deg. fahr. The sensible heat in the waste gases at these temperatures represents a sheer loss of from 20 per cent to 56 per cent of a fuel having a definite monetary value per 1,000,000 B.t.u.

This loss can be largely salvaged only by means of a recuperator, regenerator, or in extreme cases by a waste-heat boiler. To make this point even clearer, it will be shown that by the installation of a comparatively inexpensive recuperator, the effective heating value of the fuel used may readily be increased from the equivalent of 1000 B.t.u. per cu. ft. to 1350 B.t.u. per cu. ft., without change in the cost per cubic foot.

The recuperator in its simplest, but actually impractical, form resembles a fire-tube boiler. The waste gases would pass through

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[1] Mantle Recuperator Division, Surface Combustion Co., Inc. Mr. Mantle was graduated with Bachelor of Science degree from the University of London, England. During the past ten years he has been successively Chief Engineer, A. W. Cadman Manufacturing Company; Director, Calorizing Company; President, Mantle Engineering Company, and District Manager, Surface Combustion Company, Inc., all of Pittsburgh, Pa.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

the tubes and the air would be baffled across them in the shell before being delivered to the grate or burners. In passing over the tubes the air would absorb heat from the waste gases and being delivered to the burners, or grate, would definitely reduce the quantity of fuel required to liberate a given quantity of heat.

The idea of recuperation is not new. It is difficult to trace its history. Patents on recuperators were issued in the United States more than one hundred years ago. The development of the recuperator in connection with industrial furnaces has been hindered (1) by lack of true appreciation of its value, (2) by the only recent development of a construction which could withstand the conditions encountered, (3) by the lack of a material which would withstand the temperature and air pressures required by modern combustion systems, and (4) by a lack of definite information covering the possible heat transfer from waste gases to air through an air-tight ceramic or metallic wall.

All these limitations have been overcome during the past few years and metallic recuperators are now available which may safely be guaranteed, not only as to efficiency and air tightness, but as to life under even abnormal conditions of industrial furnace operation.

General Effect of Preheated Air Upon Furnace Operation

The theoretical flame temperature of any fuel of known analysis can readily be calculated, but such calculations have, as the name suggests, a purely theoretical value, and are often misleading. The calculated flame temperature cannot be attained in practice and is far from being attained in the average industrial furnace.

The theoretical flame temperature of the rich fuels—oil, coal, natural gas, city gas, and coke-oven gas—ranges between 3600 and 3900 deg. fahr., whereas the actual flame temperature attainable in an industrial furnace with cold combustion air rarely exceeds 2600 deg. fahr. Raw producer gas at a temperature of 1200 deg. fahr., when burned with cold combustion air, has a theoretical flame temperature of approximately 3450 deg. fahr., whereas the attained temperature rarely exceeds 2500 deg. fahr.

This great difference between the theoretical and actual flame temperatures is due (1) to radiation from the flame to surrounding objects during combustion, (2) to the impracticability of burning any fuel in a metallurgical furnace with the exact theoretical quantity of air, and (3) by dissociation of the carbon dioxide and water vapor in the products of combustion. The effect of dissociation is, however, negligible in the operation of furnaces in connection with which metallic recuperators should be considered, and will not be further discussed here.

The effect of the depression of flame temperature by radiation to surrounding objects is a matter of increasing interest in furnace design. It has, however, no direct relation to recuperation, except that excessive depression can in many cases be corrected by the application of preheated air. The depression of flame temperature, due to an excess or deficiency of combustion air, has a direct bearing on the subject of recuperation.

In metallurgical furnaces it is generally desirable that the fuel be burned with a deficiency of air in order to protect the material which is being heated from free oxygen. The amount of the deficiency depends largely upon the operation being performed in the furnace. The material heated in the furnace may be such that a reducing atmosphere represented by as little as 0.1 per cent of carbon monoxide in the products of combustion will afford the necessary protection to the steel. In other cases, as in the reheating of packs or pairs for full finished sheets, it may be desired that an Orsat analysis shall show as much as 8 per cent carbon monoxide. In metallurgical furnace practice, excluding the open hearth, it is generally desirable that the amount of free oxygen in the flue gases be reduced to a minimum.

With the exception of the clean gaseous fuels, no method has been developed to proportion so accurately and mix so thoroughly the air and the fuel that the products of combustion will contain neither unburned fuel nor free oxygen. Neither, with the exception just noted, has any means been developed to maintain accurately a desired combustion condition other than theoretical under all conditions of furnace operation. It has been determined by a large number of actual tests that it is not uncommon in metallurgical furnace practice to burn the fuel with a 25 per cent excess of air and at the same time permit 12 per cent of the fuel to leave the furnace either completely unburned, in the form of smoke or soot, or partially unburned, in the form of carbon monoxide. If the furnace is fired with the richer fuels, and the operating temperature is 2200 deg. fahr., the fuel loss due to the excess air alone is approximately 10 per cent; and as 12 per cent of the fuel leaves the furnace unburned, the total loss is approximately 20 per cent. In other words, the combustion efficiency, as entirely distinct from furnace efficiency, is only 80 per cent.

This statement is typical, rather than an exaggeration, of furnace conditions which are common, but which are being remedied so far as operating conditions will permit by intelligent combustion engineering. It is doubtful, though, if for many years it will be possible to attain in industrial furnace practice a general average combustion efficiency, again as distinct from furnace efficiency, of more than 85 per cent.

This direct fuel loss is not, however, the only ill effect of incomplete combustion or of excess air. Both tend to materially depress the flame temperature. The amount of this depression can readily be calculated. In the case cited, with an excess of air of 25 per cent and 12 per cent of the carbon in the fuel leaving the furnace unburned, the calculated depression in the flame temperature of oil is 800 deg. fahr. Recent experiments prove that the actual depression is considerably greater than the calculated, and in the condition named, the true drop in flame temperature would probably be 1300 deg. fahr.

The rate of heat transfer between a heating medium and a body to be heated is proportional to the difference in the fourth power of the respective temperatures by radiation, and to the direct difference in the respective temperatures by convection. It will be obvious, therefore, that any depression of flame temperature must affect the potential production of the furnace. This is particularly true in the case of the lean fuels, to which there is a marked trend in the steel industry today.

Recuperation not only effects a definite fuel saving, but it permits combustion with the minimum excess or deficiency of air which practical conditions demand. It also assures greatly increased furnace production when the richer fuels are necessarily burned with a heavy deficiency of air or when very lean fuels are employed.

Heat Loss From Industrial Furnaces in the Flue Gases

If all the heat in the fuel were to be utilized in useful work and radiation in a furnace, the gases would leave at atmospheric temperature. This is obviously impossible in industrial furnace practice, and every degree of temperature above atmospheric represents a readily calculated fuel waste. The magnitude of this waste may be a little surprising to those who look askance at an over-all boiler efficiency of less than 85 per cent.

If we assume that it is possible to attain the theoretical flame temperature of 3900 deg. fahr., in an industrial furnace, and if we ignore the variation in the specific heat of the flue gases with temperature, it will be evident that the heat lost in the flue gases alone would be, approximately, the ratio of the temperature of the flue gases to the flame temperature. For instance, if the gases leave the furnace at 2400 deg. fahr., the waste is 61.5 per cent; if at 1500 deg. fahr., 38.5 per cent; and if at 1000 deg. fahr., 25.6 per cent.

TABLE 1

Type of furnace	Fuel loss in waste gases, per cent
Copper refining	55
Brass and copper melting	52
Forging furnace, batch type	52
Billet furnace, batch type	49
Hardening furnace	47
Blue annealing sheets	42
Carburizing	36
Normalizing sheets	36
Pair heating	35
Annealing castings	33
Continuous billet furnace	30

if the waste gases can be utilized to preheat the combustion air to any degree, less of the heat in fuel will be required to raise the air to flame temperature and a definite saving must result. This is the function of a recuperator.

Curve 2, Fig. 1, shows the heat in the air required for the theoretical combustion of the richer fuels as a percentage of the total heat in the fuel. For example, the heat in the air required for the theoretical combustion of coke-oven gas is at 1000 deg. fahr., 17.5 per cent of the total heat in the fuel.

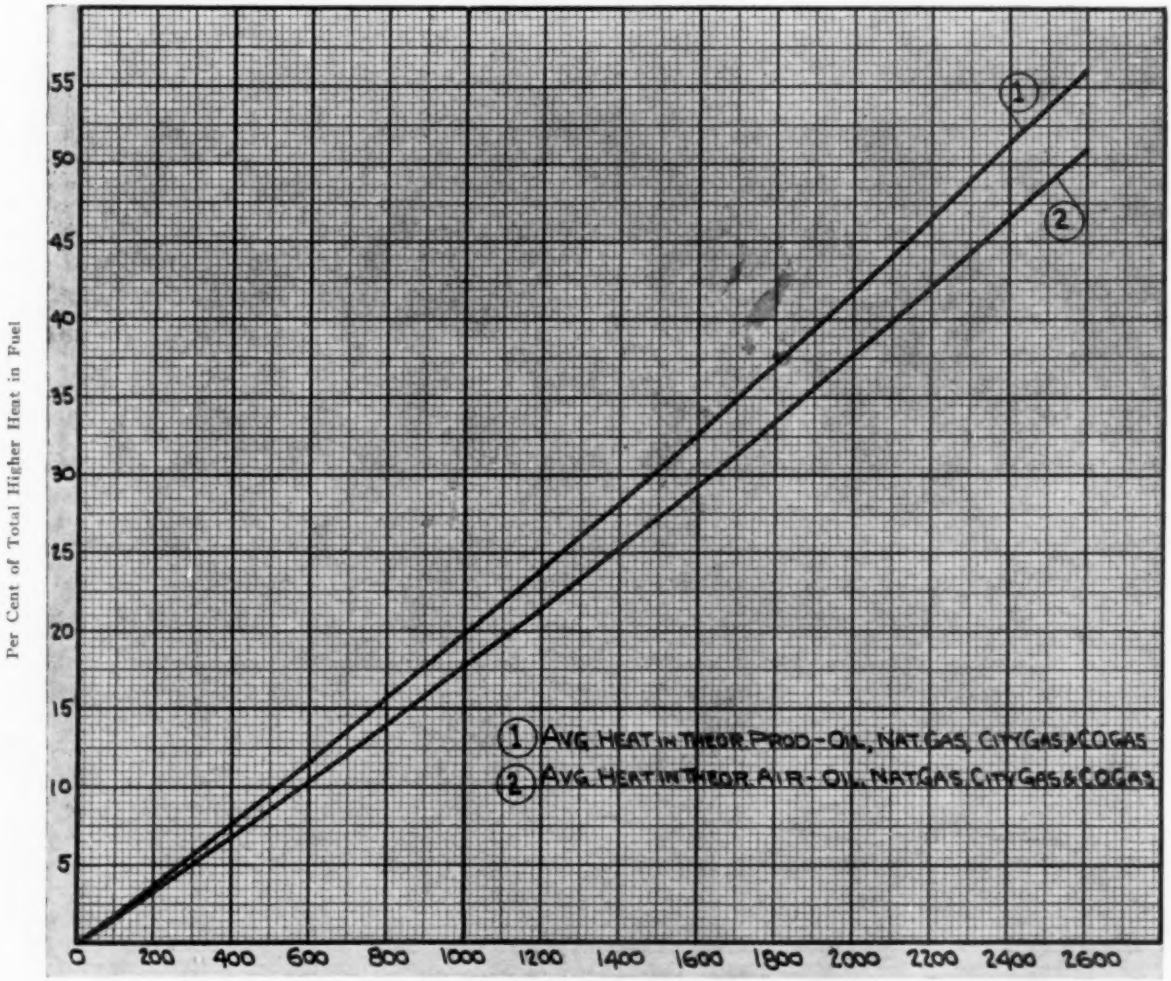


FIG. 1 HEAT CONTENT AS PER CENT HIGHER HEATING VALUE OF FUEL

The actual heat loss in the flue gases, assuming theoretical combustion, is shown as a percentage of the total heat in the fuel for the richer fuels (oil, coal, natural gas, city gas, and coke oven gas) in curve 1, Fig. 1.

The list of typical furnaces, Table 1, shows the extent of this waste.

It should again be emphasized that the losses shown are solely those due to the sensible heat in the flue gases, assuming theoretical combustion.

How Much Will a Recuperator Save?

It has been pointed out that in burning any fuel it is necessary to raise the combustion air to the flame temperature and that much of the calculated heat in the fuel is used in raising the combustion air to the flame temperature. It is equally obvious that

It would seem, therefore, that if the waste gases are utilized to preheat the air to 1000 deg. fahr. for a coke-oven gas-fired furnace, 17.5 per cent of the heat in the fuel would be returned to the furnace and the gas consumption would be reduced by that amount. Actually, however, the saving effected is considerably greater than this and increases both with the temperature of the waste gases and the inefficiency of combustion.

As an example, let us apply a recuperator to a continuous reheating furnace fired with any of the rich fuels. The exit-gas temperature is approximately 1800 deg. fahr., and the air is preheated to 900 deg. fahr. Again assume that it is possible to effect combustion with theoretical air, and ignore the loss due to the latent heat of water vapor, caused by the combustion of the hydrogen in the fuel. It is obvious that when the gases have left the furnace they are no longer available for work of any kind in

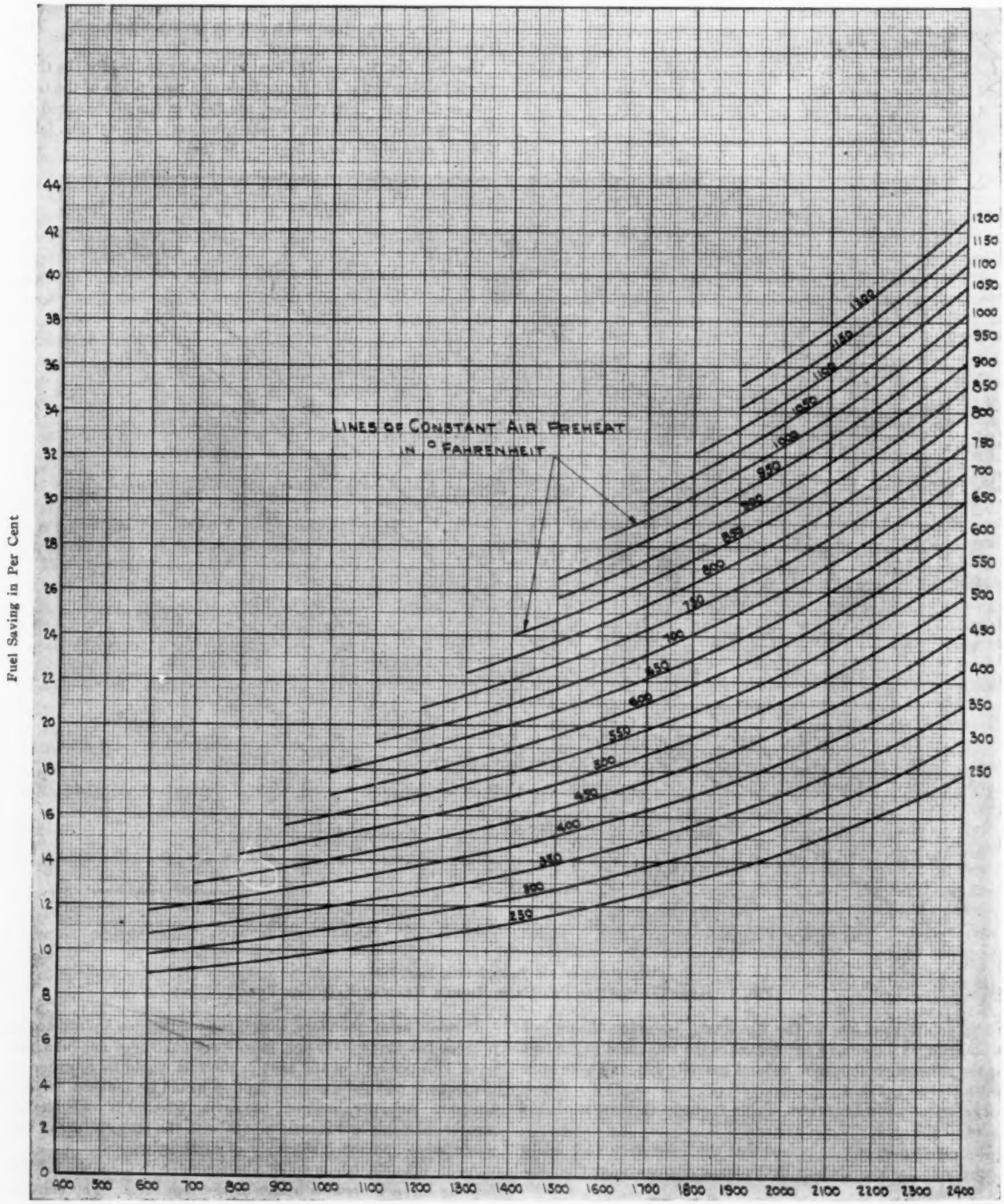


FIG. 2 PER CENT FUEL SAVING WITH VARIOUS WASTE-GAS TEMPERATURES AND AIR PREHEATS ABOVE 60 DEG. FAHR. FOR NATURAL GAS, OIL, COKE-OVEN GAS, AND CITY GAS

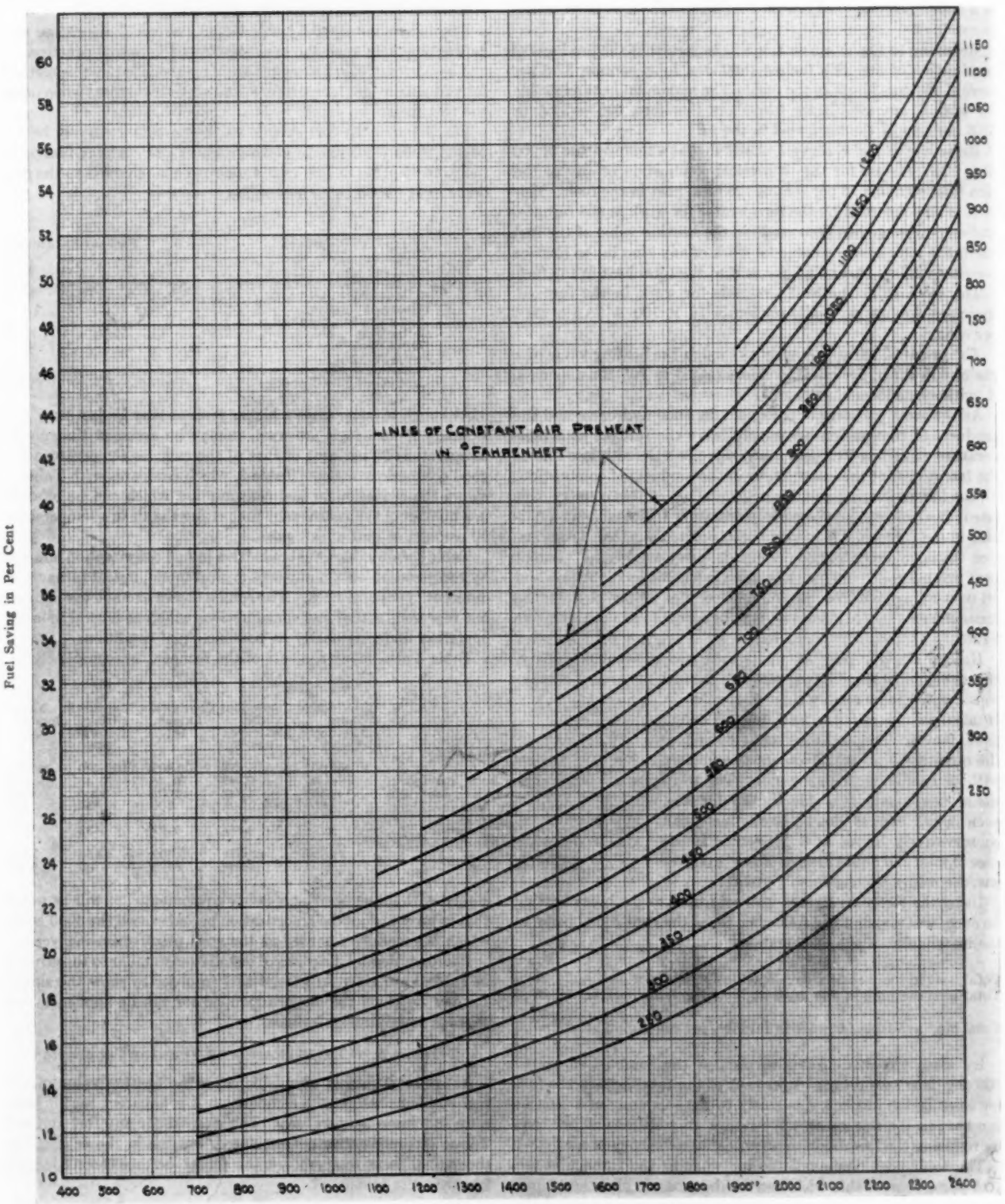


FIG. 3 PER CENT FUEL SAVING WITH VARIOUS WASTE-GAS TEMPERATURES AND AIR PREHEATS ABOVE 60 DEG. FAHR. FOR PRODUCER GAS

the furnace and that the heat available for work in the furnace is, as a maximum, the total heat in the fuel less the heat in the waste gases.

Referring to Fig. 1, curve 1, it will be seen that the products of combustion of any rich fuel at 1800 deg. fahr. contain 37.5 per cent of the total heat in the fuel. The maximum heat available for work of any kind in the furnace is, therefore, 100 — 37.5, or 62.5 per cent of the total heat in the fuel.

Reference to Fig. 1, curve 2, shows that at a temperature of 900 deg. fahr., the air for the theoretical combustion of any of the rich fuels contains 16 per cent of the total heat in the fuel. This heat is now returned by the recuperator for work in the furnace, and as no change has been made in the heating value of the fuel itself, the total heat available for work of any kind in the furnace becomes 62.5 + 16, or 78.5 per cent of the total heat in the fuel.

The increase in the heat available for work in the furnace, therefore the fuel saving, is obviously 16 × 100 ÷ 78.5 or 20.4 per cent.

The practical effect is exactly as though the heating value of a gas of 635 B.t.u. per cu. ft. were increased to 795 B.t.u. per cu. ft. without additional cost.

As the great majority of fuels contain some hydrogen, and the products of combustion necessarily leave the furnace at a temperature exceeding 212 deg. fahr., there is a definite loss due to the latent heat of the water vapor in the products of combustion. As the higher heating value is in general use in this country, this latent heat must be deduced in calculating the total heat available in the furnace. It has also been pointed out that a combustion efficiency exceeding 85 per cent is rarely attainable. Latent heat was not taken into consideration in the earlier discussion and an over-all average combustion efficiency of 85 per cent, still as distinct from furnace efficiency, may be accepted as a conservative figure.

Heated air, due possibly to its increased molecular activity, effects combination of its oxygen with the fuel considerably more readily than does cold air. It also plainly requires less time to attain flame temperature, and thus tends to reduce the quantity of air which enters the furnace without combining with the fuel. The additional heat made available in the furnace, even in the case of clean gaseous fuels, can conservatively be set at 3 per cent. In the case of solid and liquid fuels or of raw producer gas, the return is considerably greater.

Using the foregoing figures, and returning to the billet heating furnace, we now find the following to be the conditions prior to the installation of the recuperator:

Total heat in fuel actually developed in furnace, per cent	85.0
Total heat escaping in flue gases at 1800 deg. fahr., per cent.	37.5
Total heat available for work in furnace, per cent	47.5

By using the waste gases to preheat the combustion air to 900 deg. fahr., we return 16 per cent of the total heat in the fuel for work in the furnace, due solely to the salvaging of a part of the heat in the flue gases. We return an additional 3 per cent by reducing the losses due to an excess or deficiency of air.

The total heat now available for work in the furnace is the 47.5 per cent available with cold combustion air plus the 19 per cent due to the use of preheated air, and the total heat available with the recuperator becomes 66.5 per cent of the total heat in the fuel.

The increase in the available heat, therefore the fuel saving, is 19 × 100 ÷ 66.5, or 28.6 per cent, and the practical result is again as though a gas with a heating value of 475 B.t.u. per cu. ft. were increased to a heating value of 665 B.t.u. without additional cost.

The figures obtained by this method of calculation agree surprisingly closely with the savings effected in actual practice.

The curves in Fig. 2 show the savings calculated by this method for various exit-gas temperatures and air temperatures when using the rich fuels.

Fig. 3 shows the fuel saving which can conservatively be expected when using raw or clean producer gas. The reduction in fuel consumption in this case applies to coal charged to the producer, and not to cubic feet of gas.

TABLE 2

Type of furnace	Fuel saving, per cent
Copper refining	45
Brass and copper melting	42
Forging furnace, batch type	42
Billet furnace, batch type	40
Hardening furnace	37
Blue annealing sheets	32
Carburizing	26
Normalizing sheets	26
Pair heating	26
Annealing castings	25
Continuous billet furnace	20

Although in some cases a higher preheat may be justified, particularly if lean fuels are used, it may be assumed that an air temperature 50 per cent of that of the exit gas temperature is a good economic average. Table 2, which compares with Table 1, shows the reduction in fuel consumption which can be effected in a number of typical cases by preheating the air to a temperature 50 per cent of that of exit gases.

Table 3 shows the actual reduction in fuel consumption effected by recuperation on a number of typical metallurgical furnaces of different types and sizes. The results shown are not test figures, but represent actual fuel consumptions covering several months' operation under as nearly as possible equal conditions.

Table 3 also shows the cost of the recuperator, completely installed with all necessary accessory equipment. It will be noted that in no case does the amortization period exceed the 6000 hours which is generally regarded as the steel plant operating year.

TABLE 3

Type of furnace	Continuous billet	Continuous billet	Continuous forging	Galvanizing	Batch-type forging
Fuel	Producer gas	Coke-oven gas	Oil	Natural gas	Oil
Fuel consumption, B.t.u. per hour without recuperator	40,000,000	50,000,000	7,200,000	3,000,000	1,750,000
Fuel cost per million B.t.u., dollars	0.20	0.30	0.45	0.45	0.38
Temperature exit gases, deg. fahr	1600	1900	1600	1200	2400
Temperature air, deg. fahr	850	950	700	650	800
Saving, per cent	32	26	30	24	37
Value saving per hour, dollars	2.56	3.90	0.97	0.32	0.25
Cost of recuperator, installed complete with accessories, dollars	11,800	12,400	3,300	1,650	1,150
Amortization period, hours	4600	3180	3400	5150	4600

Photographs and diagrams of installations similar to those listed in Table 3 are given in Figs. 4 to 7 in order to show the simple manner in which the modern recuperator can be applied to the average metallurgical furnace.

Influence of Recuperation on Furnace Production

As recuperation, in effect, increases the heating value of the fuel without changing its composition it is apparent that it must raise the flame temperature. The increase for any fuel and any degree of preheat can be calculated, but, as in the case of the theoretical flame temperature, the actual increase is considerably less than the calculated.

The rise varies somewhat with the fuel and with furnace conditions, but tests under operating conditions indicate an increase in flame temperature of approximately 30 deg. fahr. for each 100 deg. of preheat.

The rate of heating in a furnace is of course a function of the difference in temperature between the heating gases and the

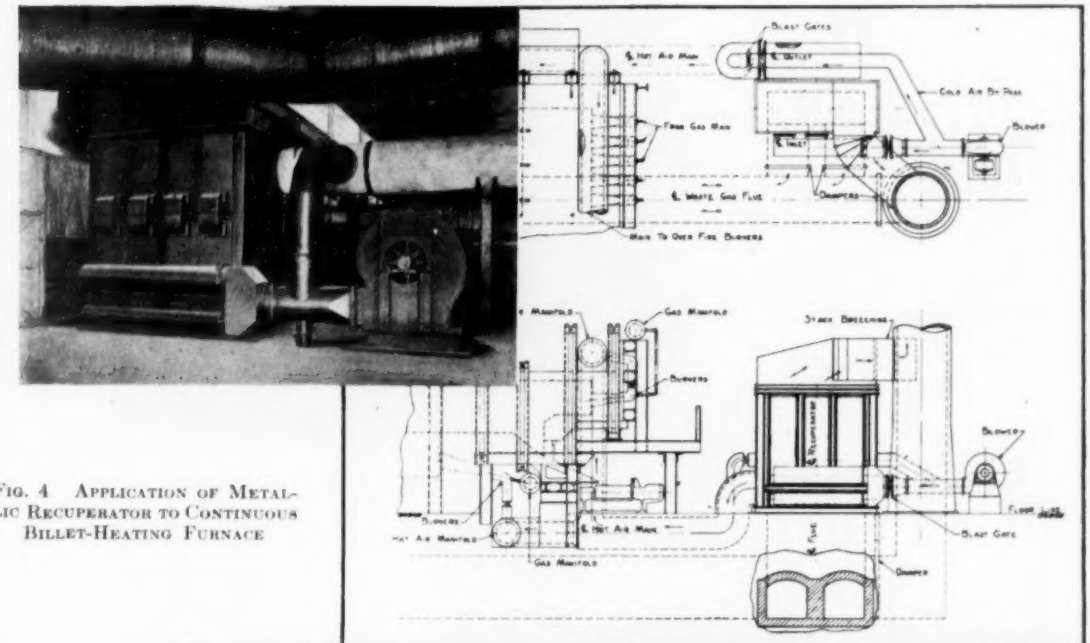


FIG. 4 APPLICATION OF METALLIC RECUPERATOR TO CONTINUOUS BILLET-HEATING FURNACE

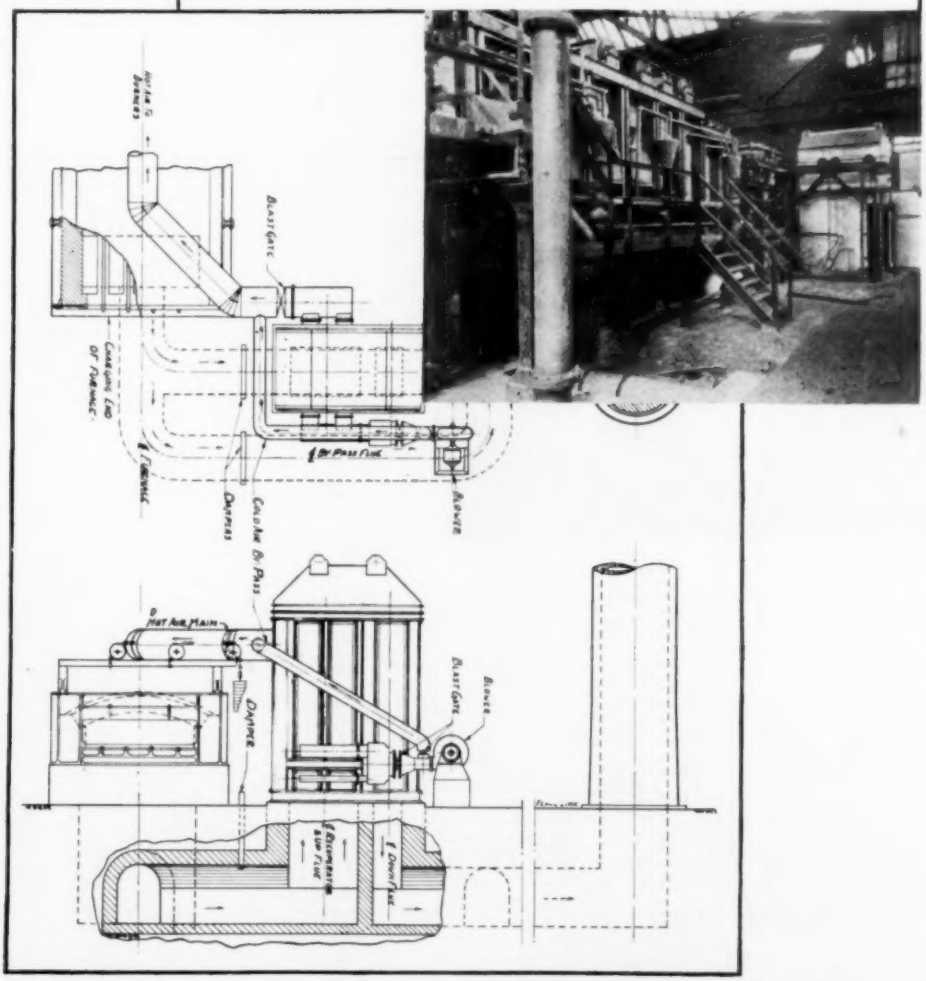


FIG. 5 ANOTHER METHOD OF APPLICATION IN BILLET-HEATING FURNACE

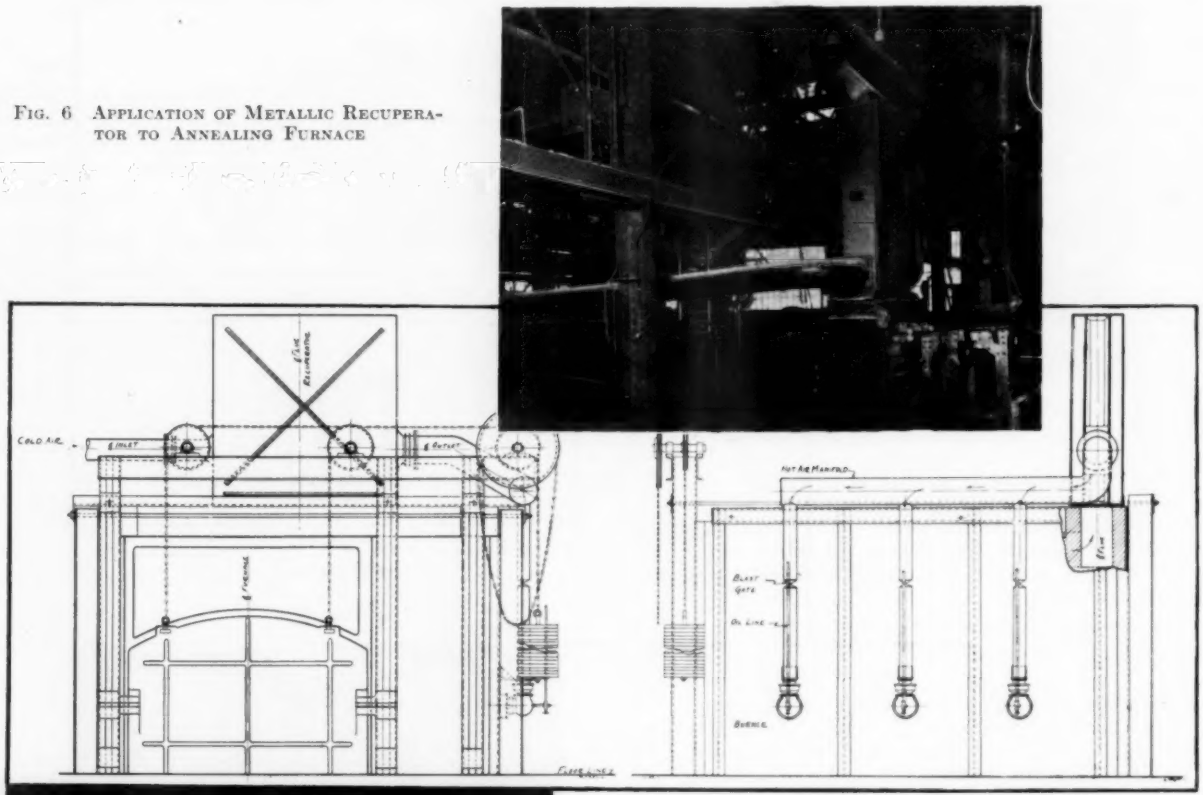


FIG. 6 APPLICATION OF METALLIC RECUPERATOR TO ANNEALING FURNACE

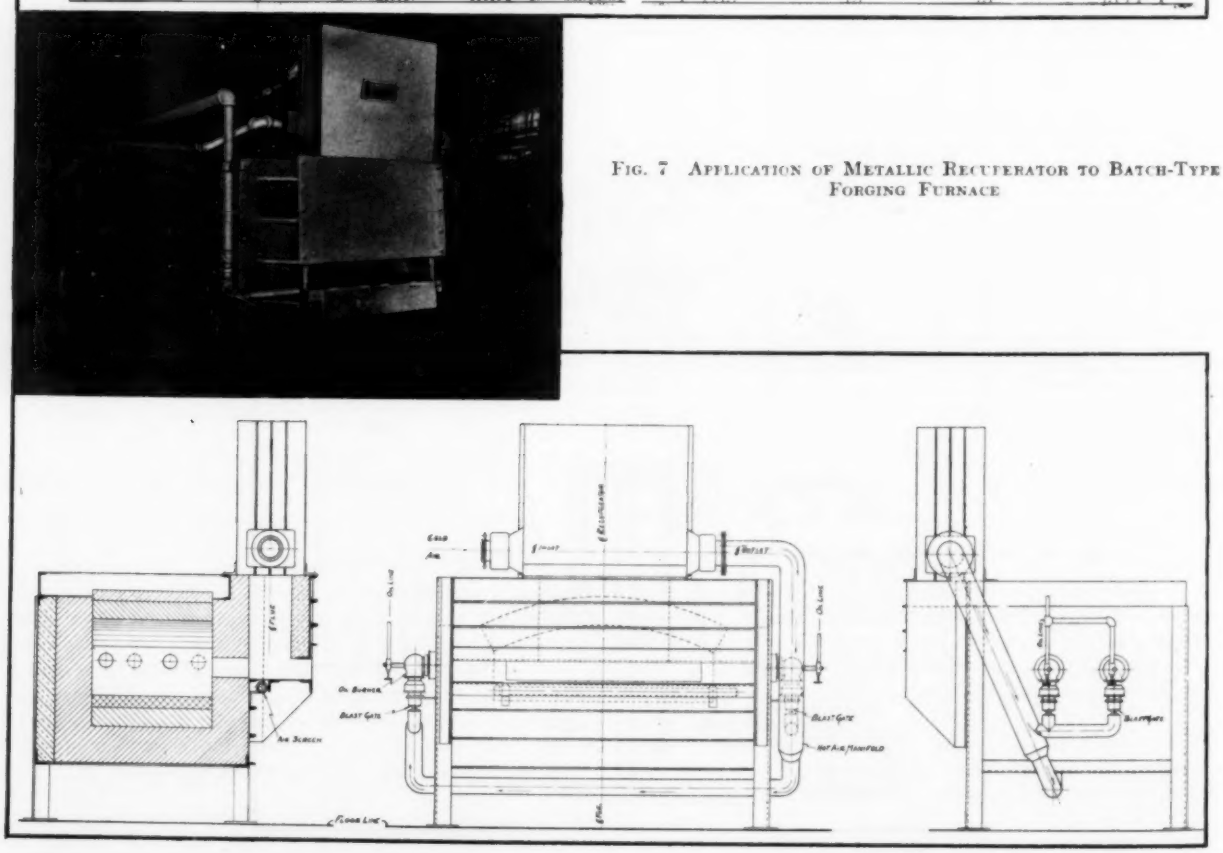


FIG. 7 APPLICATION OF METALLIC RECUPERATOR TO BATCH-TYPE FORGING FURNACE

material to be heated. When, therefore, it is possible to utilize a furnace temperature higher than that attainable with cold combustion air, a recuperator will effect a marked increase in the possible production.

With the rich fuels the increase in production can only be utilized when the material heated in the furnace is to be raised to a final temperature exceeding 2000 deg. fahr. With lean fuels, such as clean cold producer gas, low heating value mixed gases or straight blast-furnace gas, it is often difficult to obtain the flame temperature required for the higher temperature operations, and recuperation or regeneration becomes a necessity. A marked increase in production is often to be observed when these fuels are used in connection with the lower temperature operations.

If it is necessary to carry a furnace temperature exceeding 2400 deg. fahr., using either rich or lean fuels, recuperation will effect an increase in production of as much as 40 per cent.

The reduction in fuel consumption due to recuperation is also a factor in increasing the potential production of a furnace. The furnace volume in many cases limits the quantity of fuel which can effectively be burned in a furnace and, therefore, also limits the production. As recuperation increases the effective fuel capacity of a furnace it also increases the potential production.

There is a more or less general impression that the increase in flame temperature inherent to recuperation is disadvantageous in furnaces operating at temperatures below 2000 deg. fahr. There is not necessarily, however, any direct relation between flame temperature and furnace temperature. A high-temperature flame can be diffused as a comparatively low temperature atmosphere in the furnace by intelligent use of the radiated heat from walls, arch and hearth, or by recirculating the gases through the flame. The addition of a recuperator simply means that less fuel is required to maintain a given furnace temperature for a given production.

Some outstanding examples of the successful application of recuperation are in connection with such low-temperature operations as copper-strip annealing and galvanizing pots.

Use of Preheated Air in Connection With Lean Fuels

The increased production of blast-furnace gas and its more efficient application have provided many steel plants with a surplus of this fuel, and there is a rapidly increasing tendency to use this gas in connection with the higher temperature operations, such as heating furnaces and soaking pits.

The theoretical flame temperature of the average blast-furnace gas is only 2500 deg. fahr., and as it is not possible to develop more than 80 per cent of this, the maximum flame temperature attainable with cold combustion air will scarcely exceed 2000 deg. fahr. The actual flame temperature required in heating furnaces and soaking pits may be as high as 2700 deg. fahr., and it is evident that the effective heating value of the blast-furnace gas must be materially increased before it can be utilized. This is done by enriching the gas with another of higher heating value, by preheating the air, by preheating the gas, or by preheating both air and gas.

To attain a flame temperature of 2700 deg. fahr., it is desirable that the theoretical air-gas mixture have an effective heating value of 75 B.t.u. per cu. ft., as compared with 57 B.t.u. per cu. ft. of mixture when using cold air and blast-furnace gas.

If coke-oven gas is the enriching medium and the mixed fuel is to be burned with cold air, it is necessary to add 40 parts by volume of coke-oven gas, with a net heating value of 440 B.t.u. per cu. ft., to 60 parts of a blast-furnace gas, with a net heating value of 97.5 B.t.u. per cu. ft. The heating value of the mixed gas is 234.5 B.t.u. per cu. ft. and the heating value of the theoretical cold air-gas mixture is 75 B.t.u. per cu. ft.

By using a recuperator to preheat the combustion air to 1000 deg. fahr., the required 75 B.t.u. per cu. ft. of air-gas mixture can be obtained by using only 12.5 parts of coke-oven gas to 87.5 parts of blast-furnace gas—a reduction of nearly 70 per cent in the quantity of coke-oven gas and an attendant improvement in combustion and furnace atmosphere. This mixed gas has a net heating value of 140 B.t.u. per cu. ft. and requires 1.16 cu. ft. of air for the theoretical combustion of 1 cu. ft. This air when raised to 1000 deg. fahr. contains approximately 22 B.t.u., raising the effective net heating value of the mixed gas to 162 B.t.u. per cu. ft. and giving a heating value for the air-gas mixture of 75 B.t.u. per cu. ft.

An interesting installation of recuperators is planned in a large steel plant in the Chicago district. In this case it is proposed to use straight blast-furnace gas to raise steel to 2400 deg. fahr. in large continuous billet-heating furnaces. Both air and gas are to be heated to 1000 deg. fahr. in special metallic recuperators.

Assuming a gas with a net heating value of 97.5 B.t.u., weighing 0.08204 lb. per cu. ft. wet and requiring 0.05566 lb. of air for the theoretical combustion of 1 cu. ft. of gas, the heat in the air at 1000 deg. fahr. is approximately 13 B.t.u., and the heat in the gas approximately 20 B.t.u., making the total effective heating value 130 B.t.u. per cu. ft. The heating value of the air-gas mixture is here again 75 B.t.u. per cu. ft.

Conclusion

It has been the prime object of this paper to effect a realization of the value to industry of recuperation. For this reason, any description of the highly developed types of commercial equipment now available has been omitted. The huge expenditures on fuel and its increasing value are rapidly making recuperation essential to the metallurgical industries. Even more essential, therefore, is a wider appreciation of its advantages.

Discussion

Max Sklovsky.[2] The author has made a substantial contribution on the subject of economies of fuel in industrial furnaces through the process of recuperation. The data are conservative, and his claims are frank and modest.

Comments made on this paper show that there has been considerable disappointment in the results obtained from recuperators. The writer does not believe that the disappointments in a number of cases offset the actual merit obtained from recuperation; therefore, as a user of recuperators, he desires to indicate the following:

1 The writer has used recuperators regularly for 11 years on a large number and a large variety of furnaces, and nearly all of these are still in use, and in most cases the original recuperators are still serving, although the earlier recuperators were not a type to insure the lasting qualities with which some of the present types are made.

2 The failure of recuperation is frequently attributed to lack of cooperation on the part of users. The abuse to which furnace equipment is put is inexcusable.

3 The purchaser of the recuperative equipment expects too much of such equipment. In many instances the furnace construction does not permit proper application of recuperators, and frequently the user is unwilling to modify the furnace construction for the purpose, thus forcing the recuperator agency to apply recuperation under conditions that are not at all favorable. The recuperator manufacturer under the present conditions is obliged to take on the burden which does not belong to him of a redesign of the furnace, and the responsibility is too readily accepted by the recuperator salesmen.

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[2] Chief Engineer, Deere & Co., Moline, Ill. Mem. A.S.M.E.

4 The application of a recuperator properly carried out requires close observation of the particular furnace, an engineering analysis of same, and frequently supplying a partial redesign to permit the recuperator to be installed. It also requires follow-up observations, adjustments and engineering attention, all of which result in very high cost of application of recuperators; and with the recuperator manufacturer absorbing all of the contingent expenses and responsibility for the results, the installation price is frequently excessive. A little attention and cooperation on the part of the purchaser will materially reduce the cost and improve the results of recuperation. Without such cooperation, fewer successes will be attainable.

5 The results of recuperation are numerous, among which can be listed reduced fuel consumption, higher temperatures, more rapid heating, simpler control, and greater output.

E. W. Wagenseil.[3] The author deserves thanks for the clear and logical manner in which he has covered the subject. The paper points the way to definite results in the way of lowered heating costs, increased tonnage, and higher quality heating, and it should create a wide interest among steel-plant engineers and mill-operating officials.

There are one or two things which might perhaps be somewhat misleading, and which should have some further discussion. The writer would like to call attention to the fact that the temperature to which air can be preheated is by no means limited to 50 per cent of the waste-gas temperature, which the author mentions as being "a good economic average." The curves which he shows in Figs. 2 and 3 show air temperatures up to about 65 per cent of the waste-gas temperature. We have several installations of Blaw-Knox preheaters in connection with continuous furnaces where the air is heated to as high as 75 or 80 per cent of the waste-gas temperature. This high preheat is particularly desirable in furnaces where cold billets are charged and the waste-gas temperature leaving the furnace ranges from 1400 deg. fahr. to as low as, perhaps, 1000 deg. fahr. In such cases it is desirable to heat the air to as high a percentage of the waste-gas temperature as possible.

It should be pointed out, however, that in order to utilize high air temperature successfully it is necessary that the burners and the furnace be properly designed for the use of high air preheat; otherwise burned steel and serious cinder conditions may be the result.

It has occasionally happened in the past that cases of this kind have been offered as proof that high air preheat cannot be used advantageously on steel-heating furnaces. Actually such cases merely demonstrated that the burners and furnace were not properly designed for the amount of preheat being used. Mr. W. P. Chandler, Jr., in his article on "Economical Combustion," in the June issue of *Iron and Steel Engineer*, points out some of the factors which have a vital bearing on the subject of proper burner design for high air temperatures.

The author mentions that metallic recuperators are in successful operation on furnaces with waste-gas temperatures as high as 2600 deg. fahr. The writer is sure that this statement was not intentionally meant to be misleading and that the author does not claim that there are metallic recuperators in service where the temperature of the waste gases actually entering the recuperators is anything like 2600 deg. fahr. In cases of this kind it is absolutely essential that the waste-gas temperature should be brought down to something like 1800 deg. fahr. in order not to subject the preheater at times to temperatures which might be destructive to the materials of which it is made.

It is unsafe to try to accomplish this by admitting cold air to the waste-gas flue as the waste gas frequently contains fairly high percentages of CO, and in such gases the admission of cold air will at once start secondary combustion in the preheater, with disastrous results. The ideal way to reduce high waste-gas temperature is by dilution with some of the same gases which have been through the preheater and are at a comparatively low temperature. This can only be done, however, if a waste-gas exhaust fan is used in connection with the preheater.

There are a considerable number of Blaw-Knox regenerative air preheaters in service on one-way fired soaking pits where the waste-gas temperature leaving the pits when the ingots are up to rolling temperature sometimes reaches 2400 deg. fahr. or higher. We reduce this temperature to a maximum of 1800 deg. fahr. entering the preheater by means of a thermostatic control (the arrangement of which is located in the hot waste-gas intake to the preheater), which automatically takes part of the cooled waste gas which has been through the preheater and exhaust fan and mixes this cooled gas with the hot waste gas from the pits, bringing the temperature down to 1800 deg. entering the preheater. This method makes it possible to operate the preheater with safety in connection with all types of heating furnaces no matter how high the temperature of the waste gas leaving the furnace.

Harold F. Wood.[4] American industry has reached a point in its development where costs of every detail operation must be watched with the utmost care and the maximum of efficiency be obtained if profitable operation is to be secured.

In the metal-working industry the question of furnaces is a very considerable item of plant equipment, and the question of furnace efficiency is a very important consideration in product cost. It is no longer sufficient merely to install a piece of heating equipment that will heat metal for a certain operation uniformly and of a certain volume, but it is necessary to give detail study to the question of the most economical fuel to use and the burning of that fuel in the most economical way.

Recuperation is a perfectly logical way of reducing fuel cost. The author has pointed out clearly how this can be done. The tables presented are correct and represent in each case exactly what the possibilities for heat salvage are. From a practical standpoint it is not, however, always possible to realize the savings shown in the tables. This is due to two reasons. In the first place, in continuous furnace operations, where the flow of material through the furnace is on a definite time cycle, equilibrium conditions of the gases are upset considerably by the frequent opening and closing of the doors, making it impossible to get all the waste gases through the recuperator. This is very noticeable in low-temperature furnaces used at temperatures around 1000 to 1200 deg. fahr. In the second place it is necessary to use cold air for atomization, and the volume of cold air runs, in the majority of cases, with present available burner equipment, approximately 25 per cent of total air. This of course reduces possible economies. It is hoped that serious thought will be put on the subject of burners for use with preheated air with the hope of developing one that requires little or no cold air for its operation. It is also hoped that the furnace engineers will be able to design their furnaces so as to assure maximum amount of waste gases passing through recuperator.

The metallic recuperator as a unit is very good. The writer cannot agree entirely with the thought of using metallic recuperators on waste gases entering the recuperator at temperatures of over 2000 deg. fahr. He has been very much interested in recuperators since the first ones were built and will say that he

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[4] Chief Metallurgist, Ingalls-Shepard Division, Wyman-Gordon Co., Harvey, Ill.

has had no thought of installing a furnace in the past eight years without recuperation. We have twelve continuous billet furnaces operating at forging temperatures and seven large continuous heat-treating furnaces all equipped with metallic recuperators. Although we have had some recuperator trouble, this trouble has been the least per dollar returned on investment of any equipment in the plant, and any furnace installation made by our company in the future will include recuperation.

IS-52-10

Design and Construction of Hot Saws for Cutting Heavy Sections

By A. B. PEARSON,[1] MUNHALL, PA.

Sawing is an ancient art and it may be said without fear of contradiction that its origin dates from primeval times. A stone or a shell with a ragged edge may have served as the implement, and the pressure or feed, as well as the speed, were then, as now, very important factors in the undertaking. Other methods of cutting have been found, but the abrasion process is still the one that is most generally employed for cutting heavy material to length. In the use of this process, as in sawing, the cutting tool is generally harder than the material being sawed. The author describes saws for cutting hot and cold metals and gives results of tests on a variety of saws.



IN THE sawing of metals such as steel there are two distinct fields, that of sawing steel cold and that of sawing steel hot. When sawing steel cold, the heat developed is not an aid to the cutting and must be kept within limits, whereas when sawing steel hot, the heat must be created at a rapid rate to insure successful cutting. In the method of sawing steel hot it is necessary to create this difference in temperature and consequent hardness between the cutting tool and the material being cut, and it is clear that without this difference in hardness no cutting could take place. Therefore in hot-sawing the rotating steel disk has two functions to perform (1) to heat the steel by friction, and (2) to remove at least a part of the steel so heated.

Some cooling agent has to be used to cool the saw blade for continued cutting. It is evident that the friction required is in inverse proportion to the temperature of the steel sawed. This method of cutting steel is used extensively in steel manufacturing plants and warehouses throughout the country.

In steel-mill practice the name "hot saw" is only applied to a sawing machine used for cutting red-hot steel coming directly from the mill. The name "cold saw" is applied to a similar machine used for cutting cold steel coming from the cooling bed or the stock pile. In both of these machines the cutting is done with a rotating saw blade or disk having a peripheral speed of several thousand feet per minute. The name "cold saw" as applied to this type of machine is a misnomer, for by cold saw ordinarily is meant a saw with a rotating disk having hardened teeth and which, with the aid of a cutting lubricant, rotates at a speed of approximately 100 ft. per min. Some writers to avoid this inconsistency, have given the cold saw, known as such in the steel mills, the name cutting-off machine, friction-sawing machine, fusion-disk machine, or simply "cold saw." Acknowledging the error the author will also use the name "cold saw."

In mill practice, a delay between the finishing mill and the hot saw may cause the steel to cool, making it necessary to use a saw blade for cutting cold steel on a saw that is normally used for hot sawing. This illustrates the relationship between the two saws, the principal difference being the cutting edge and the speed of the saw blade.

The introduction of the rolled shaped steel sections brought about the necessity of cutting without distortion and at a higher speed than had been done up to that time. Therefore the high-speed hot saw as well as the cold saw came into being. The Carnegie Phipps Company, Ltd., which later became the Carnegie Steel Company, has a record of installing a cold saw in the year of 1887. Two years later in 1889 the same company installed a hot saw for sawing hot steel.

The early high-speed hot saws used for cutting steel were of the stationary-frame type, having a manual-feed arrangement. For light work these saws have been and still are efficient, but as the material to be cut has increased in size, the manual feed had to give place to some sort of power feed. With the introduction of the power feed, a greater ease in the manipulation of the feed was accomplished. However, the power feed brought a less sensitive feed movement.

The stationary-frame type of saw makes it necessary to feed the material to the saw while cutting. This is satisfactory for short lengths, but for long lengths the material has to be clamped in position for each cut. The clamping caused delays in production and therefore a change became imperative. However, the disadvantage could not be easily overcome without a radical departure in the design of the saw.

Thus came into use the drop saw and the pendulum saw which allows the material to be stationary while the saw frame is moved relative to the material being cut. The drop saw, also called the balance saw, is fed from the top down; and the pendulum saw is fed from the side in a swinging fashion. The drop saw is now used for cutting rails and material of a similar character.

With the drive and the saw blade mounted on opposite ends of a walking beam, the drop saw is fed by tilting the walking beam so that the saw blade comes in contact with the material to be cut. For steel with a varied rectangular cross-section this saw is not suitable except for narrow widths.

The pendulum saw, although having the advantage of cutting from the side, has other features not conducive to good design. The two most outstanding are the high suspension and the vertical belt drive, the latter feature being perhaps the more objectionable. It should be said, however, where floor space is limited the pendulum type of saw may be justified. It was but natural that the saw development should take this course because of simplicity of design and the consequent low cost of production. As the structural material increased in size it became necessary to design a saw free from the limitations previously encountered.

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Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

After trying and finding use for the stationary saw, the tilting, and the swinging saw one comes to the sliding saw. The stationary and the sliding-frame saw does the sawing from the side in the same manner except that the material being cut is moved in a stationary saw and remains stationary in a sliding saw.

The sliding-frame saw has met with success particularly when used for the cutting of heavy structural-steel sections. There may be a number of details that can be improved or altered to suit different conditions, but the design of the sliding frame moving horizontally on a stationary base is a basic principle in design that will remain.

The Sliding Frame Saw

The so-called sliding frame is really a rolling frame for the larger saws because the frame rolls on a stationary base plate.

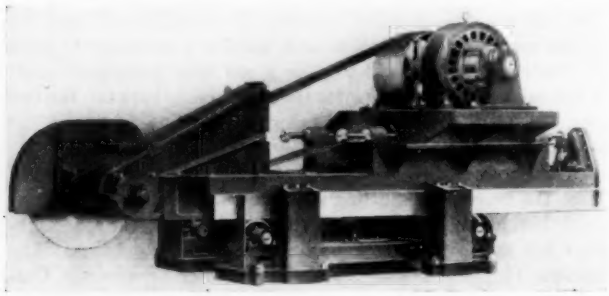


Fig. 1 Belt-Driven Sliding-Frame Saw

The belt-driven saw (Fig. 1) of this type has a motor mounted on the frame with a belt from the drive pulley to the saw arbor. The saw blade is mounted on one end of the arbor shaft which permits a rapid change of saw blades; however, a heavy construction is required as the load is overhung and one sided. The horizontal travel of the saw frame is ordinarily about 6 ft. for saws now used for cutting the heavier structural beams. With this travel it is evident that the saw frame must be long and also heavy, not only for strength, but also to balance the loads when the saw frame is in one of the two extreme positions. The saw frame must be heavy to absorb the minute rapid vibrations set up by the rotation of the saw blade.

Saws of this type have advantages, the most important being that the saw frame is outside and adjacent to the table on which the material is moved to sawing position. This permits a ready means for the disposal of crop ends and scrap into a pit located immediately under the table. In mill practice this is a very important factor where several thousand tons have to be sawed in a day's time. The sliding frame lends itself equally well for sawing either hot or cold steel.

Another sliding frame saw (Fig. 2) has met with considerable success. The frame of this type of saw is built in a goose-necked fashion, the lower part extending under the table which supports the material as it is being cut. The saw-drive motor is mounted directly on the saw arbor, thus eliminating the belt. This saw has so far been used only for sawing cold material, but with means for cooling the motor, there seems to be no reason why the direct drive cannot be applied to a saw for sawing hot steel. It can readily be seen that the direct drive has potentialities which if realized would in time eliminate the belt-driven type. However, this has not been done, and it seems that the difficulty is in the main an electric problem.

The Saw Feed

In general the larger hot saws are equipped with hand-controlled power feed having a hydraulic or hydro-pneumatic, and in later years, an entire electric arrangement. The feed is arranged so that the movement is variable in speed when the saw blade is making the cutting stroke and either variable or constant in speed on the return stroke.

In the hydraulic-feed arrangement the liquid is forced by a pump to the feed cylinder. A spring-actuated relief valve and choke are placed in the pipe line between the operating valve and the cylinder to limit the pressure and the flow respectively. A fast return stroke is secured with bypassing the liquid through a one-way check valve. An air cushion is sometimes used as a shock absorber when the feed cylinder is connected to a hydraulic accumulator system. The rate of feed is dependent on the opening of a valve and in practice is not as flexible as a geared motor-driven feed system. There may be some objection to possible leakage at cylinder packing and by freezing, but in the main this feed is simple and perhaps less costly than any other power feed.

The electrical power feeds are principally of two kinds, namely, what may be called the direct and the indirect types. The direct electrical feed has two motors running in opposite directions driving through an enclosed differential set of gearing. When the two motors run at the same speed there is no movement of drive shaft and no feed, but as the speed relation is altered the drive shaft rotates. This is analogous to the differential in an automobile with motors in place of the back wheels. Power is transmitted from the drive shaft to the saw frame through a geared pinion and rack. This feed admits an accuracy of control, but the disadvantage is that two continuous-running motors have to be used when one motor really does the work.

The indirect electrical type of feed is driven by one adjustable-

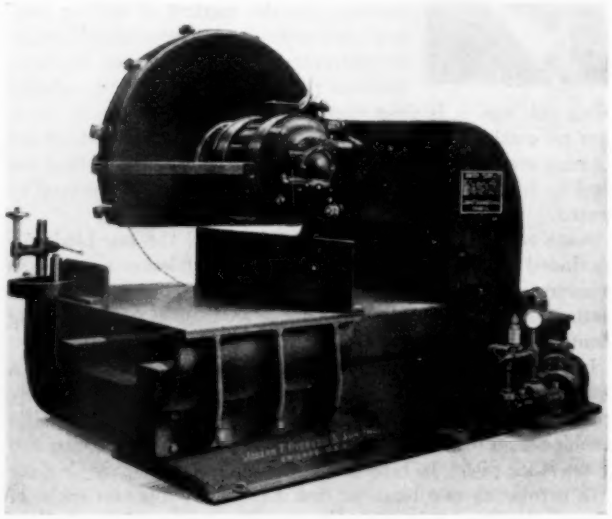


Fig. 2 Direct-Drive Saw With Goose-Neck Frame

speed motor through a set of spur reduction gears to a drive shaft. The current to drive this motor in the feed direction is obtained from a generator coupled directly to the saw-motor drive shaft. However, when the feed motion is reversed the current comes from a separate mill circuit. Thus a variable voltage is used in the cutting direction and a constant voltage on the return stroke. An added safety factor is the tripping of an overload relay, that stops the feed by dynamic braking when the saw motor reaches a value near that at which the saw stalls. Generally, in motor-driven saws having power feed, an ammeter is used to indicate to the operator the rate of feed permissible without stalling the saw motor. As the feed is entirely hand controlled, the saw is therefore subjected to considerable abuse and at times to the very

limit of doing work. In the sawing of varied sections of heavy-shaped steel it is desirable to have considerable range in the speed of the feed movement. This range should be from 1 to 60 ft. per min. in feed direction, having the latter speed on the return stroke.

The Saw Drive

The belt-driven saw used for cutting the heavier Carnegie beam section as now rolled at the Homestead steel works requires a 300-hp. motor. A constant speed 220-volt induction motor is being used running at 720 r.p.m., having a belt pulley mounted on an extended motor shaft. The belt used is of canvas and endless, running at a speed of 7000 ft. per min.

The direct-driven saw used for similar, although lighter, service has a motor designed with a minimum outside diameter as the motor has to pass over the material being cut. The motor used

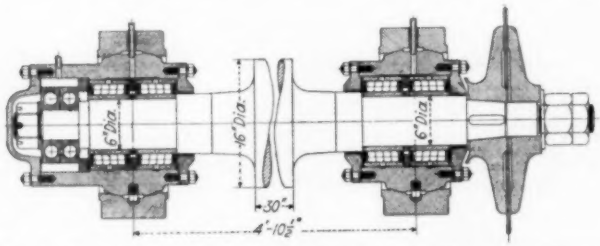


Fig. 3 Saw-Arbor Bearings Applied to a 300-Hp. Belt-Driven Saw

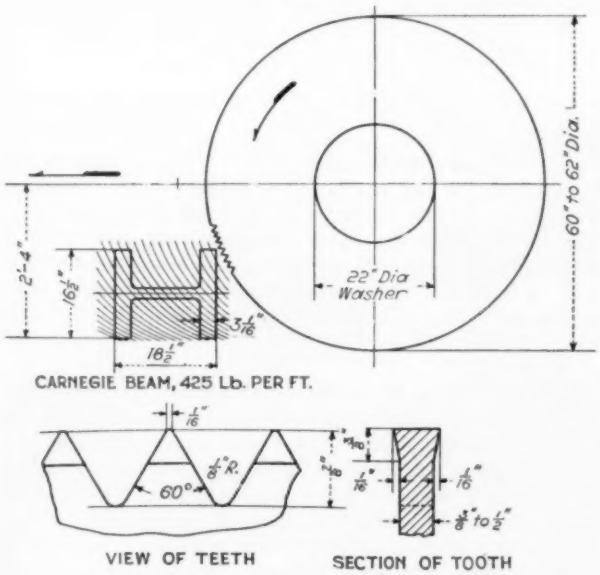


Fig. 4 Saw Blade for Cutting Hot Steel

(Chemical composition: C = 0.30–0.36, Mn = 0.45–0.60, P = not over 0.04, S = not over 0.04, Si = 0.06–0.08. Heat treatment: Heated to 810 deg. cent., held ½ hr. and air cooled, then heated to 500 deg. cent., held ½ hr. and air cooled.)

for driving this type of saw is a specially designed 125-hp. induction motor, running at a speed of 1500 r.p.m.

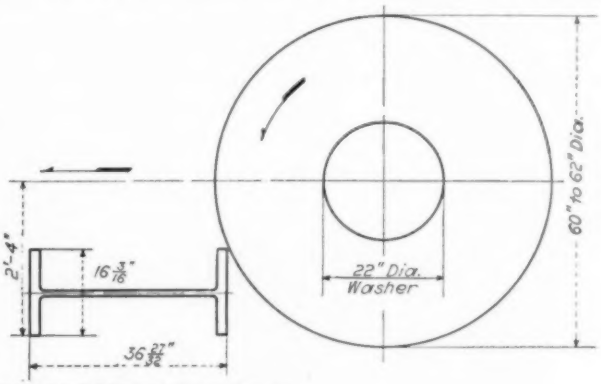
The Saw Arbor

The saw-arbor bearings as applied to the belt-driven saw have gone through stages of trials and changes more than any other part of the saw. At a speed of approximately 1500 to 1800 r.p.m. and a comparatively heavy load, the ball and roller bearings have here found a ready application. This, however, has not been accomplished without some grief, more due to error of application than to any inherent fault of the bearings.

Fig. 3 shows a design which has been found satisfactory when applied to a belt-driven saw arbor. This design admits of linear expansion of saw arbor without binding of bearing races relative to the arbor. It is important that a separate thrust bearing be used for the heavier work.

The Saw Blade

The saw blade is to the operating man the most important part of a saw. Because of rapid wear and replacement it is a source of high upkeep cost. In giving type and size of saw blades



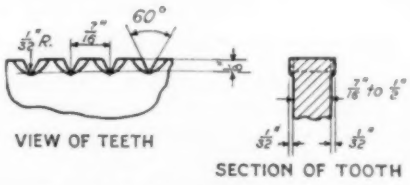


Fig. 5 Saw Blade for Cutting Cold Steel

(Chemical composition: C = 0.24–0.30, Mn = 0.40–0.50, P = not over 0.04, S = not over 0.04, Si = 0.06–0.08, Cr = 0.80–1.00, V = 0.12 to 0.20. Heat treatment: Heated to 865 deg. cent., held ½ hr. and air cooled, then heated to 570 deg. cent., held ½ hr. and air cooled.)

used for cutting heavy structural sections of hot and cold steel, together with the chemical analysis and heat treatment used for these blades, there may be others which have been found to give greater success.

Fig. 4 shows a saw blade used for cutting hot steel and the relation of the saw blade to the beam about to be cut. The teeth of this saw blade are machine punched and later widened with an air-driven swaging tool. Swaging is necessary for running clearance as the sides are not hollow ground, although a saw-tooth grinder is used to shape the teeth. Hardening of the teeth to about one-half their depth prolongs the service of the saw blade. Running at a speed of 18,000 ft. per min., this saw blade is used for cutting red-hot steel.

Fig. 5 shows a saw blade used for cutting cold steel. The early cold saws had blades without nicks, the edge of the blade being smooth. Later it was found that by nicking the cutting edge a faster cutting could be obtained and this is the practice at the present time. The nicks as shown are driven with an air chisel which also widens the edge of the saw blade. The peripheral speed of the saw blade should be from 24,000 to 26,000 ft. per min. for successful cutting.

The saw blades, Figs. 4 and 5, are flat disks which are sheared from a rolled plate. The plate is rolled in a plate mill shortly after a roll change to get as uniform thickness of the plate as possible without having to grind the sides. It should be noted, however, that the grinding of the sides, particularly the hollow-

ground sides, with milled and ground teeth, must necessarily, if correctly done, produce a better balanced blade than a disk not so treated.

All circular saw blades running at a high speed have to be tensioned in order to be safe and to make a straight cut. The tensioning process can be said to be similar to the tightening of the spokes in a bicycle wheel except in a reverse manner. That is, the imaginary spokes in a saw blade, if that term may be used, have to be elongated to distribute and to partly equalize the tangential stress set up when the saw blade is rotating at a high rate of speed. Tensioning has to be done from both sides of the saw blade and with considerable care so as not to dish the blade.

As yet no practical means, fixed rule, or measure, so far as the author knows, has been devised whereby the tension can be put

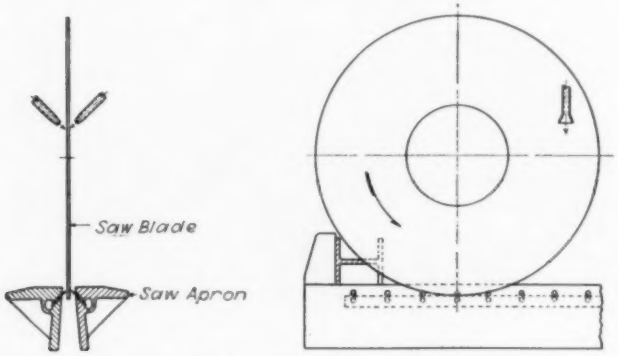


FIG. 6 WATER SPRAYER FOR HOT AND COLD SAWS

into a saw blade with any scientific certainty. At present this operation is left wholly to a sawsmith who with the aid of a hammer, anvil, and a short straightedge does the work according to vision and a sense of touch.

The cooling of the saw blade when sawing cold steel is of considerable importance. Any excessive amount of water flowing down with the saw blade to the line of contact will retard the cutting and also produce a hardened surface at the ends of the material being cut. This hardness was found to be from 45 to 80 Shore scleroscope, and difficulties were experienced when the ends of the beams had to be machined.

A number of tests were made and it was found that when the cooling water is applied as indicated in Fig. 6 no appreciable hardening takes place in beams of less than 300 lb. per ft. Water pressure at 500 lb. per sq. in. is used to cool the saw blade and to clear the teeth or nicks of cuttings.

To determine the relative worth of saw blades a test was carried on over a period of eight months. A total of 103 saw blades were used and grouped as per Table 1. The product sawed was the normal daily run made up from any and all sections. No record was kept of material being removed by the saw blade, neither was any record kept of redressing, although approximately 800 cuts per saw dressing is a fair average. The test was terminated when 49 of the 103 saw blades had been scrapped as unfit for further service.

In all probability if the test had continued the ratio of number of cuts per scrapped blade would not have altered materially. The table shows that, in Group 1, four blades give less than one thousand cuts, two blades give less than two thousand cuts, etc. The greater number of failures occurred before the saw blade had made 5000 cuts, and thus far there seems to be very little difference between the straight and the hollow-ground blade although thereafter the hollow-ground blade has some advantage.

The failure of a saw blade is generally from cracking at the nicked edge, the crack starting at the periphery and then continuing in a more or less radial line toward the center. The cause may be laid to a number of variables such as homogeneity of material and method employed when machining the saw blade, cross-section, and quality of material being cut, as well as the cooling of the saw blade while cutting. Other factors such as the tensioning of the saw blade and the rate of feed are very important items which now are left to the operator's skill.

The life of a saw blade would be prolonged if the saw approach and the rate of feed were automatic and errors on the part of the operator were thus eliminated. The author has no remedy to offer for the difficulty of putting the proper tension into a saw blade; indeed, this problem may be food for thought to any one interested in this subject. As to the quality of the saw blade it goes without saying that a more homogeneous steel having the proper chemical constituents and the highest class of workmanship must necessarily be superior to any lesser quality.

SAW PERFORMANCE

A comparative test was run between the motor-driven belted saw and the direct motor-driven saw, as per Figs. 1 and 2, respec-

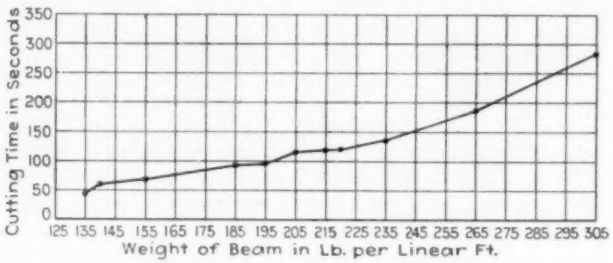


FIG. 7 RELATION OF WEIGHT OF BEAM SECTIONS AND CUTTING TIME

tively. This was done to determine the relative power consumption between the two saws cutting identical beams. The test showed that the belted saw requires approximately from 40 to 45 per cent more power to cut a given beam than does the direct motor-driven type of saw. In sawing, however, the actual cutting time is generally only a small fraction of the running time, and this has to be considered in a fair comparison of the two saws.

CUTTING TIME

A test was run to determine the actual time it takes to saw cold-steel beam sections under normal running conditions. The saw blade used was of the cold-saw type, Fig. 5, 61 in. diameter by $^{7}/_{16}$ in. thick, running at a peripheral speed of 26,000 ft. per min. The saw was of the belted type, as per Fig. 1, driven by a 300-hp. motor. With the hand-operated feed control the saw-drive motor load was held as near the rated motor capacity as possible. The weight of the Carnegie beam sections sawed and the average cutting time in seconds are given in the chart, Fig. 7.

The total cross-sectional area of beams as shown is 663 sq. in. and the total cutting time is 1319 sec. Therefore, the average cutting is approximately $^{1}/_{2}$ sq. in. per sec. The cutting time of this saw was estimated at 2 sq. in. per sec. for sawing cold steel. The saw capacity was based on experience in sawing standard structural sections, and from the foregoing test it is evident that the estimated capacity was too high.

The path of the saw blade as indicated in Fig. 4 shows that cutting takes place in both flanges as well as the web simultaneously, and this together with the heavier section accounts for the difference in time. Sawing beams of a similar section weighing more than 305 lb. per ft. will cause the saw blade to wear rapidly, and as the cutting is retarded the end surfaces of the beam become hard. The limited capacity of the saw which causes the slippage

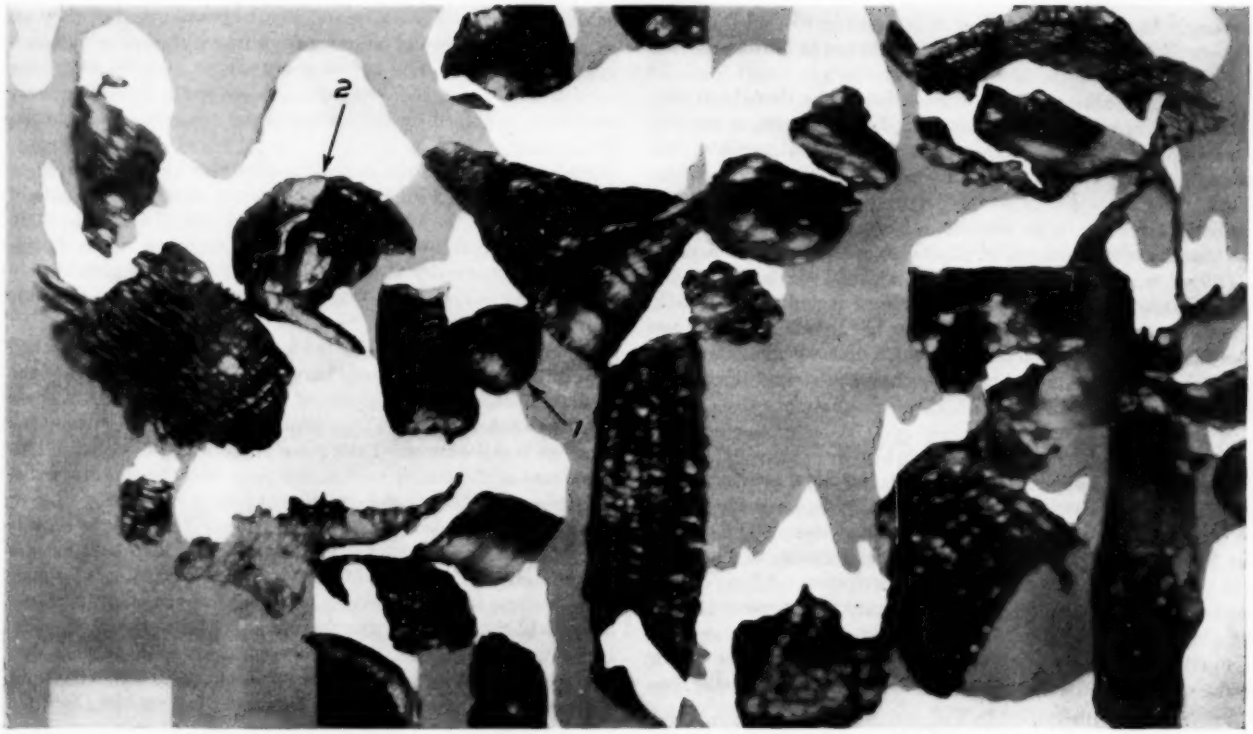


FIG. 8 SAW DUST FROM BEAM HAVING TEMPERATURE OF APPROXIMATELY 1400 DEG. FAHR.



FIG. 9 SAW DUST FROM COLD BEAM

of the saw blade without cutting is therefore indirectly the reason for the hardening of the beam. From this it can be seen that the cutting time is a very important factor, and a smaller saw cannot be used by allowing more time for the cutting.

For successful sawing the cutting capacity should be at least 1 sq. in. per sec. to avoid hardening of the beam ends. Added horsepower for the saw-motor drive will increase the sawing capacity, although there must necessarily be a maximum of

saw-blade contact after which it is impossible to cool the blade sufficiently to maintain the relative difference in hardness which alone makes it possible to hot-saw steel.

In sawing beams at temperatures of not less than 1400 deg. fahr., and of sizes as per the chart, Fig. 7, using a saw, as per Fig. 1, having a toothed saw blade running at 18,000 ft. per min., the average cutting time is approximately 1/8 of that given in the chart.

All the Carnegie beam sections as now rolled are sawed while the beams are red hot. The 425-lb. section having an area of 124.99 sq. in. can thus be cut in from 45 to 60 sec.

To cold-saw all Carnegie beam sections including the 425-lb. beam successfully, a more powerful saw has to be built. The saw blade should also, if possible, be of such a diameter that the blade will leave the first flange before entering the second during the sawing operation.

Principles of Sawing

A difference of opinion exists, which has been previously referred to in this article, as to what actually happens when a revolving disk cuts its way through a piece of steel. The process has been called friction, burning, fusing, cutting, or abrasion, and perhaps other names equally as descriptive. All, no doubt, have seen the large volume of brilliant sparks that passes off during the sawing operation. These sparks and cuttings, coming from either a hot or a cold saw, generally go by the name of saw dust. A very marked difference may be observed in the two kinds of chips.

To present some light on this subject magnified photographs of the chips removed by the saw blade are shown. A structural beam having a temperature of approximately 1400 deg. fahr. was sawed. The saw blade used was of the hot-saw type with teeth as shown in Fig. 4 running at a peripheral speed of 18,000 ft. per min. Saw dust collected from the cuts was sorted and the chips photographed, the magnification of the originals being 7.2 diameters. Fig. 8 shows the irregular shape of the chips.

Although the beam had a high initial temperature, additional heat was generated by the friction of the saw blade to cause oxidation along the edges of the chips. This oxidation caused the globules to form after the chips were removed from the beam. A typical globule may be noticed at the point of arrow 1. Some of these globules expand until they explode, and a case of this may be noticed at arrow 2. The brilliant sparks seen in sawing steel are the globules as they form, expand, and explode during their travel.

The chips when examined under a microscope will show on the one side lines caused by the face of the saw blade in its travel similar to the lines left at the end of the beam after sawing. These lines are in the line of travel of the saw blade and lengthwise of the chips and not crosswise, which may have been expected because of the toothed blade. On the side of the chip that faced the beam when sawed a tearing action is indicated, which shows very clearly on the photograph. It is therefore evident that the chips were not in a molten state when they were removed from the beam.

For comparison a cold structural beam was sawed having a temperature of approximately 60 deg. fahr. before sawing, the saw blade used being of the cold-saw type with nicked edge, as per Fig. 5, running at a peripheral speed of 26,000 ft. per min. The saw dust was sorted and photographed in the same manner as that from the hot saw. Fig. 9 shows the chips, which in this case resemble flakes. The heat produced by a cold saw is caused entirely by friction, and as compared to the hot saw a more rapid oxidation takes place because of the smaller chips or flakes. There are not any globules shown in this photograph as they were eliminated when the saw dust was sorted. These globules do exist, however, and are given off in a large volume, but they always either explode and become grit or fuse with each other forming a solid mass. The reason there are no globules along the edge of the flake is that the flake adheres to the face of the saw blade and is chilled by the cooling water before the globule can be formed.

The saw blade in its travel has also here caused grooves or lines which under a microscope show very clearly on the side of the flake facing the blade. On the other side of the flake there is no indication of the saw blade, instead a tearing action is shown. It is also noteworthy that the flakes are curved approximately to the same radius as the saw blade in line of travel. An oxide film covers the flakes.

If two pieces of steel having the same characteristics are rubbed against each other sufficiently hard, the most work and hence the greatest heat will be at the line of contact. Now if one of these pieces is cooled and heat thus eliminated by outside means, the heat line is shifted toward the piece of steel not so cooled. This is because of the stored heat in this piece and the continued disturbance of the fibers and consequent heat generation. In sawing, for a similar reason this heat line is located somewhere outside the face of the saw blade. The distance is very small and approximately the thickness of the chip. This explains why lines from the saw blade can be seen on the side of the chip that faced the saw blade as well as the tearing action of the opposite side. By actual measurement the average thickness of the hot-saw chip is about 0.01 in. and those from the cold saw 0.003 to 0.004 in.

The hot-sawing action of steel is essentially a rubbing, heating, and a tearing action. The necessary rubbing to create sufficient heat for sawing depends on any prior heat in the steel to be sawed, the feed pressure, and the speed of the saw blade, as well as the continued cooling of the edge of the blade. Hot-sawing of steel is possible when the friction between the steel to be sawed and the saw blade exceeds the tensile strength of the heated steel at a distance from the saw blade equivalent to the thickness of the chip.

Discussion

George C. Farkell.[2] The possibilities of hot-sawing steel sections so obviously depend upon the ability of the blade to withstand the work that this part of the hot saw deserves the most careful attention. The remarks made by the author on the principles of hot-sawing merit the most careful consideration and indicate a tearing action at the time the blade is in contact with the hot steel. This action seems to be similar to the cutting action of milling operation on cold steel, so that the fundamental principles involved are probably much the same.

The variables when hot-sawing which have an influence on the life of the blade, listed roughly in the order of the effect, are the temperature of the steel being sawed, the cooling of the blade, the rate of cutting as determined by the feed, the circumferential speed of the blade, the shape and thickness of the tooth, the chemical and physical properties of the blade, and the shape of the section being sawed. A thorough investigation of these variables to determine their effect on the life of the blade would give very valuable information for the designer as well as the operator of hot saws.

In the hope of adding something to the information now available on the action of the hot saw blade when sawing steel, the writer wishes to present the following practical results observed when sawing large rounds of from 20 to 95 sq. in. in section intended to be manufactured into large seamless pipe. Sliding frame saws are used, the blades of which are 50 in. in diameter

[2] Superintendent, Rolling Mills, National Tube Company, Lorain, Ohio. Mem. A.S.M.E.

by 1/4 in. thick, running at a peripheral speed of 16,200 f.p.m. when new, and at 13,500 f.p.m. when the blade is finally discarded at 41 1/2 in. in diameter. These blades are furnished by experienced saw manufacturers whose recommendations for physical and chemical properties and the tensioning and grinding of clearance in the sides of the blade are accepted. The form of tooth furnished, however, is specified by us and is shown by Fig. 10.

With these blades it is possible to saw the largest rounds at a rate of 6 sq. in. per second as long as the teeth are kept cool.

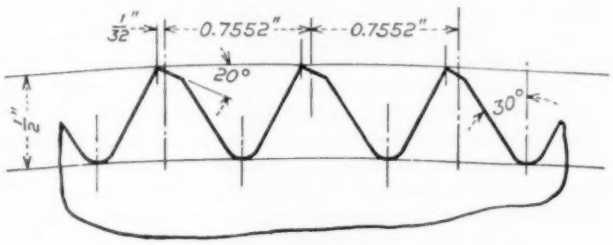


FIG. 10

As soon, however, as there is evidence of heating of the points of the teeth, as shown by the tooth being red when entering the cut, the power required rapidly increases and feed must be reduced. The blade must then be changed. Under such conditions the teeth will be found somewhat in the condition shown by Figure 11. Such a blade will require a reduction diameter of about 1/4 in. to bring back the tooth to the original contour.



FIG. 11

An examination of the cuttings taken at different times during which a blade is in service indicates that the cuttings up to the time the blade gets dull have very much the same characteristics, all being well represented by Fig. 8 of the paper. This fact indicates that the sawing action is very similar up to the point when the tooth starts to lose its cutting ability and that when this takes place deterioration is rapid. This would, it is believed, indicate that the form of tooth is important and that study should be given to this factor to get the most efficient contour for cutting hot steel which is best adapted for effective cooling.

The temperature of the piece being sawed has obviously an important bearing on the problem and is such that it is doubtful if results obtained on the average life of a large number of blades are of any great value in arriving at a conclusion. It is frequently necessary, regardless of blade life, to saw material which is very cold due to delays in getting the product to the saws. Under such conditions, blades cannot be changed, sawing approaches a cold-sawing operation, and the life of the blade is short.

It is believed that best performances are better indicators of service, because these results probably indicate that more uniform conditions of temperature prevailed. As an indication of the effect of temperature, we find that under best conditions we are able to saw 16,000 sq. in. per 1/16 in. of blade diameter when sawing 5 1/4 in. diameter rounds which require about one-half minute longer from the pits to the saws as compared with 28,000 sq. in. for 1/16 in. of blade diameter when sawing 11 in. diameter rounds. The 11 in. is hotter, not only on account of shorter time from pits to saws and having less radiating surface, but also on account of sawing fewer units.

Effective cooling of the blade is vital. In addition to the methods referred to by the author, we have made provision so that the bottom edge of the blade runs in a pool of water with a high pressure jet pointed tangentially at the teeth on the bottom of the blade. With a blade properly cooled, changed as soon as indication of heating of the points of the teeth is observed, we have not found any difficulty in keeping the teeth free of cuttings, although at times of emergency when blades cannot be changed the spaces between teeth are at times found plugged with cuttings which cannot get out due to the upsetting of the points of the teeth.

An investigation in the art of hot-sawing steel as practiced at the various works will show considerable variations, which would indicate that careful investigations to determine blade characteristics for greatest life would give very interesting results, in regard to which the writer has found but very little accurate information available.

J. J. DUNN.[3] A statement of experience in the limited field of hot-sawing rounds may be of value. The Mannesmann process for seamless tubes is now requiring a very large tonnage of rounds, varying in diameter from 3 in. to 12 in. and in weight from 25 lb. to 5000 lb. and over. As an average the hot saw for this work must make 160 cuts per hour on 4 1/2-in. diameter steel. It must be capable of cutting 3-in. steel at a rate of at least 33 cuts per minute, and have capacity for the largest diameter required.

At one bar mill the sliding-frame type of saw was used for about ten years. When this mill was rebuilt, at a time when this type of saw was growing in favor in preference to the pendulum and drop saws, careful consideration of the subject led to the selection of the drop type for the new installation. The reasons for the selection were:

(1) Saw chips could not be kept out of the slide of the saw frame previously used. This caused delays and increased cost of upkeep
(2) Variable resistances to motion made it difficult to maintain the rate of cutting required by the smaller diameters
(3) The cost of the sliding frame type was greater than that of the drop frame type.

Reasons (1) and (2) could probably have been overcome by good design, but nearly 15 years' use of the drop-frame design selected has proved that no serious error was made in the choice.

The design of these drop-frame saws embodied no unusual features, with the exception that they were considerably heavier than was then the practice. The following data regarding details of design and service may be of interest:

Motor. Induction type, 150 hp., three phase, 60 cycle, 2200 volts, 695 r.p.m., full-load speed. Driving pulley 24 in. diameter by 18 in. face, giving a surface speed of 4365 f.p.m.

Saw Arbor. The saw arbor has journals 4 3/4 in. in diameter by 14 in. long, spaced 28 in. centers. These run in self-aligning babbitted shells; lubrication is the gravity type with 12 ft. head. The life of these bearings is about 225,000 tons, requiring 1,500,000 cuts.

Two types of roller bearings have been tried and discarded because they gave a life less than half that of the babbitted bearings without any compensation advantages.

The pulley on the saw arbor is 12 in. in diameter. It is grooved parallel to the axis with flutes 3/16 in. wide by 1/8 in. in depth, spaced about 5/8 in. center to center. These grooves are effective in reducing the belt tension.

Belt. The belt is of selected leather 14 in. wide by 38 ft. 6 in. long. The average life is about 90,000 tons.

Saw. The saws are 52 in. diameter, 5/16 in. gage. They are used until the diameter is reduced to 46 in. Peripheral speed

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of the saws varies from 19,000 to 16,700 ft. when the saw is discarded. The saw tooth is a sharp V, $^3/_4$ in. pitch and $^{19}/_{32}$ in. in height to the bottom of fillet of $^7/_{32}$ in. radius. In grinding no attempt is made to maintain the form with accuracy. Saws are ground after eight hours' service. This represents cutting an average of 16,500 sq. in. of metal.

Cost of Maintenance. The heaviest item in the cost of maintenance is repairs to the hydraulic feed mechanism. The average total maintenance cost for six months' period was $0.023 per ton, or $0.19 per 1000 sq. in. of metal cut.

J. M. Lewis.[4] The friction saw blade operating at a rim speed in excess of 19,000 f.p.m. cuts steel while cold by a melting action. Each point on the blade periphery rubs against the work, creating friction which causes both the blade and the work to be heated. As each point on the blade passes out of the work, it is cooled off in the air before passing down again past the work, while the work is continuously being heated by the blade periphery.

It is essential to keep the blade periphery cold as it passes down again past the work. It should not become red hot on the periphery. If it shows a red heat, the cooling should be assisted by applying cooling water in some manner directly against the rim, in sufficient volume and pressure to penetrate the air film surrounding the blade periphery, so as to keep the blade cold during the entire cut.

In cutting low-carbon structural steel the blade heats the material to a putty stage rather than a liquid stage, and to assist the removal of the heated metal in this stage, it is necessary to indent the periphery of the blade. High-carbon steels (above approximately 0.35 per cent carbon) change readily to a liquid stage under the action of a friction saw blade and the blade periphery need not be indented.

If the carbon content of the steel being cut is high enough to allow a tempering action, the sawed surfaces will be tempered to a depth of only about $^1/_{64}$ in. For low-carbon steel there is no hardening effect on the sawed surfaces.

The usual method of redressing friction saw blades for cutting low-carbon steels has been to nick the periphery with a cold chisel. A newer and better method has been developed whereby the blade periphery is hobbed by a combination serrating and topping spiral hob which forms a uniformly indented periphery and also trues the periphery concentric with the bore of the blade, removing the skin surface from the rim, which may have developed surface cracks during the sawing operation. Side-milling cutters trim the burred sides of the blade during the hobbing operation, leaving only a small amount on each side of the blade for clearance.

S. M. Weckstein.[5] From the discussions which have preceded and referring to Fig. 3 in the paper, the writer wishes to bring out a few points on the design of hot and cold saws and the bearing application problem which has come to attention within the last few years. He will limit his remarks only to the design of the arbor and will make no reference to the blade.

In our own mills we had considerable difficulty with the operation of the bearings on the saw arbor. In order to eliminate costly shutdowns, we decided to make some tests on the use of Timken bearings. After considerable development work, we worked out a design which has been in operation for over three years and has given very satisfactory service. Based on the experience and the very satisfactory operation, quite a number of steel mills throughout the country have adopted this design and are using Timken-equipped hot and cold saws. In the application of bearings on saw arbors, quite a number of points must be considered:

(1) High speed of operation
(2) Extremely heavy load, which is also a shock load
(3) Deflections in the saw-arbor spindles due to the extremely high belt pull and the overhung saw blade.

It has been our experience that most of the bearing difficulties encountered have been due to deflection in the saw arbor and consequent unbalanced loads. One of the larger mills in the country has quite a battery of Timken-equipped saws, and it advises that the secret of the proper operation of these saws is the periodic checking of the saw arbor to make sure that it is not bent and unbalanced in any way.

Another important point which should be considered in the design is the provision for a proper lubrication system. We have found that, due to the large size bearings required and due to the high operation speeds, the best type of lubrication system is a continuous lubrication system supplying a stream of oil of $^1/_{16}$ to $^1/_8$ in. in diameter to the bearings by gravity feed. This oil is taken away from the bearings and pumped from the system back to the gravity tank.

It has also been our experience that a number of jobs which operated originally on plain bearings and were changed over to anti-friction bearings gave considerable trouble both on the plain bearings and anti-friction bearings until such a lubrication system was worked out.

W. Trinks.[6] At Carnegie Tech we have a course in mechanical engineering of steel works. Among other machinery, the students design a hot saw. It is comparatively easy to determine the smallest feasible diameter of the saw blade, but difficulty arises in determining the thickness of that saw blade. It is obvious that the thicker the saw blade, the more material is removed by it, and the greater is the power requirement. Consequently the saw blade is always made as thin as possible, without having it go around a corner. The question which the writer would like to have answered is, How thin can the saw blade be made in a given case without having it go around a corner? Beams must be sawed so straight that no further finishing of the ends is needed.

The speed of cold saws is greater than that of hot saws, because hot saws can cut at a lower speed than cold saws. Has any one tried to run a hot saw as fast as a cold saw? If so, what happens?

Author's Closure

The information as to the minimum thickness of a given diameter of saw blade is not available. The user generally arrives at the most economical saw-blade thickness by trial, and information obtained thus is based on the average of both the temperature and the section of the material being sawed.

One user desiring to increase the diameter of a $^7/_{16}$-in. thick saw blade from 60 in. to 66 in. to allow for more wear, later found it necessary to decrease the diameter to 62 in. to avoid "going around the corner." The material being sawed was red-hot structural steel up to 425 lb. per lineal foot. The most desirable ratio of thickness to the diameter of a saw blade will generally be found between 0.005 and 0.009 to 1, for sawing both hot and cold steel. This ratio of thickness to diameter will serve as a guide in arriving at a thickness giving the best result. In regards to speed it has been found that a saw blade, when sawing cold steel running at 20,000 f.p.m., has a very much shorter life than a blade when running at 24,000 f.p.m. or over.

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[5] Industrial Equipment Engineer, Timken Roller Bearing Company, Canton, Ohio. Jun. A.S.M.E.
[6] Professor of Mechanical Engineering, Carnegie Institute of Technology, Pittsburgh, Pa. Mem. A.S.M.E.

IS-52-11

Forces Acting on Roll-Neck Bearings

Report of the Special Research Committee on Heavy-Duty Anti-Friction Bearings

By W. TRINKS,[1] CHAIRMAN, AND J. H. HITCHCOCK,[2] RESEARCH ASSOCIATE

FOR the intelligent design of anti-friction bearings for application to roll necks, the average load to which the bearings are subjected must be known with reasonable accuracy, and the maximum load to which the bearings will at any time be subjected must be known at least approximately. The reasons for these requirements are the following: Unless flaws exist in one or more of the balls, the final failure of a bearing is usually caused by fatigue due to many applications of the average load. A bearing can withstand occasional loads considerably higher than the average load, but its capacity to withstand such loads without injury is not unlimited, and an excessive load might bring about bearing failure. Consequently, we must know the average load accurately and the maximum load approximately.

There are now available five methods for determining the bearing loads on roll-neck bearings. The first, the use of Brinell balls and plates, is suitable for the determination of maximum loads only. The hydraulic cylinder, second, and the strain gage, third, are suitable principally for the determination of average loads. The fourth and fifth methods, of comparatively recent development, are the telemeter, or carbon pile, and the electrical condenser. These latter two methods are practically free from inertia effects and may be used to determine either maximum or average loads.

The efforts of the committee up to the present time have been directed largely toward the collection, coordination, and analysis of existing data. It has been attempted to obtain and evaluate all existing data on this subject, from tests in other countries as well as our own.

Valuable data on hot-rolling steel shapes were obtained from the results of tests made by Dr. Johann Puppe, as published in his book, "Untersuchungen über Walzdruck und Kraftbedarf beim Auswalzen von Knuppeln, Winkeln, U and I Eisen." Equally valuable data on cold rolling were found in a report of tests made by Pomp and Seibel, published in "Mitteilungen aus dem Kaiser-Wilhelm Institut für Eisenforschung" under the title, "Walzdruck und Walzarbeit beim Kaltwalzen von Metallen." The results of extensive tests made by the Timken Roller Bearing Company constituted a further source of data. Recent tests made by this company have not been completely evaluated at the time of writing this paper. The results of these tests can be expected to add a great deal to our knowledge of rolling pressure. Tests results were also obtained from S-K-F Industries, Inc., Rollway Bearing Company, Jones and Laughlin Steel Corporation, National Tube Company, and Republic Steel Corporation. Because this paper is read under the auspices of the Iron and Steel Division, the data presented herein apply only to iron and steel. Valuable data have been and still are being collected on other metals, including copper, brass, zinc, and aluminum, which, for the reason stated, will not be included in the present paper. It should be mentioned that test data were being accumulated rapidly in midsummer when this paper was written. Consequently, the present available knowledge of rolling pressures is considerably in advance of that incorporated in this paper. At the time of writing, sufficient information has been compiled, not only to allow some conclusions to be drawn, but to indicate the direction in which additional tests should be made.

Review of Forces in Rolling Mills

For a thorough understanding of the forces to which roll necks are subjected, a review is advisable of the nature of the forces which occur in rolling mills.

A rolling mill deforms material between the rolls by means of compression. The resultant force which is exerted on the material being rolled, which is hereafter called the bar, is the compression force which acts at right-angles to the direction of rolling. This statement refers only to rolling in one single stand, and in that case, only to rolling at constant speed. If rolling is done at varying speed, forces in the direction of rolling enter into the problem through inertia effects. If the rolling is done in several stands continuously, forces which cause tension or compression in the bar may exist between successive stands. However, if we limit ourselves for the present to rolling in one stand at one constant speed, the force which is exerted between the rolls and the bar is solely a compression force acting at right-angles to the direction of rolling. In that case the bar is free at both ends and cannot, in general, take up any force in the direction of rolling. As exceptions to this statement, we must note the very small force which is caused by the friction of the rollers on the roller table, and the very small force which is due to the speed change of the bar, which enters the rolls at a low speed and leaves at a slightly higher speed. Both of these forces are usually less than 1 per cent of the compression force, and can therefore be neglected. All of the foregoing statements refer solely to the resultant force between the rolls and the bar. It should be fully understood that local tangential forces are exerted at various points of the contact area between the rolls and the bar; and it must also be understood that such tangential forces cancel each other, leaving in effect only the resultant force at right angles to the direction of rolling. This resultant force which acts between the rolls and the bar will hereafter be called the separating force.

The separating force may be considered the product of the projected contact area by the average pressure which is distributed over that area. The projected contact area is that area, common to the bar and the roll, which would be seen through the roll if the latter were transparent. The projected contact area

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The other members of the committee are:

S. M. Weckstein, Industrial Equipment Engineer, Timken Roller Bearing Company.

W. R. Clark, Works Manager, Bridgeport Brass Company.

H. H. Talbot, Chief Engineer, United Engineering and Foundry Company.

H. E. Brunner, Chief Engineer, S-K-F Industries, Inc.

R. C. Stiefel, Vice-President, Aetna Standard Engineering Company.

S. G. Koon, Associate Editor, *The Iron Age*.

G. C. Farkell, National Tube Company.

Presented at the Fourth National Iron and Steel Division Meeting, Chicago, Ill., September 22 to 24, 1930, of The American Society of Mechanical Engineers.

Note: Statements and opinions advanced in papers are to be understood as individual expressions of their authors, and not those of the Society.

can be determined by simple methods of descriptive geometry, but its determination does not properly constitute a part of this paper. The determination of the average pressure, however, and of its relation to various factors, constitute the problem with which we are concerned.

There is an essential difference between the hot and cold rolling of steel due to the fact that the hardness of steel is very much greater at low temperatures than at usual temperatures of hot rolling. Consequently, much greater average pressures are encountered in cold rolling than in hot rolling, which result in the

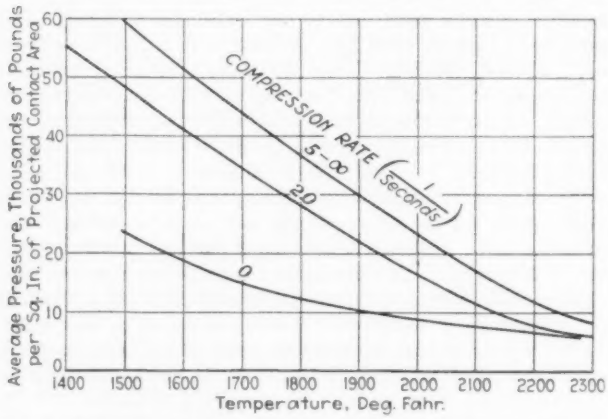


FIG. 1 RELATION BETWEEN AVERAGE PRESSURE, TEMPERATURE, AND COMPRESSION RATE FOR MILD STEEL, 0.10 TO 0.20 CARBON
(For square sections and values of contact length/thickness less than 1.)

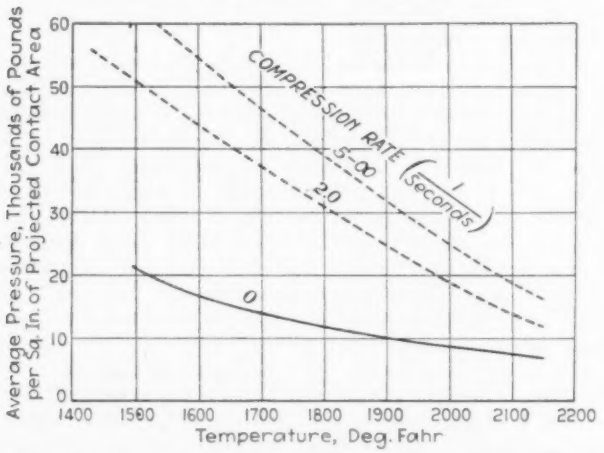


FIG. 2 RELATION BETWEEN AVERAGE PRESSURE, TEMPERATURE, AND COMPRESSION RATE FOR HARD STEEL, 1.00 CARBON
(For square sections and values of contact length/thickness less than 1.)

deformation or flattening of the rolls to a considerable extent. This effect will later be considered in detail.

HOT ROLLING

The average pressure in hot rolling depends upon the following variables:

a Temperature of bar
b Composition of bar
c Compression rate[3]
d Internal friction of bar
e Friction between bar and rolls.

[3] The compression rate is determined by means of the formula draft/(thickness × time) in which the time is that required to pass through the angle of contact. The unit in which the compression rate is expressed is therefore 1/seconds.

The effect of temperature upon the average pressure has long been known. This is also true, to some extent, of the effect of compression. The influences of compression rate and of the internal and external friction are not well known; in fact, the influence of external friction was not even suspected.

A great mass of more or less fragmentary data has been collected interconnecting these variables, and the problem immediately arose to construct an equation or a chart which would represent their relation. But the preparation of a chart connecting six variables presents great difficulties, and especially so if the data are very meager. Consequently, we shall have to be content

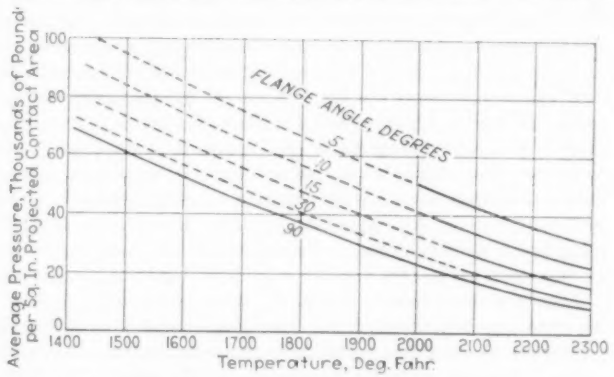


FIG. 3 EFFECT OF FRICTION BETWEEN THE BAR AND THE ROLLS AT HIGH COMPRESSION RATES FOR MILD STEEL, 0.10 TO 0.20 CARBON
(For values of contact length/thickness less than 1.)

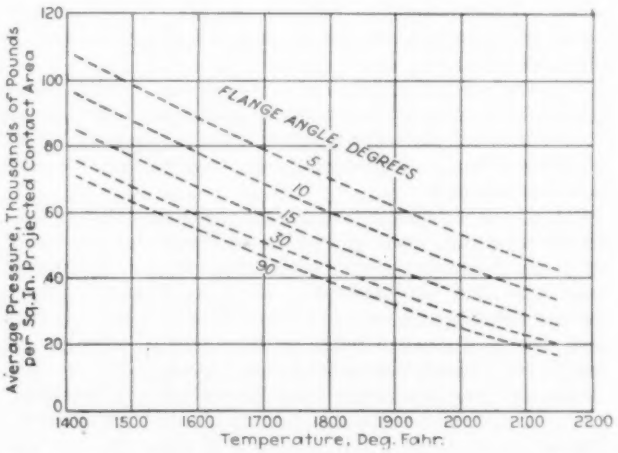


FIG. 4 EFFECT OF FRICTION BETWEEN THE BAR AND THE ROLLS AT HIGH COMPRESSION RATES FOR HARD STEEL, 1.00 CARBON
(For values of contact length/thickness less than 1.)

for the present with charts showing only partially the relation between the average pressure and the factors upon which it depends.

The curves of Fig. 1 illustrate to some extent the effect of temperature and compression rate upon the average pressure developed in hot-rolling mild steel. The curves of Fig. 2 show the partial influence of the same factors for high-carbon steel of the composition indicated. In each case the curves apply to sections which are nearly square, rolled with the compression rates indicated. The curve of zero compression rate in each of these figures represents the static resistance to compression. The values used in plotting these curves were taken from "Walzwerkswesen," by J. Puppe.

The curves of Figs. 3 and 4 indicate to some extent the effect of external friction. In these latter figures, the values given in

the column at the right-hand side refer to flanged sections and denote the size of the flanged angle of the section being rolled. They apply therefore to the roughing and finishing passes in the rolling of I-beams, channels, and rails. This angle may be defined as the inclination of the inner wall of the flange from the vertical. It is obvious that as the inner wall of the flange approaches the vertical, the flange angle becomes smaller and the force due to external friction increases. The approximate magnitude of the resulting increase in pressure can be observed in the figures. The curves for which the flange angle is 90 deg. apply to rectangular, almost square, sections, and are the same as the curves of high-compression rate in Figs. 1 and 2. In each figure shown, the dotted portions of the curves represent data of uncertain accuracy, or approximations which have not yet been checked by test data.

In each figure shown, the dotted portions of the curves represent data of uncertain accuracy, or approximations which have not yet been checked by test data.

The influence of compression rate upon the average pressure is due to the internal friction, or viscosity, of the bar being rolled. Steel above the critical range of temperature where all hot-rolling is being done resists change of shape in the same manner as tar, pitch, clay, or plasticene, and its resistance increases with the rapidity with which the change is produced. The compression rate, as indicated in footnote (3), is a measure of the rapidity with which the change is produced. The resistance to change, or internal friction, therefore increases with increasing compression rates; and since the force due to internal friction must be overcome by rolling pressure, the latter also increases. This proportional relation between average pressure and compression rate does not apply, however, to all values of the latter. For values of compression rates equal to or greater than about 5 per second, the average pressure remains sensibly constant. This is probably due to the fact that at high compression rates the bar being rolled is heated considerably, and thereby softened, by the compression.[4]

The effect of internal friction upon the average pressure is very marked in the rolling of thin strips, for the following reason: The surface layers of the bar being rolled which are in immediate contact with the roll surface are considerably impeded by that contact from lending themselves to the general change of shape. As a result, a very large part of the deformation must occur within the inner layers of the bar, against the restraining influence of the outer layers. It is at once evident that, as the bar becomes thinner and thinner, a point is reached at which the deformation becomes very difficult. Consequently, the average pressure for rolling very thin sections is greater than for thick sections, other effects being equal. It is very difficult to obtain test data showing the effect of internal friction as a separate and distinct factor from external friction, because each of these friction effects is present in some degree in every case. Also, discrepancies are often found in test data, such as that shown in Fig. 5. In this figure, there is no apparent reason why the two curves for strips should not coincide, and yet there is a difference between them of approximately 60 per cent of the higher value. This illustration is typical of many baffling results which are found, and the work of this committee will be incomplete until sufficient test data are obtained to make known the reasons for such wide variations. It is felt that all such apparent discrepancies will cease to exist when all the facts of the case are known.

The external friction between the rolls and the bar being rolled

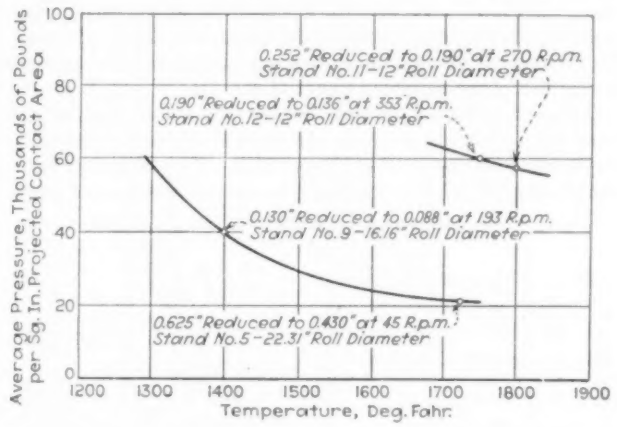


FIG. 5 TYPICAL DISCREPANCIES IN TEST RESULTS, SHOWING THE NEED FOR FURTHER TESTS
(Mild steel strips rolled in continuous mills.)

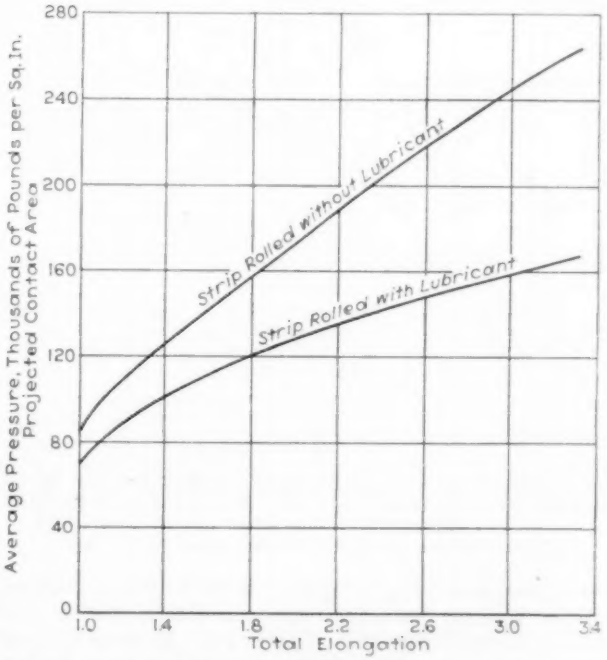


FIG. 6 EFFECT OF FRICTION BETWEEN THE BAR AND THE ROLLS AT HIGH COMPRESSION RATES IN COLD ROLLING MILD STEEL (0.10 CARBON) BASED ON RIGID ROLLS
(For values of contact length/thickness between 6.0 and 17.0.)

has an effect upon the roll pressure which was not suspected until recent test data brought it to light. The force due to this friction acts, of course, in a direction tangential to the surface of the roll. As long as both ends of the bar are free (that is, unless the bar is rolled in a continuous mill), the force due to external friction must be overcome by the rolling pressure alone. The latter must act in a vertical direction, as explained earlier, and at some distance x from the vertical center line of the mill. The additional

[4] The statement that the specific resistance to compression does not grow perceptibly beyond compression rates of 5 per second may be doubted by those who have lived with rolling mills and have watched the destructive effects of higher and higher speeds, particularly in blooming mills and other mills of heavy type. However, it is practically certain that the destructive effect of high speed is due entirely to stress caused by impact. If a small but long hot bar is suddenly gripped by the rolls, only the front end is accelerated the maximum amount because the bar is ductile and can be stretched. If, on the other hand, a heavy ingot is gripped by the rolls and accelerated, practically all of the mass is accelerated instantaneously, which produces enormous stresses.

rolling pressure required to overcome the external friction is then equal to

$$\frac{\text{external friction force} \times \text{radius of roll}}{x}$$

The length x of the moment arm at which the resulting separating force acts varies with the shape of the section rolled, with the uniformity of temperature of the bar, and with other factors. For rectangular sections under ordinary conditions it is approximately 2/3 of the projected length of contact.

It should be pointed out here that the effects of internal and external friction are cumulative. As the internal friction increases, the pressure increases, as has been noted. The tangential force due to external friction is a function of the pressure, and must

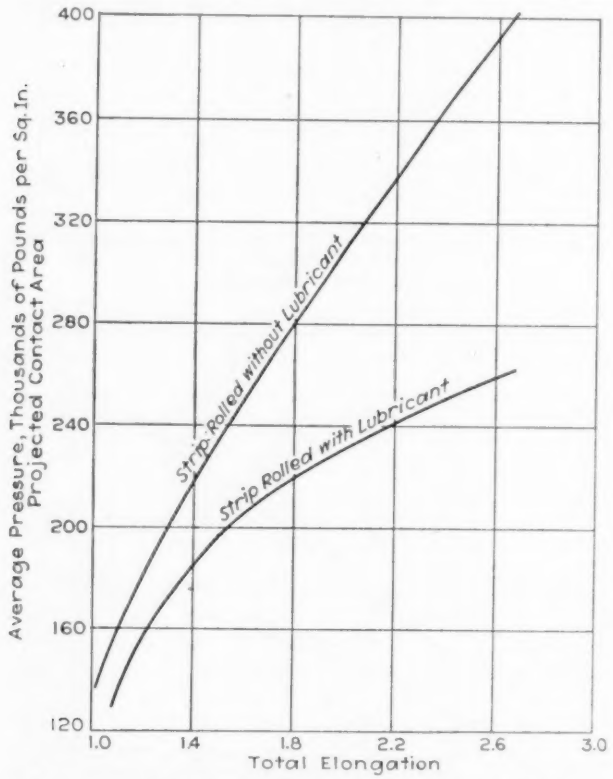


FIG. 7 EFFECT OF FRICTION BETWEEN THE BAR AND THE ROLLS AT HIGH COMPRESSION RATES IN COLD ROLLING MILD STEEL (0.63 CARBON) BASED ON RIGID ROLLS
(For values of contact length/thickness between 6.0 and 15.0.)

increase also as a result of increased internal friction. But we have just observed that increase in external friction must be overcome by increased pressure. Consequently, the pressure must continue to increase until equilibrium is reached, which means that an increase in internal friction produces a more than proportional increase in pressure. Examples of test data illustrating this point will be seen in the curves for flanged sections, Figs. 3 and 4, and in Figs. 6, 7, 12, and 13 under cold rolling.

In the very near future, test data on the hot rolling of alloy steels of various compositions are expected to be obtained. In the meantime, the data given in Table 1 will be of value. The table gives the ratio of the power required for rolling various alloys to that required for rolling mild steel at the same temperature. Inasmuch as the pressure is approximately proportional to the power required, these ratios can be used to determine the approximate pressure required for rolling alloy steels of the compositions given. The ratios were computed from a chart published in the Dec. 12, 1929, issue of *Stahl und Eisen*.

COLD ROLLING

In cold rolling steel, the average pressure depends upon the following factors:

a Elongation from annealed condition
b Compression rate
c Composition of bar
d Internal friction of the bar
e Friction between the bar and the rolls
f Modulus of elasticity of the roll (indirectly).[5]

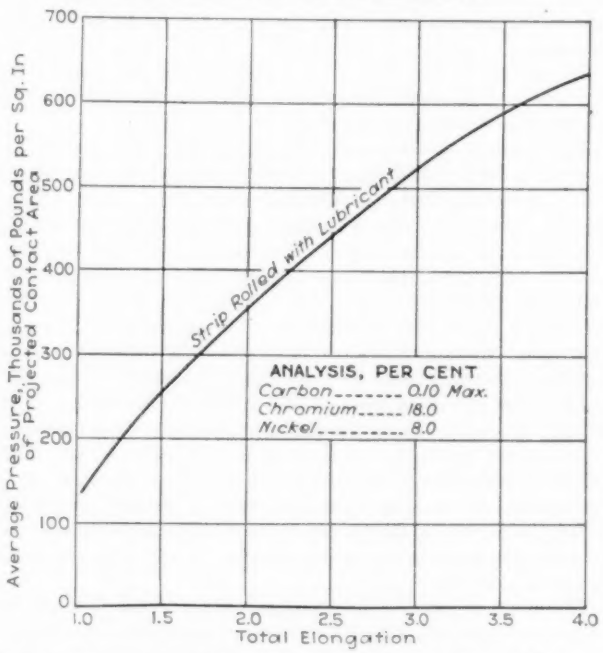


FIG. 8 PRESSURE FOR ALLOY STEEL COLD ROLLED UNDER TENSION AT HIGH COMPRESSION RATES BASED ON RIGID ROLLS
(For values of contact length/thickness between 2 and 5.)

TABLE 1

Material				Rolling temperature, deg. fahr.	Ratio of power required to that of mild steel at the same temperature
Fe	Ni	Cr	Mo		
100	..	..	..	2300	1.00
50	50	..	..	2120	1.62
20	63	17	..	2102	2.31
16	62	15	7	2102	3.86
..	80	20	..	2102	3.08
..	100	..	..	2102	1.27

With the exception of the first and the last, these factors are the same as those which determine the average pressure in hot rolling. The statements previously made regarding the extent of our knowledge of them apply here with equal force. Because of the need for further data, we shall have to be content, as before, with the charts showing only partially the relations between the average pressure and the factors which determine it.

The curves of Figs. 6, 7, and 8 are based on the cold rolling of low-carbon steel, high-carbon steel, and alloy steel, respectively, each having the composition indicated. In these figures, the abscissa is total elongation, or the length of the bar after rolling

[5] The average pressure in cold rolling is directly dependent only upon factors *a* to *e*. The modulus of elasticity of the roll has an indirect effect, the reason for which will be explained later.

divided by the length when the previous hot rolling or annealing was completed. The values of pressure in these three figures are based on the assumption that the rolls remain rigid.

As we observed earlier, in cold rolling the rolls are deformed or flattened very appreciably. This deformation has a distinct effect upon the calculated average pressure, as will be understood after consideration.

Fig. 9 illustrates in cross-section the reduction in thickness of the strip by a roll. Let us assume for a moment that the roll remains rigid and that its surface is truly cylindrical, as shown by the arc marked (10)-(9). A strip the entering surface of which is represented by the line (11)-(10) is being cold rolled and is reduced in thickness to such an extent that its leaving surface will be as represented by the line (9)-(6). The reduction in thickness is then twice the distance (8)-(9), and the projected length of contact is the distance (10)-(8). If the rolling pressure produced in this operation is measured, we can compute the average pressure

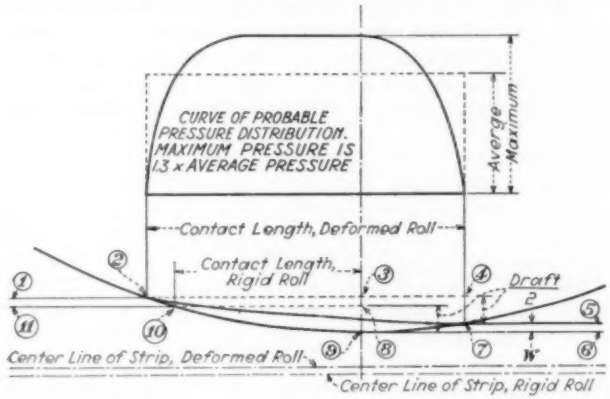


Fig. 9 Reduction in Thickness of the Strip by a Roll

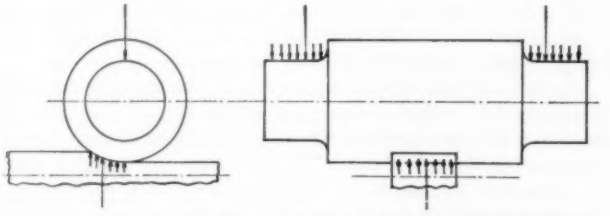


Fig. 11 Arrangement of Forces in Rolling

by dividing the measured load by the projected area of contact. The latter area is, of course, the length (10)-(8) multiplied by the width of the strip. In other words, if the width remains constant, the average pressure is inversely proportional to the length of contact. The average pressure calculated in this way for cold rolling steel is sometimes higher than 500,000 lb. per sq. in.

It is at once apparent that no roll material can remain rigid under such a pressure, and from that premise it follows that there must be deformation of the roll. Assume that the roll surface is flattened by some amount W, as shown in Fig. 9. In order to fulfill the same rolling conditions, the draft must be the same. If the leaving surface of the strip is represented by the line (7)-(5) and the new reduction (7)-(4) is made equal to the former reduction (9)-(8), the conditions will be fulfilled, and the entering surface of the strips will be as indicated by the line (1)-(2). The probable shape of the deformed roll surface is indicated by the curved line (2)-(7). Contact between the roll and strip must now begin at point (2), and extend considerably beyond the vertical center line to point (7). The length of contact therefore becomes (2)-(4), and from that increase over (10)-(8) it is obvious that the calculated average pressure will become very much less.

No really pertinent theory is in existence for this particular case of stress and strain. At the time of writing this paper, the engineers of one of the companies represented on the committee had undertaken the task of developing a satisfactory theory. At this time, it is believed that the most nearly applicable theory is that which was developed by Föppl for the compression of a cylinder between two flat plates. This theory can be found in Vol. 5 of Föppl's book, "Vorlesungen über Technische Mechanik." The arrangement of forces in this case is shown in Fig. 10, and in Fig. 11 is shown the arrangement of forces in rolling. It is evident from a comparison of Figs. 10 and 11 that the conditions in the two cases are somewhat different. With the certainty, however, that the rolls must be flattened under such high pressures, and in the absence of

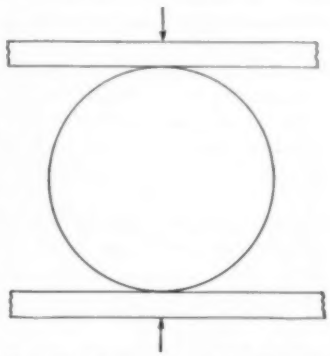


Fig. 10 Arrangement of Forces for a Cylinder Compressed Laterally Between Two Flat Plates

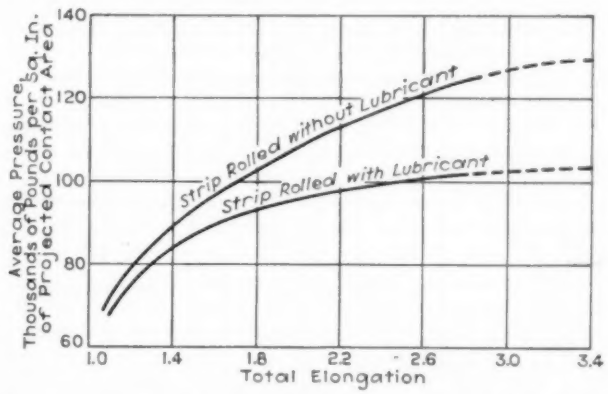


Fig. 12 Effect of Friction Between the Bar and the Rolls at High Compression Rates in Cold Rolling Mild Steel (0.10 Carbon) Based on Deformed Rolls

(For values of contact length/thickness between 6.0 and 17.0.)

a suitable theory to apply to the case, the average pressures based on deformed rolls were computed by means of equations developed from those given by Föppl. These equations, for cast-iron rolls, are:

$$W = 0.386 \times 10^{-7} \times F \left({}^{1}/_{3} + \log_e \frac{2a}{b_1} \right)$$

$$b_1 = 0.392 \times 10^{-3} \sqrt{Fa}$$

In these equations, F is the separating force per inch of width of the strip, a is the roll radius, and W is the deformation.

The curves of average pressure, based on deformed rolls, are shown in Figs. 12, 13, and 14, for the same types of steel and the same tests as those shown in Figs. 6, 7, and 8, respectively. The effect of roll deformation upon the average pressure is clearly shown by a comparison of the curves.

The effect of roll deformation is dependent upon the draft, upon the thickness and hardness of the strip being rolled, and upon the modulus of elasticity of the roll. As we have already noted, the average pressure can be calculated on the basis of rigid rolls. For comparison between cases in which the con-

ditions of draft, hardness, thickness, and roll material are similar, the latter basis is easier; and because of the uncertain accuracy of the foregoing equations in this application, the rigid-roll basis is preferable for such a comparison. In cases with dissimilar conditions of draft, hardness, thickness, or roll material, however, a comparison based on rigid rolls may produce very great error. In such cases, comparison should be made on the basis of deformed rolls because the possible inaccuracies of the formulas

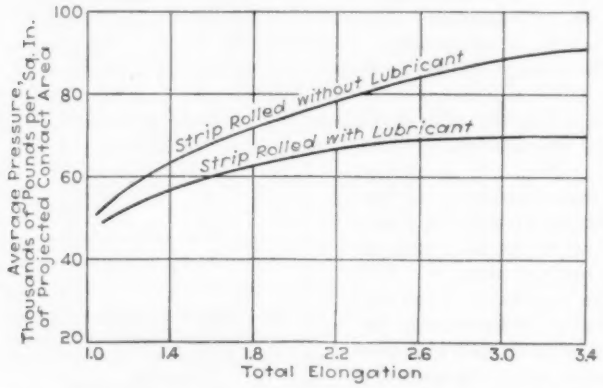


FIG. 13 EFFECT OF FRICTION BETWEEN THE BAR AND THE ROLLS AT HIGH COMPRESSION RATES IN COLD ROLLING HARD STEEL (0.63 CARBON) BASED ON DEFORMED ROLLS

(For values of contact length/thickness between 6.0 and 15.0.)

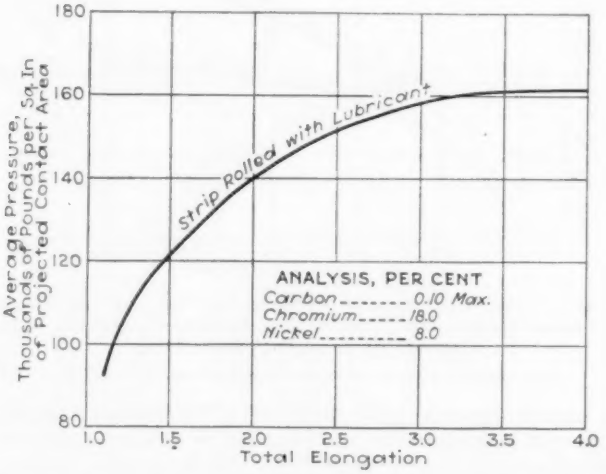


FIG. 14 PRESSURE FOR ALLOY STEEL COLD ROLLED UNDER TENSION AT HIGH COMPRESSION RATES WITH LUBRICANT BASED ON DEFORMED ROLLS

(For values of contact length/thickness between 2 and 5.)

are less than the acknowledged error of assuming rigid rolls. Further study of roll deformation and the consequent verification or modification of Föppl's formulas might well be undertaken by the Society's experts in Applied Mechanics.

It will be observed in Figs. 6, 7, 8, 12, 13, and 14 that for increasing values of total elongation the average pressure increases. This effect is due to change in internal friction of the steel. The hardness and tensile strength of steel are known to be increased by cold deformation, a familiar phenomenon which is called "flow-hardening." It is also known that these qualities are increased further with each successive amount of cold deformation. It is natural to expect that increased hardness of the steel being rolled would result in increased internal friction, and consequently in increased average pressure. Denoting by total elongation the amount of cold deformation to which the steel has been subjected without annealing, it is readily understood why the curves of the figures show increasing pressure with increasing total elongation. This point might be expressed by the statement that the average pressure depends upon the previous treatment of the metal.

The effect of compression rate is also due to internal friction, as was observed in the discussion on hot rolling. A further effect of internal friction is noted in the cold rolling of very thin sheets, in which case the increase in pressure is even more marked than in hot rolling, and for the same reasons.

The effect of external friction between the strip and the rolls can be observed by comparing the curves in Figs. 6, 7, 12, and 13. When a lubricant is introduced between the strip and the rolls, the force due to this friction is reduced and the pressure is decreased also. This also is a cumulative process, for the reasons mentioned in the previous discussion of friction.

The modulus of elasticity of the roll has an indirect effect upon the average pressure, the reason for which can now be made clear. A roll which has a low modulus of elasticity will have a greater deformation and a greater length of contact than a roll which has a high modulus of elasticity, all other effects being

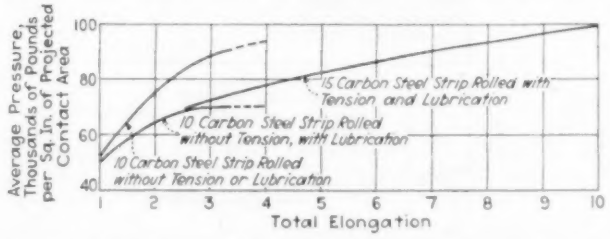


FIG. 15 COLD ROLLING MILD STEEL AT HIGH COMPRESSION RATES BASED ON DEFORMED ROLLS

equal. Increasing the length of contact produces increased friction between the rolls and the bar, with the result that the average pressure must also be increased.

It is believed that when a bar is rolled in a continuous mill and is subjected to tension between successive stands, the average pressure will be less than that which is found under corresponding conditions in a single stand. The reason for this belief is that the tension in the bar, acting in a direction opposite to that of the external friction force, cancels a portion of that force and thereby reduces the average pressure. However, when tension in the bar is used to draw the bar through the rolls of a single stand, and the rolls are driven solely by the friction between them and the bar (see Figs. 8 and 14), the average pressure appears to be approximately the same, for corresponding conditions, as that which is found in a single stand where rolling is done without tension. This fact is shown in Fig. 15, in which the two short curves are those of Fig. 12, drawn to a smaller scale, and in which the long curve represents steel of nearly the same composition cold rolled under tension. In studying these curves it should be noted that two different tests are represented, in which the methods of lubricating the strip and the extent of flow-hardening may differ.

The data upon which the curves shown are based are inadequate to complete our knowledge of rolling pressure. A definite qualitative relation between the average pressure and the factors which determine it has been found. The quantitative relations have been found for some conditions, but for a great many conditions these relations have yet to be determined. As we observed in the beginning, this knowledge is essential to intelligent design of anti-friction bearings for roll-necks. Additional test data are required, therefore, for the accomplishment of our purpose;

and the efforts of this committee will continue to be directed toward the fulfilment of that need.

The information contained herein is released with regret by the authors, because it is as yet incomplete. It is hoped, however, that the information, although incomplete, will be of assistance to those who have to deal with rolling-mill problems.

Discussion

S. M. Weckstein.[6] The paper is interesting and is of much wider scope than appears from a casual reading. To familiarize you further with the object of this paper and the work of the committee, the writer would like to make the following statements:

A few years ago the subject of the application of anti-friction bearings in steel mills and especially in the rolling mills themselves was brought up. At that time very little information was had by the bearing manufacturers and to a large extent by the mill builders and rolling mills as to the actual loads encountered in rolling steel and non-ferrous alloys.

The various bearing manufacturers as well as mill builders independently conducted a number of tests by means of hydraulic cylinders, Brinell plates, and other approved methods to determine these loads, and on the basis of these tests submitted bearing recommendations which were adopted and which have proved to be satisfactory in quite a large number of cases. There was no unified effort, however, to study this problem and to work up standards on which to base recommendations. Through the efforts of the A.S.M.E. and through the able assistance of Professor Trinks, a committee was formed to investigate this subject. This paper covers generally the few preliminary reports which have been made by the committee. The work which the committee has done so far is merely a small beginning of the work which is ahead of them. All of the bearing manufacturers, the mill builders, and quite a few of the rolling mills are cooperating with the committee in extending this information.

Just recently the Timken Roller Bearing Company finished a complete series of tests on the various rolling mills in its plant. Preliminary results of these tests have already been published by our Mr. Waldorf in recent issues of *The Electrical Iron & Steel Engineer*.

Information is now being turned over to the committee for further analysis, and these data will be included in forthcoming reports of the committee. Since this test covers the rolling of various alloy steels in a 35-in. blooming mill and 28- and 22-in. bar mills and in a continuous 10-in. stand 18- to 10-in. bar mill, and was also run on plain bearings and on Timken bearings, the information obtained from this test should be quite a contribution to the industry.

The cooperation of all the rolling mills is requested in assisting and further increasing the sources of information for the committee. As a representative of one of the bearing companies which is interested in this work, the writer feels quite certain that the data which this committee will compile will be not only of extreme value to the bearing manufacturers, but will also be a great help to the mill builders and mill operators. The application of anti-friction bearings in rolling mills is at the present time in its infancy, and the field is important enough to deserve everybody's closest study.

W. Trinks.[7] As a co-author, I should not be called upon to discuss the paper. Nevertheless a few remarks are in order. The data presented in the paper apply only to iron and steel. Data on other metals have also been collected, but were not included in this paper, because of its being read under the auspices of the Iron and Steel Division. It should be mentioned that the investigation of the forces involved in rolling is only the first step. When that investigation shall have been completed, the next step consists in analyzing the stresses which occur in the roll body and the roll neck. The committee is already at work on this problem, and hopes to have results of great interest to present in the near future. All iron and steel works in possession of test data have willingly contributed them. Also the manufacturers of metals other than iron and steel have, with one exception, been very willing to detail their experience.

The work of the committee is financed by contributions of $200 each from companies interested in the work. No difficulty was attached to securing contributions.

With regard to the application of high-duty anti-friction bearings, the following may be remarked. A few months ago the writer had the opportunity to go through several mills employing such bearings. In one of the mills the superintendent led the way to the "boneyard" and showed a large number of bearings which had failed in service. He said that in each case failure could be traced back to the failure of an individual roller and that the investigation of that roller showed that internal cracks or other defects were responsible. It therefore appears that it would be advisable to test each individual roller, either by X-ray analysis or by magnetic analysis, so as to positively eliminate the danger of internal defects.

The cost of high-duty anti-friction bearings is so great and the cost of a shutdown is so great that the cost of inspection of each individual roller by one of the cited methods should be only a small fraction, a very small fraction indeed, of the total cost of the bearing.

J. L. Young.[8] Professor Trinks raised the question that the various bearing manufacturers should do everything within their power to prevent flaws and defects in the rolls of their bearings in order to eliminate the difficulties he cited.

He referred especially to his recent trip to a steel mill in the Chicago district where he saw a number of failed straight-roll back-up roll-neck bearings from a four-high mill and the manager of that plant stated that in nearly every case they could trace the failure to the breakage of the rolls and to a defect or flaw in one or more rolls in that bearing. He suggested the use of the X-ray test or the Sperry magnetic test as a final inspection of bearings to determine these defects.

The point raised by Professor Trinks is a good one, but the writer believes there is another angle from which to approach the subject of defects in the rolls of large bearings, for the type of steel and method of heat treatment have as much or more to do with the elimination of defects as the proposed extra final inspection to catch these defects. The rolls referred to were made of a high-carbon chrome-type steel.

The Timken Company has experimented for years with the object in view of making the best possible bearing. Based upon those experiments we have arrived at a conclusion that a properly carburized and heat-treated large section bearing of proper analysis is superior to a bearing of similar section made of high-carbon chrome steel.

These are certain inherent difficulties that accompany the processing and heat-treating of large sections of high-carbon chrome steel. They are to some extent eliminated by the use

[6] Industrial Equipment Engineer, Timken Roller Bearing Co., Canton, Ohio. Jun. A.S.M.E.

[7] Professor of Mechanical Engineering, Carnegie Institute of Technology, Pittsburgh, Pa. Mem. A.S.M.E.

[8] Development Engineer, The Timken Roller Bearing Company, Pittsburgh, Pa.

of a proper carburizing steel that will give the same hardness for the bearing surface that is possible to obtain with high-carbon chrome steel and at the same time give a tough and softer core of proper gradation. This will eliminate the brittleness of the entire roll.

Based upon experiments, the Timken Company has adopted a nickle-chrome carburizing steel that when properly carburized and heat-treated gives the desired hardness gradation. The success of large section Timken bearings in steel mills is sufficient proof that this method is correct.

The original bearings on two installations of four-high mills in the Pittsburgh district are now entering their fifth year of successful service.

We are now producing a quantity of bearings for a new four-high mill, to be located in the Chicago district, that will be the largest rolling-mill bearings ever made. The neck diameter will be $29^1/_2$ in. The bearing will be $46^1/_2$ in. and the length will be 29 in. Each bearing will weigh about $3^1/_2$ tons without the chock.

Instead of the X-ray or magnetic methods of final inspection, the Timken Company has developed a radio testing machine that is now used extensively throughout the automotive industry and which apparatus gives us a final check on the condition of the bearings from a standpoint of surface defects that cannot be seen by the visual method. This machine is nothing more than a radio that transmits the noise developed by rotating a bearing under load to the dial of a galvanometer. If there is any sort of a surface defect that may interfere with the proper functioning of the bearing, it will cause an over-noise that will be transmitted through the radio to the galvanometer.

The Timken Company is vitally interested in manufacturing the best possible anti-friction bearings for steel mill service, and if by the use of either the X-ray or the magnetic methods of testing we can better our product, we will gladly resort to those methods, but so far we feel that the thirty-two or more inspections that a bearing receives in our plant before it is shipped to the customer allows the production of a bearing that is as nearly correct as is commercially possible to produce.

Index photographed at the
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